University of Maribor

Faculty of Chemistry and
Chemical Engineering

PROCESS EQUIPMENT
Instructions for exercises

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1 VESSELS

1.1 PRESSURE VESSELS DIMENSIONING

Pressure vessels made of Mb13 (Č.1202) should be dimensioned. The shell is sealed with a unilateral butted weld. The bottom is elliptical in shape and coupled to the shell by a butted one side weld. At the bottom are two openings with external diameters of 50 mm to release and drain the fluid in the pressure vessel. The lid and the bottom are joined by screws via a welded flange. The volume of the liquid phase is 1.1 m^3, to take up 70% of the volume of the pressure vessel. Operating pressure in the vessel is 700 kPa.

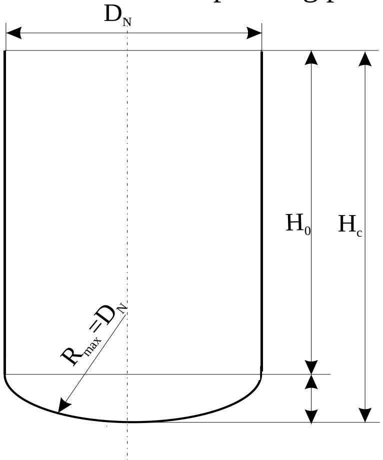


Figure 1.1: Characteristic dimensions of a pressure vessel

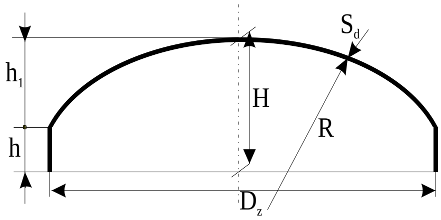


Figure 1.2: Half elliptical bottom of the vessel

D_N	–	inner diameter
H_0	–	wall height
h_1	–	the height of the elliptical part of the bottom/lid
h	–	the height of the cylindrical part of the bottom/lid

Pressure vessel characteristic dimensions

$V_p' = \frac{V_T}{0.7} = 1.57\ \mathrm{m}^3$ rounded $V_p = 1.6\ m^3$

V_T – liquid volume
V_p' – minimal vessel capacity without lid

The characteristic dimensions for pressure vessels are taken from Table 1.9:
D_N= 1000 mm H_0= 2000 mm H_c= 2275 mm

The characteristic dimensions of the bottom are selected from table 1.11, according to previously selected D_N:
h_1= 250 mm
h= 40 mm
V_d= 162 dm³ V_d – elliptical bottom volume

The actual volume of the pressure vessel without its lid:

$$V_p = V_o + V_d = \frac{\pi \cdot D_N^2}{4} \cdot H_o + V_d = 1.733 \text{ m}^3$$

V_0 – the pressure vessel capacity without lid and bottom

Determination of the pressure vessel shell (equations 1.9 and 1.7)

$$S_0 = \frac{D_N \cdot p}{2.3 \cdot z \cdot \frac{K}{\nu} \cdot \varphi - p}$$

$$S_1 = S_0 + c_1 + c_2$$

p - pressure in vessel,
K - strength number (table 1.5),
ν - safety coefficient (table 1.6),
φ - resistance factor of longitudinal weld (table 1.7)
z - factor due to openings
S_0 - theoretical pressure vessel shell thickness without empirical supplements,
S_1 - theoretical pressure vessel shell thickness with empirical supplements c_1 in c_2,
S_2 - rounded pressure vessel shell thickness,
c_1 - corrosion supplement,
c_2 - supplement for plate thickness tolerances (table 1.1)

z = 1, because there are no openings in the shell.

Data for K, ν in φ are taken from table:. K= 230 N/mm^2 ν = 1.5 φ= 0.7

S_0 is calculated according to above equation: S_0=0.0028 m.

The supplements are determined and S_1 is calculated:
c_1=1 mm; c_2=0.6 mm
S_1=4.4 mm

Determination of half – elliptical bottom thickness (equation 1.12)

$$S_0 = \frac{p \cdot D_N}{4 \cdot z \cdot \frac{K}{\nu} \cdot \varphi - p} \cdot \frac{D_N}{2 \cdot h_1} \qquad z = \prod_i (1 - \frac{d_{zi}}{D_N})$$

$$z_1 = 1 - \frac{d_{z1}}{D_N} \qquad z_2 = 1 - \frac{d_{z2}}{D_N}$$

$$z = z_1 \cdot z_2 = 0.9025$$

S_0 is calculated according to the above equation S_0=0.0036 m.

The supplements are determined and S_1 is calculated:
c_1=1 mm; c_2=0.6 mm

S_1=5.2 mm

Determination of the half – elliptic lid thickness

The fillet is usually used for the flanges, that is why $\varphi = 0.8$:

$$S_0 = \frac{p \cdot D_N}{4 \cdot z \cdot \frac{K}{\nu} \cdot \varphi - p} \cdot \frac{D_N}{2 \cdot h_1}$$

z = 1, because there are no openings

S_0 is calculated according to the above equation S_0 = 0.0029 m.

The supplements are determined and S_1 is calculated:
c_1=1 mm; c_2=0.6 mm

S_1=4.5 mm

The shell, bottom and lid are made from plates with the same thickness:
S_2 = 6 mm

1.2 FLANGES

Define the required diameter of the screw for fastening the lid flange on the pressure vessel shell flange. The operating pressure in the vessel is 0.26 MPa and the temperature is 100°C. The pressure vessel inner diameter is 1350 mm and the width of the membrane steel sealing ring is 45 mm. The screws are made of Č.0645.5 material.

$p = 0.26$ MPa
$D_N = 1350$ mm
$D_T = 1395$ mm the average sealing diameter
$b_T = 45$ mm sealing diameter

Data from table 1.12
$k_0 = b_T = 0.045$ m

Data from table 1.13
$\sigma_T = 360\ \frac{\text{N}}{\text{mm}^2}$

Data from table 1.4
$\nu_o := 1.5$ operation
$\nu_M := 1.1$ mounting

Deformation resistance of seal material
$k_D = 380\ \frac{\text{N}}{\text{cm}^2}$

S_B – supplement depending on seal type
$S_B = 1.3$

The force exerted on the screws in flanges (equation 1.21):
$F = F_R + F_p + F_0 = 4.714 \cdot 10^5$N

F_R –force due to inner pressure
$F_R = \frac{p \cdot \pi \cdot D_N^2}{4} = 3.722 \cdot 10^5$ N

F_P – the force on the front flange
$F_p = \frac{p \cdot \pi \cdot (D_T^2 - D_N^2)}{4} = 2.522 \cdot 10^5$ N

F_0 – sealing force during operation
$F_0 = p \cdot \pi \cdot D_T \cdot k_1 \cdot S_B = 7.406 \cdot 10^4$N

F_M – sealing force during mounting
$F_M = \pi \cdot D_T \cdot k_0 \cdot k_D = 7.494 \cdot 10^5$N

The screw number is selected as a multiple of 4. For example N=20.

Force determination on 1 screw:

- during operation
  $F_{v0} = \frac{F}{N} = 2.357 \cdot 10^4$N
- during mounting
  $F_{v0} = \frac{F_M}{N} = 3.747 \cdot 10^4$N

Determination of screw diameter without thread:

- according to operating force

$$d_{1,0} = \sqrt{4 \cdot \frac{F_{v0}}{\pi \cdot \frac{\sigma_T}{\nu_0}}} + c_3 = 0.014 \text{ m}$$

- according to mounting force

$$d_{1,M} = \sqrt{4 \cdot \frac{F_{vM}}{\pi \cdot \frac{\sigma_T}{\nu_M}}} + c_3 = 0.015 \text{ m}$$

According to table 1.13, the M20 screws are selected.

1.3 CYLINDRICAL WALLS UNDER THE INFLUENCE OF EXTERNAL PRESSURE

Determine the required thickness of inner and outer pressure vessel shells. The pressure during the operation is 0.27 MPa at 80°C. The pressure in the steam heated shell is 0.34 MPa at 200°C. The vessel is made of Mb13 steel with a – test.
The vessel shell is sealed automatically by a bilateral butted seam. The steam pipes are welded to the outer layer with a unilateral butted seam.

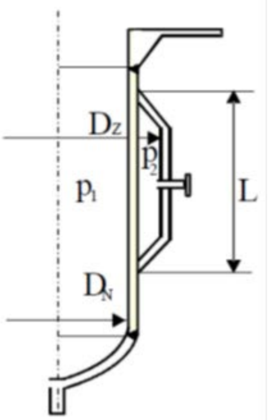


Figure 1.3: Cylindrical vessel wall

$T_1 = 80°\mathrm{C}$
$p_1 = 0.27 \cdot 10^6 \,\frac{\mathrm{N}}{\mathrm{m}^2}$
$T_2 = 200°\mathrm{C}$
$p_2 = 0.34 \cdot 10^6 \,\frac{\mathrm{N}}{\mathrm{m}^2}$
$D_N = 2000 \text{ mm}$
$D_z = 2160 \text{ mm}$
$d_z = 159 \text{ mm}$
$D_N = 2000 \text{ mm}$
$l = 960 \text{ mm}$
$E = 1.95 \cdot 10^5 \,\frac{\mathrm{N}}{\mathrm{mm}^2}$

Data from table 1.12:
$K = 180 \,\frac{\mathrm{N}}{\mathrm{mm}^2}$
$z = 1$
$v = 1.5$

$\varphi = 1$
$c_1 = 1 \text{ mm}$
$c_2 = 0.6 \text{ mm}$

The wall thickness determination for the inner vessel:

$$S_{p0} = \frac{p_1 \cdot D_N}{2.3 \cdot z \cdot \frac{K}{v} \cdot \varphi - p_1} = 0.00196 \text{ m}$$

$$S_{p1} = S_{p0} + C_1 + C_2 = 0.04 \text{ m}$$

Determination of the wall thickness according to overpressure in inter – layer area:
$U = 1.5$

$$D_{sr} = \frac{D_z + D_N}{2} = 2.08 \text{ m}$$

$$S_{p01} = \frac{1 + \sqrt{1 + 0.24 \cdot U \cdot \frac{K}{v \cdot p_2} \cdot \frac{\left(1 + 0.1 \cdot \frac{D_{sr}}{l}\right)}{\left(1 + 5 \cdot \frac{D_{sr}}{l}\right)}}}{\frac{4}{D_{sr}} \cdot \frac{K}{v \cdot p_2} \cdot \left(1 + 0.1 \cdot \frac{D_{sr}}{l}\right)} = 0.006 \text{ m}$$

$$S_{p11} = S_{p01} + C_1 + C_2 = 0.00735 \text{ m}$$

The wall thickness determination for the inner vessel according to elastic deformation

Firstly, it should be determined if this is a short or long cylindrical wall, table 1.2.

Conditions for short cylindrical wall:

$$\left(\frac{l}{D_{sr}}\right)^2 < 1.4 \cdot \frac{D_{sr}}{Y}$$

Y=0.00575

$$S_{p02} = 2.7 \cdot 10^{-3} \cdot D_{sr} \left(\frac{p_2 \cdot l \cdot v}{D_{sr} \cdot E \cdot 10^{-6}}\right)^{\frac{2}{5}}$$

$$S_{p12} = S_{p02} + C_1 + C_2$$

Determination of outer wall thickness:

$$z_1 = 1 - \frac{d_z}{D_N} = 0.9264$$

$$z_{11} = z_1 \cdot z_1 = 0.8582$$

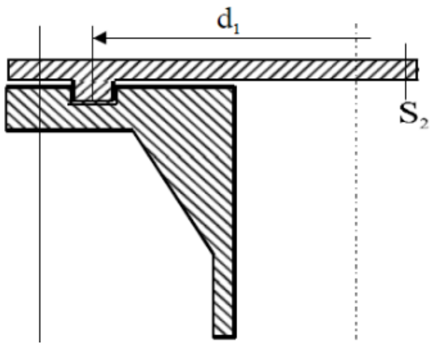
$$\varphi_1 = 0.7$$

$$S_{p03} = \frac{p_2 \cdot D_z}{2.3 \cdot z_{11} \cdot \frac{K}{v} \cdot \varphi_1 - p_2} = 0.0044 \text{ m}$$

$$S_{p13} = S_{p03} + C_1 + C_2 = 0.00604 \text{ m}$$

1.4 FLAT BOTTOMS AND LIDS

Determine the required thickness of a flat lid fixed by screws, if the operating pressure is 2.5 MPa at 20°C. The material is Mb 19 steel with a- test.



p = 2.5 MPa
D_N = 662 mm
D_{NT} = 649 mm
$D0$ = 770 mm

Figure 1.4: Flat vessel lid

Data from table 1.5:
K = 270 N/mm^2

Data from table 1.6:
$\nu := 1.5$

The ratio D_0/D_N is determined and from table 1.3, coefficient C is evaluated:

$$\frac{D_0}{D_N} = 1.163$$

$C := 0.58$

The lid thickness is calculated:

$$S_0 = C \cdot D_N \cdot \sqrt{p \cdot \frac{\nu}{K}} = 0.045 \text{ mm}$$

The supplements are determined and S1 is calculated;
$c_1=0$
$c_2=0$

$$S_1 = S_0 + C_1 + C_2 = 0.045 \text{ m}$$

The rounded steel thickness is:
$S_2 = 46$ mm

1.5 PRESSURE VESSEL SHELL

Determine the shell thickness of a pressure vessel on which is an opening Φ50/59. The material is Mb 19 steel with a – test. The operating pressure is 1.5 MPa at 200°C. The shell is sealed with a unilateral butted seam. The vessel inner diameter is 700 mm.

$p = 1.5$ MPa
$D_N = 700$ mm
$d_z = 59$ mm

Data from table 1.5:
K = 230 N/mm²

Data from table 1.6:
$\upsilon = 1.5$

Data from table 1.7:
$\varphi \quad = 0.7$

$$z = 1 - \frac{d_z}{D_N} = 0.916$$

$$S_{p0} = \frac{p \quad \cdot D_N}{2.3 \cdot z \quad \cdot \frac{K}{\upsilon} \cdot \varphi \quad - p} = 0.005 \text{ m}$$

$$S_1 = S_0 + C_1 + C_2 = 0.006 \text{ m}$$

$$S_2 = 7 \text{ mm}$$

2 SEPARATORS

2.1 GAS - LIQUID SEPARATORS

2.1.1 Vertical gas – liquid separator

Determine the characteristic dimensions of a vertical gas - liquid separator for the following operating conditions:
- gas flow rate 1500 m^3/h
- liquid flow 7 m^3/h
- gas density 13 kg/m^3
- liquid density 950 kg/m^3.

$q_{vg} = 1500\ \mathrm{m^3/h}$
$q_{vl} = 7\ \mathrm{m^3/h}$
$\rho_g = 13\ \mathrm{kg/m^3}$
$\rho_l = 950\ \mathrm{kg/m^3}$

Determination of the gas limited velocity (equation 2.3):
K = 0.0692 m/s

Constant K is usually in the range of 0.0305 – 0.1067 m/s.

$$v_m = K\left(\frac{\rho_l - \rho_g}{\rho_g}\right)^{\frac{1}{2}} = 0.587\ \mathrm{m/s}$$

Determination of permitted gas velocity (equation 2.4):
$v_a = 0.15 \cdot v_m = 0.088\ \mathrm{m/s}$

Separator diameter (equation 2.5):
$A = \dfrac{q_{vg}}{v_a} = 4.728\ \mathrm{m^2}$
Rounded diameter:
d = 2.5 m

The retention time of the liquid in the separator is usually in the interval of 5 – 10 min. It is supposed that:
t = 5 min

Determination of separator volume:
$V_l = q_{vl} \cdot t = 0.583\ \mathrm{m^3}$

The actual separator surface:
$A_1 = d^2 \cdot \dfrac{\pi}{4} = 4.909\ \mathrm{m^2}$

The liquid height in the separator:

$$h_0 = \frac{V_l}{A_1} = 0.119 \text{ m}$$

Rounded value:

$$h_1 = 0.15 \text{ m}$$

The other dimensions of the separator are determined according to figure 2.1.

- The height from the connection to the top of the separator:

$$h_3 = d = 2.5 \text{ m}$$

- Height from the liquid level to the connection:

$$h_2 = \frac{d}{2} = 1.25 \text{ m}$$

- The entire height of the separator:

$$h = h_1 + h_2 + h_3 = 3.9 \text{ m}$$

The ratio between the height and the separator diameter must be $h/d \geq 3$

$$\frac{h}{d} = 1.56$$

$$h_{larger} = 3 \cdot d = 7.5 \text{ m}$$

The calculated separator is too high, so we calculate the separator with a built-in grid for discharging liquid and gas droplets.

The grid is selected depending on the separation degree (Table 2.1):
K_1 = 0.1219 m/s

The grid flood velocity is checked (equation 2.7):

$$X = \frac{q_{vl} \cdot \rho_l}{q_{vg} \cdot \rho_g} \cdot \left(\frac{\rho_g}{\rho_l}\right)^{\frac{1}{2}} = 0.04$$

The ratio between grid flood velocity and average velocity:
a = - 0.0022 m/s g = 0.0802 m/s

$$K_2 = a + \frac{g}{X^{1.294} + 0.573} = 0.134 \text{ m/s}$$

$$K_{1p} = \frac{K_2}{1.2} = 0.112 \text{ m/s}$$

Allowed gas velocity in the separator

$$v_{m1} = K_1 \cdot p \left(\frac{\rho_l - \rho_g}{\rho_g}\right)^{\frac{1}{2}} = 0.949 \text{ m/s}$$

New separator diameter is determined

$$d_1 = \left(\frac{4 \cdot q_{vg}}{\pi \cdot v_m}\right)^{0.5} = 0.95 \text{ m}$$

The required surface for the liquid volume is 0.442 m^2, and the rounded liquid height is 1.4m.

The separator dimensions are evaluated according to figure 2.1:
$f = h_l = 1.4$ m
$a = 0.3$ m
$b = 0.15$ m
$c = 0.4$ m
$e = 0.3$ m

2.1.2 Horizontal gas – liquid separator

Determine the characteristic dimensions of a horizontal gas – liquid separator for the following operating conditions:

- Gas flow 550 m^3/h
- Liquid flow 35 m^3/h
- Gas density 25 kg/m^3
- Liquid density 950 kg/m^3
- The retention time for liquid 10 min
- Operating pressure $30 \cdot 10^5$ Pa.

$q_{vg} = 550\ \mathrm{m^3/h}$
$q_{vl} = 35\ \mathrm{m^3/h}$
$\sigma_g = 25\ \mathrm{kg/m^3}$
$\sigma_l = 950\ \mathrm{kg/m^3}$

$t_l = 10\ \mathrm{min}$
$P = 30 \cdot 10^5$ Pa
$K = 0.0692$ m/s

Determination of the limited velocity

$$v_m = K\left(\frac{\sigma_l - \sigma_g}{\sigma_g}\right)^{\frac{1}{2}} = 0.421\ \mathrm{m/s}$$

The allowed velocity (equation 2.4)

$$v_a = 0.15 \cdot v_m = 0.063\mathrm{m/s}$$

The free surface area above the liquid is 0.2 of the total area (Figure 2.2)

$f_{ag} = 0.2$ $f_{hg} = 0.25$ $f_{al} = 1 - f_{ag} = 0.8$

According to table 2.2, r is determined as:
$r = 4$

The separator diameter is determined as:

- According to equation 2.8

$$d_{28} = 1.1284 \cdot \sqrt{\frac{q_{vg} \cdot f_{hg}}{f_{ag} \cdot r \cdot v_a}} = 0.981\ \mathrm{m}$$

- According to equation 2.9

$$d_{29} = 1.084 \cdot \sqrt[3]{\frac{q_{vl} \cdot t_l}{f_{al} \cdot r}} = 1.324\ \mathrm{m}$$

The calculation is done for larger *d*.

Recalculating the height of the free space above the liquid
$h_g = f_{hg} \cdot d_{29} = 0.331\ \mathrm{m}$

The height of the space is less than the permissible (380 mm); therefore, a calculation is made for: $f_{al2} = 0.7$ $f_{hg2} = 0.3$ $h_g = f_{hg} \cdot d_{29} = 0.331$ m

$$d_{nov} = 1.084 \cdot \sqrt[3]{\frac{q_{vl} \cdot t_l}{f_{al2} \cdot r}} = 1.384\ \mathrm{m}$$

$h_g = f_{hg2} \cdot d_{nov} = 0.415$ m

The rounded separator diameter:
$d_{novoz} = 1.4\ \mathrm{m}$

The exact retention time is calculated:

$$t_{dej} = d_{novoz}^3 \cdot r \cdot \frac{f_{al2}}{1.084^3 \cdot q_{vl}} = 620.424\ \mathrm{s}$$

The separator length:
$L = r \cdot d_{novoz} = 5.6\ \mathrm{m}$

2.1.3 Centrifugal gas – liquid separator

Determine the characteristic dimensions of a centrifugal gas – liquid separator under the following operating conditions:

- Mass flow of water vapour 1000 kg/h
- Urea mass flow 2000 kg/h
- Operating pressure 40000 Pa
- operating temperature 80°C
- urea density 1185 kg/m³.

$q_{mp} = 1000\ \text{kg/h}$
$q_{mr} = 2000\ \text{kg/h}$

$p = 40000\ Pa$
$T = 353\ K$

$\rho = 1185 \dfrac{kg}{m^3}$

Determination of water vapour volume flow:

$$\rho_g = \frac{p \cdot M}{R \cdot T} = 0.245\ \text{kg/m}^3$$

$$q_{vg} = \frac{q_{mp}}{\rho_g} = \frac{1.132\ \text{m}^3}{s}$$

Determination of urea volume flow:

$$q_{vl} = \frac{q_{mr}}{\rho} = 4.688 \cdot 10^{-4}\ \text{m}^3/\text{s}$$

The entire volume flow:

$$q_v = q_{vg} + q_{vl} = 1.133\ \text{m}^3/\text{s}$$

Standard pipe diameter selected for feed pipe:
$d_z = 0.15405\ \text{m}$

The inlet velocity for the gas – liquid mixture:

$$v = \frac{4 \cdot q_v}{\pi \cdot d_z^2} = 60.774\ \text{m/s}$$

Inlet velocity should be in the range of 30 to 120 m/s. For a gas-liquid system, the rate of velocity of the mixture in the separator must be 0.002 to 0.2 times the value of the inlet velocity.
$v_z = v \cdot 0.101 = 6.138\ \text{m/s}$

Velocity for water vapour:

$$v_a = 0.1885 \cdot v_z \cdot \left(\frac{\rho - \rho_g}{\rho_g}\right)^{0.25} = 9.645\ \text{m/s}$$

Diameter of centrifugal separator:

$$d = \sqrt{\frac{4 \cdot q_v}{\pi \cdot v_a}} = 0.387\ \text{m}$$

The height of the centrifugal separator is determined according to conditions in figure 2.4:
$L = 2 \cdot 0.6\ \text{m} = 1.2\ \text{m}$

The diameter of the outlet pipe should remain the same as the diameter of the inlet pipe.

2.2 LIQUID – LIQUID SEPARATORS

2.2.1 Vertical liquid – liquid separator

Determine the characteristic dimensions of a vertical liquid – liquid separator for the following conditions:

	Hydrocarbons	Water
Volume flow	2.0 m^3/h	3.0 m^3/h
Density	850 kg/m^3	1000 kg/m^3
Dynamic viscosity	$3 \cdot 10^{-3}$ Pas	$1 \cdot 10^{-3}$ Pas

The diameter of the liquid drops is 100 μm.

$$q_{vl} = 2\ \frac{\mathrm{m}^3}{3600\ \mathrm{s}}$$

$$\rho_l = 850\ \mathrm{kg/m^3}$$
$$\rho_t = 1000\ \mathrm{kg/m^3}$$
$$\mu_l = 3 \cdot 10^{-3} Pas$$

$$\mu_t = 1 \cdot 10^{-3} Pas$$
$$d_d = 100\ \mathrm{\mu m}$$

$$q_{vt} = 3\ \frac{\mathrm{m}^3}{3600\ \mathrm{s}}$$

1. Determination of the continuum phase (Equation 2.13):

$$\Theta = \frac{q_{vl}}{q_{vt}} \cdot \left(\frac{\rho_l \cdot \mu_t}{\rho_t \cdot \mu_l}\right)^{0.3} = 0.457$$

According to Table 2.3, the lighter phase is dispersed and the water is a continuous phase.

2. The velocity for drops separation:

$$v = \frac{g \cdot d_d^2 \cdot (\rho_l - \rho_t)}{18 \cdot \mu_t} = -8.172 \cdot 10^{-4}\ \mathrm{m/s}$$
$$v_{abs} = 8.172 \cdot 10^{-4}\ \mathrm{m/s}$$

Negative signs indicate that drops are moved upward.

3. The flow rate of the water must be less than the rate of separation of hydrocarbon droplets:

$$d = \sqrt{\frac{4 \cdot q_{vt}}{\pi \cdot v_{abs}}} = 1.139\ \mathrm{m}$$

4. Rounded:

$$d = 1.2\ \mathrm{m}$$

The height of the separator should be 2.5× larger than the diameter: $h = L = 3$ m
The layer of dispersed liquid mixture should occupy 15% of the height of the separator: h=0.45 m

5. Check the size of the drops to be taken away with hydrocarbons (in this case, hydrocarbons are a continuous phase):

$$A = \frac{\pi \cdot d^2}{4} = 1.131 \text{ m}^2$$

6. Flow rate of hydrocarbons:

$$v_1 = \frac{q_{vl}}{A} = 4.912 \cdot 10^{-4} \text{ m/s}$$

7. Determine d_d

$$d_d = \sqrt{\frac{18 \cdot v_{abs} \cdot \mu_t}{g \cdot (\rho_l - \rho_t)}} = 140\ \mu\text{m}$$

The droplet size is greater than 100 μm, thus increasing the diameter of the separator: $d_1 = 1.5$ m

$$A_1 = \frac{\pi \cdot d_1^2}{4} = 1.767 \text{ m}^2$$

$$v_2 = \frac{q_{vl}}{A_1} = 3.144 \cdot 10^{-4} \text{ m/s}$$

$$d_{d1} = \sqrt{\frac{18 \cdot v_2 \cdot \mu_t}{g \cdot (\rho_l - \rho_t)}} = 100\ \mu\text{m}$$

8. To minimize the effect of the mixture jet on the separation of the two-phase mixture at the separator inlet, limit the flow velocity to 1 m / s.

9. Total volume flow:

$$q_v = q_{vt} + q_{vl} = 0.001 \ \frac{\text{m}^3}{\text{s}}$$

10. Inlet tube intersection area:

$$A_{cev} = \frac{q_v}{v} = 0.002 \text{ m}^2$$

11. Inlet tube diameter:

$$d_c = \sqrt{\frac{4 \cdot A_{cev}}{\pi}} = 0.042 \text{ m}$$

12. Rounded on standard pipe: *d* = 50 mm
The supply point is at the bottom of the separator, the inlet at the half-tank height and the hydrocarbon outlet at 90% of the separator height.

13. The dimensions of the separator are:
- diameter *d* = 1.5 m
- height *h* = 3.75 m
- Connection for flow inlet *h* = 1.88 m
- Connection for hydrocarbon outlet *h* = 3.38 m

2.2.2 Horizontal gravity liquid – liquid separator

Determine the characteristic dimensions of a horizontal gravity liquid – liquid separator for the following conditions:

- Hydrocarbon volume flow 0.006 m^3/s
- Hydrocarbon density 520 kg/m^3
- Hydrocarbon dynamic viscosity 0.1 $\cdot 10^{-3}$ Pas
- Water volume flow 0.0025 m^3/s
- Water density 1100 kg/m^3
- Dynamic viscosity 1.5 $\cdot$ 10^{-3} Pas.

$q_{v1} = 0.006\ \frac{\mathrm{m}^3}{\mathrm{s}}$ $\qquad$ $q_{v2} = 0.0025\frac{\mathrm{m}^3}{\mathrm{s}}$

$\rho_l = 520\ \mathrm{kg/m^3}$ $\qquad$ $\rho_t = 1100\ \mathrm{kg/m^3}$

$\mu_l = 0.1 \cdot 10^{-3}\mathrm{Pas}$ $\qquad$ $\mu_l = 1.5 \cdot 10^{-3} Pas$

Determination of droplet separation rate (assuming a size of 100 µm):

$d_d = 100\ \mu\mathrm{m}$

$$v_l = \frac{g \cdot d_d^2 \cdot (\rho_t - \rho_l)}{18 \cdot \mu_l} = 0.032\ \mathrm{m/s}$$

$$v_t = \frac{g \cdot d_d^2 \cdot (\rho_t - \rho_l)}{18 \cdot \mu_t} = 0.002\ \mathrm{m/s}$$

Maximum separation velocity:

$$v_{max} = 1.2 \cdot v_{min} \cdot \frac{q_{v1}}{q_{v2}} = 0.006\ \mathrm{m/s}$$

Since the maximal speed is greater than 0.00423 m/s, the value is set as:

$v_l = 0.00423\ \mathrm{m/s}$

$q_v = q_{v1} + q_{v2} = 0.009\ \mathrm{m^3/s}$

It is assumed that:

$r = 3$

$$d = 1.6351 \cdot \sqrt{\frac{q_v}{(v_l + v_t) \cdot r}} = 1.093\ \mathrm{m}$$

Separator length:

$L = r \cdot d = 3.28\ \mathrm{m}$

The height of the lighter layer:

$$h_l = 0.3142 \cdot d^2 \cdot \frac{v_l}{q_v} \cdot L = 0.613\ \mathrm{m}$$

The height of water layer:

$$h_t = 0.3142 \cdot d^2 \cdot \frac{v_t}{q_v} \cdot L = 0.305 \text{ m}$$

Height control of individual layers, which must be between 0.3 and 0.7:

$$\frac{h_l}{d} = 0.561$$

$$\frac{h_t}{d} = 0.279$$

Because the ratio is less than 0.3, the calculation should be repeated with reduced *r*:

$$r_{new} = 2.5$$

Separator dimensions:
d= 1.198 m
l = 2.995 m
h_L=0.737 m
h_T = 0.366 m

h_L/d=0.616

h_T/d = 0.306

For these operating conditions, we select a separator with a diameter of 1.198 m and a length of 2.99 m, which means that the nearest standard vessel is selected.

2.2.3 Horizontal gravity liquid – liquid separator with a barrier

Determine the characteristic dimensions of a horizontal gravity separator with a barrier at the following operating conditions:

$q_{vt} = 0.0025\ \frac{\mathrm{m}^3}{\mathrm{s}}$

$q_{vl} = 0.006\frac{\mathrm{m}^3}{\mathrm{s}}$

$\rho_l = 520\ \mathrm{kg/m^3}$

$\mu_l = 0.1 \cdot 10^{-3}\mathrm{Pas}$

$\rho_t = 1100\ \mathrm{kg/m^3}$

$\mu_t = 1.5 \cdot 10^{-3}\mathrm{Pas}$

$t = 300\ \mathrm{s}$

1. Determination of the continuum phase (Equation 2.13):

$$\Theta = \frac{q_{vl}}{q_{vt}} \cdot \left(\frac{\rho_l \cdot \mu_t}{\rho_t \cdot \mu_l}\right)^{0.3} = 4.319$$

According to Table 2.3, it is clear that the lighter phase is a continuous phase.

2. Determination of the retention time of both liquids in the separator:

$$t_1 = 3.6 \cdot 10^8 \cdot \frac{\mu_l}{\rho_t - \rho_l} = 62.069\ \mathrm{m^2/s}$$

The retention is short, so take t = 10 min.

3. Determination of the diameter of the separator for r = 3:

$$d = \sqrt[3]{\frac{(q_{vt} + q_{vl}) \cdot t_2 + 1.712 \cdot q_{vl} \cdot t}{0.6723 \cdot r + 0.2241}} = 1.54\ \mathrm{m}$$

4. Length of the separator:

$l = r \cdot d = 4.619$ m

5. Length of individual sections separated by barrier:

$$L_1 = \frac{1.4874 \cdot (q_{vt} + q_{vl}) \cdot t_2}{d^2} - \frac{d}{6} = 2.943\ \mathrm{m}$$

$$L_2 = \frac{2.5465 \cdot (q_{vl}) \cdot t}{d^2} - \frac{d}{6} = 1.677\ \mathrm{m}$$

6. Length control:

$L = L_1 + L_2 = 4.619$ m

7. Characteristic dimensions of the separator:

L= 4.62 m d=1.54 m l_1=2.95 m l_2=1.68 m h_{light}=0.77 m h_{mix}=1.23 m

3 HEAT EXCHANGERS

3.1 PLATE HEAT EXCHANGER

Calculate the characteristics of a plate heat exchanger with the following information:

	Gas oil	Water
Volume flow	5000 kg/h	
Inlet temperature	100°C	25°C
Outlet temperature	70°C	35°C
Dynamic viscosity	$2.4 \cdot 10^{-3}$ Pas	$0.719 \cdot 10^{-3}$ Pas
Thermal conductivity	1.264 W/mK	0.629 W/mK
Specific heat	1.25 kJ/kgK	4.176 kJ/kgK
Density	810 kg/m^3	994 kg/m^3

Characteristics of heat exchanger:
Plate width 0.2 m
Plate length 0.75 mm
Space between plates 3.55 mm
Plate thickness 1.5 mm
Plate surface 0.15 m^2
Plate heat conductivity 55.8 W/mK

$a = 0.2$ m
$b = 0.00355$ m
$l = 0.75$ m

$\rho = 810$ kg/m^3
$q_m = 1.389$ kg/s
$\eta_0 = 2.4 \cdot 10^{-3}$ Pas

Pressure drop determination for gas oil:

- The equivalent hydraulic diameter

$$d_{eh} = \sqrt{\frac{4 \cdot a \cdot b}{\pi}} = 0.03 \text{ m}$$

$$v = \frac{q_m}{\rho \cdot a \cdot b} = 2.415 \text{ m/s}$$

$$Re = \frac{d_{eh} \cdot v \cdot \rho}{\eta_0} = 2.451 \cdot 10^4$$

$$\lambda = \frac{2.5}{Re^{0.3}} = 0.121$$

$$\Delta p = \frac{2 \cdot \rho \cdot v^2 \cdot \lambda \cdot l}{d_{eh}} = 2.842 \cdot 10^4 \text{ Pa}$$

Pressure drop determination for water:
$c_{po} = 1.25 \cdot 10^3$ J/(kg·K)
$c_{pv} = 4.176 \cdot 10^3$ J/(kg·K)
$\rho_v = 994$ kg/m^3
$\eta_v = 0.719 \cdot 10^{-3}$ Pas

$$\Phi_0 = q_m \cdot c_{po} \cdot (100 - 70) = 5.209 \cdot 10^4 W$$

$$q_{mv} = \frac{\Phi_0}{c_{pv} \cdot (35 - 25)} = 1.247 \text{ kg/s}$$

$$Re = \frac{d_{eh} \cdot v_v \cdot \rho_v}{\eta_v} = 7.346 \cdot 10^4$$

$$v_v = \frac{q_{mv}}{\rho_v \cdot a \cdot b} = 1.767 \text{ m/s}$$

$$\lambda = \frac{2.5}{{Re_v}^{0.3}} = 0.087$$

$$\Delta p = \frac{2 \cdot \rho_v \cdot {v_v}^2 \cdot \lambda_v \cdot l}{d_{eh}} = 1.343 \cdot 10^4 \text{ Pa}$$

Pressure drops are large, because only one plate was assumed for the heat exchanger. A scheme of parallel streams is used:

$$\Delta T_m = \frac{(100 - 35) - (70 - 25)}{ln \frac{100 - 35}{70 - 25}} = 54.389 \text{ K}$$

Three plates are used, so that the flow allocation is equal (2 streams of gas/oil and 2 streams of water)

Gas oil:

$d_e = \frac{2 \cdot a \cdot b}{a+b} = 0.007 \text{ m}$ $n_p = 2$ $\lambda_{tpo} = 1.264 \ \frac{\text{W}}{\text{m·K}}$

$Re_0 = \frac{d_e \cdot v \quad \cdot \rho}{n_p \cdot \eta_0} = 2.843 \cdot 10^4$ $Pr = \frac{c_{po} \cdot \eta_0}{\lambda_{tpo}} = 2.373$

$$\alpha_t = 0.2536 \cdot \frac{\lambda_{tpo}}{d_e} \cdot {Re_0}^{0.65} \cdot Pr^{0.4} = 1.141 \cdot 10^4 \frac{\text{kg}}{\text{s}^3 \cdot \text{K}}$$

Cooling water:

$$\lambda_{tpv} = 0.629 \ \frac{\text{W}}{\text{m·K}}$$

$Re_v = \frac{d_e \cdot v_v \cdot \rho_v}{n_p \cdot \eta_v} = 8.523 \cdot 10^3$ $Pr_v = \frac{c_{pv} \cdot \eta_v}{\lambda_{tpv}} = 4.774$

$$\alpha_h = 0.2536 \cdot \frac{\lambda_{tpv}}{d_e} \cdot {Re_v}^{0.65} \cdot {Pr_v}^{0.4} = 1.533 \cdot 10^4 \frac{\text{kg}}{\text{s}^3 \cdot \text{K}}$$

Thermal resistance of cooling water:

$$R_h = 0.688 \cdot 10^{-5} \frac{\text{m}^2 \cdot \text{K}}{\text{W}}$$

Thermal resistance for gas flow:

$R_t = 0.6 \cdot 10^{-5}\,\frac{\mathrm{m^2 \cdot K}}{\mathrm{W}}$ $\qquad \lambda_p = 55.8\,\frac{\mathrm{W}}{\mathrm{m \cdot K}}$ $\qquad d_p = 1.5 \cdot 10^{-3}\ \mathrm{m}$

$$k = \left(\frac{1}{\alpha_t} + \frac{d_p}{\lambda_p} + \frac{1}{\alpha_h} + R_t + R_h\right)^{-1} = 5.192 \cdot 10^3\,\frac{\mathrm{kg}}{\mathrm{s^3 \cdot K}}$$

Required area of the heat exchanger:

F=1

$$A = \frac{\Phi_0}{k \cdot \Delta T_m \cdot F} = 0.184\ \mathrm{m^2}$$

$$A_p = 0.15\ \mathrm{m^2}$$

$$n_p = \frac{A}{A_p} = 1.23$$

$$n_{pr} = 2$$

$$NTU = \frac{k \cdot A \cdot n_{pr}}{q_m \cdot c_{po}} = 1.103$$

F is determined from figure 5.12 for system 1/1

F_{od}=0.98

$$A = \frac{\Phi_0}{k \cdot \Delta T_m \cdot F_{od}} = 0.188\ \mathrm{m^2}$$

The required heat flux is achieved by the two plates of the plate heat exchanger. One of the streams will have 1 pass, and the other will have 2 passes through the heat exchanger. The selection of whether to use 3 or 2 plates depends on the allowable pressure drop.

3.2 PIPE HEAT EXCHANGER

With 40,000 kg/h hot oil for absorption, we heat 41,000 kg/h cold saturated oil, which is fed from the absorber. Temperature of the hot oil at the entrance to the heat exchanger is 175°C and at the outlet 70°C. The temperature of saturated oil at the entrance is 8°C and at the outlet of the exchanger it is 146°C. Allowed pressure drop in the heat exchanger for each pressure flow is 10^5 Pa. The heat exchanger has a tube bundle outer diameter of 19.05 mm. The length of the tube bundle is 8 m. The profile of tubes in the tube bundle is triangular with a step of 25.4 mm. Hot oil for absorption flows through the pipe, and cold saturated oil through the shell.
Cp of oil for absorption is 2.34 kJ/kgK
Cp saturated oil is 2.22 kJ/kgK

$q_{mc} = 40000$ kg/h
$q_{mp} = 41000$ kg/h
$c_{pp} = 2.22 \cdot 10^3$J/(kg·K)
$c_{pp} = 2.34 \cdot 10^3$J/(kg·K)

index c - fluid in the tube
index p - fluid in the shell

$t_{c1} = (175 + 273.15)$ K
$t_{c2} = (70 + 273.15)$ K

$t_{p1} = (38 + 273.15)$ K
$t_{p2} = (146 + 273.15)$ K

Heat flux
$d_z = 19.05$ mm $c = 25.4$mm $d_n = 15.75$ mm

$$\Phi_c = q_{mc} \cdot c_{pc} \cdot (t_{c1} - t_{c2}) = 2.73 \cdot 10^6 W$$

$$\Phi_p = q_{mp} \cdot c_{pp} \cdot \left(\left|t_{p1} - t_{p2}\right|\right) = 2.731 \cdot 10^6 W$$

$$\Delta T_m = \frac{(t_{c1} - t_{p2}) - (t_{c2} - t_{p1})}{ln \frac{t_{c1} - t_{p2}}{t_{c2} - t_{p1}}} = 30.475 \text{ K}$$

$R = \frac{t_{p1} - t_{p2}}{t_{c2} - t_{c1}} = 1.029$ $R = \frac{t_{c2} - t_{c1}}{t_{p1} - t_{c1}} = 0.766$

We choose
$n_{pp} = 4$ $n_{pc} = 8$ $F = 0.86$

$$\Delta t_{lnkor} = F \cdot \Delta T_m = 26.209 \text{ K}$$

Find the physical properties of fluids at medium temperatures:

$t_h = \frac{t_{c1} + t_{c2}}{2} = 122.5\ °\mathrm{C}$	$t_c = \frac{t_{p1} + t_{p2}}{2} = 92\ °\mathrm{C}$
$\mu_c = 0.000875\mathrm{Pas}$	$\mu_p = 0.00134\mathrm{Pas}$
$c_{pc} = 2344\mathrm{J/(kg{\cdot}K)}$	$c_{pp} = 2220\mathrm{J/(kg{\cdot}K)}$
$\lambda_c = 0.128\ \frac{\mathrm{W}}{\mathrm{m{\cdot}K}}$	$\lambda_p = 0.132\ \frac{\mathrm{W}}{\mathrm{m{\cdot}K}}$
$\rho_c = 775\ \mathrm{kg/m^3}$	$\rho_p = 795\ \mathrm{kg/m^3}$

$$k = 250\ \frac{\mathrm{W}}{\mathrm{m^2{\cdot}K}}$$

$$A = \frac{\Phi_c}{k \cdot \Delta T_{lnko}} = 416.653\ \mathrm{m^2}$$

$$L = 8\ \mathrm{m}$$

$$n_c = \frac{A}{\pi \cdot d_z \cdot L} = 870.242$$

$$n_{ci} = 870$$

$$d_{pl} = 990.6\ \mathrm{m}$$

$$A_d = n_{ci} \cdot \pi \cdot d_z \cdot L = 416.537\ \mathrm{m^2}$$

Determining the heat transfer coefficient:
Heat transfer coefficient in the tube (hot oil):

$$A_c = \frac{\pi \cdot d_n^2}{4} = 1.948 \cdot 10^{-4}\ \mathrm{m^2}$$

$$v_c = \frac{q_{mc} \cdot n_{pc}}{A_c \cdot n_{ci} \cdot \rho_c} = 0.677\ \mathrm{m/s}$$

$$Re_c = \frac{d_n \cdot v_c \cdot \rho_c}{\eta_c} = 9.44 \cdot 10^3$$

In the transition region using the image 5.7

$\frac{L}{d_n} = 507.937$ Y=22

$$\alpha_h = \frac{\lambda_c \cdot Y}{d_n \cdot \left(\frac{c_{pc} \cdot \mu_c}{\lambda_c}\right)^{-\frac{1}{3}}} = 450.752$$

Heat transfer coefficient in the shell:
The initial value of the number of partitions is determined by the equation:

$$n_{pr} = \frac{L}{\frac{d_{pl}}{2}} = 16.152$$

A selected number of divisions:
$n_{pri} = 35$

The minimum distance allowed between the partitionsis 50 mm, and we check

$$l_{pr} = \frac{L}{n_{pri}} = 0.229\ \mathrm{m}$$

Free surface of the tubular enters

$$A_s = \frac{d_{pl} \cdot l_{pr} \cdot (c - d_z)}{c} = 0.057\ \mathrm{m}^2$$

Equivalent diameter of the pipe

$$d_e = \frac{3.44 \cdot c^2 - {d_z}^2 \cdot \pi}{d_z \cdot \pi} = 0.018\ \mathrm{m}$$

$$v_p = \frac{q_{mp}}{A_s \cdot \rho_p} = 0.2531\ \mathrm{m/s}$$

$$Re_c = \frac{d_e \cdot v_p \cdot \rho_p}{\mu_p} = 2.708 \cdot 10^3$$

$$\alpha_p = 0.36 \cdot \frac{\lambda_p}{d_e} \cdot {Re_p}^{0.55} \cdot \left(\frac{c_{pp} \cdot \mu_p}{\lambda_p}\right)^{\frac{1}{3}} = 575.023\,\frac{\mathrm{W}}{\mathrm{m^2K}}$$

$$R_0 = 0.00036\,\frac{\mathrm{m^2 \cdot K}}{\mathrm{W}} \qquad R_i = 0.00036\,\frac{\mathrm{m^2 \cdot K}}{\mathrm{W}}$$

$$k_{izr} = \left(\frac{1}{\alpha_h} + R_0 + R_i \frac{d_z}{d_n} + \frac{1}{\alpha_p}\frac{d_z}{d_n}\right)^{-1} = 195.412\,\frac{\mathrm{W}}{\mathrm{m^2K}}$$

Pressure drop
In pipes:
$Re_c = 9.44 \cdot 10^3$

λ_t evaluated according to figure 5.9

$\lambda_t := 0.04$ $\qquad \Phi_t := 1$ $\qquad \xi := 4$

The equations (5.14) and (5.19):

$$\Delta p_{pc} = \frac{\rho_c}{2} \cdot v_c^2 \cdot \left(\frac{\lambda_t \cdot L \cdot n_{pc}}{d_n \cdot \Phi_t} + n_{pc} \cdot \zeta\right) = 3.452 \cdot 10^4\ \mathrm{Pa}$$

Shell:
$Re_p = 2.708 \cdot 10^3$

λ_{tp} evaluated according to figure 5.10 for 25% of free surface

$\lambda_{tp} := 0.45$

Equation (5.20):

$$\Delta p_{pp} = \frac{\rho_p}{2} \cdot v_p^2 \cdot \lambda_{tp} \left(\frac{d_{pl} \cdot \left(n_{pri} + 1\right)}{d_e \cdot \Phi_t}\right) = 2.266 \cdot 10^4\ \mathrm{Pa}$$

Finally, we calculate the thickness of the wall of the shell and the lid, as in the case of pressure vessels.

3.3 CONDENSER

Determine the characteristic dimensions of a condenser for the condensation of 27,000 kg/h of saturated steam of n-propanol at 118°C and a pressure of $2.03\cdot10^5$ Pa. Water at a temperature of 30°C is used for cooling and is heated to 50°C. The condenser should be lying tube heat exchanger. Fluid in the shell should be n-propanol and in the tubes, cooling water. The permitted pressure drop in the shell is $2\cdot10^4$ Pa, and in the tube $5\cdot10^4$ Pa. The pipes have a diameter of 19.05 mm and a length of 2.44 m.

Δh_{izp}=643750 J/kg

$q_{mc} = 27000\ \mathrm{kg/h}$

$c_{pc} = 4184\mathrm{J/(kg\cdot K)}$

$t_p = (118 + 273.15)\ \mathrm{K}$

index c - fluid in pipes   index p - fluid in shell

$t_{c1} = (30 + 273.15)\ \mathrm{K}$

$t_{c2} = (50 + 273.15)\ \mathrm{K}$

$d_z = 19.05\ \mathrm{mm}$ $c = 25.4\mathrm{mm}$ $d_n = 15.75\ \mathrm{mm}$

Heat flux:

$\Phi \quad = q_{mp}\cdot\Delta h_{izp} = 4.828\cdot10^6 W$

$$q_{mc} = \frac{\Phi}{c_{pc}\cdot(t_{c2}-t_{c1})} = 57.697\ \mathrm{kg/s}$$

The counter flow heat exchanger is selected with one pass through the shell:

$n_{pp} := 1$

$F := 1$

$$\Delta T_{ln} = \frac{(t_p - t_{c2}) - (t_p - t_{c1})}{ln\frac{t_p - t_{c2}}{t_p - t_{c1}}} = 77.571\ \mathrm{K}$$

$$\Delta t_{lnkor} = F\cdot\Delta T_m = 77.571\ \mathrm{K}$$

The physical properties of water at average temperature are evaluated:

$t_c = \frac{t_{c1}+t_{c2}}{2} = 313K$
$\mu_c = 0.000722\mathrm{Pas}$
$\lambda_c = 0.615\ \frac{\mathrm{W}}{\mathrm{m\cdot K}}$
$\rho_c = 992\ \mathrm{kg/m^3}$

The length of the exchanger or the number of passes through the tubes are increased to ensure sufficiently high velocity of the cold fluid through tubes

$n_{pc} := 4$

The K is selected and takes into account thermal resistance caused by deposits.
R=0.00054 m^2K/W

$$k = 555 \frac{\mathrm{W}}{\mathrm{m^2 \cdot K}}$$

$$A = \frac{\Phi_c}{k \cdot \Delta T_{lnko}} = 112.147\ \mathrm{m}^2$$

triangular schedule

$$n_{ci} = 774$$

$$L = 2.44\ \mathrm{m}$$

$$d_{pl} = 838.2\ \mathrm{m}$$

$$n_c = \frac{A}{\pi \cdot d_z \cdot L} = 767.985$$

$$A_d = n_{ci} \cdot \pi \cdot d_z \cdot L = 113.025\ \mathrm{m}^2$$

Determination of heat transfer coefficient:
The convectivity coefficient of the tube bundle:

$$A_c = \frac{\pi \cdot d_n^2}{4} = 1.948 \cdot 10^{-4}\ \mathrm{m}^2$$

$$v_c = \frac{q_{mc} \cdot n_{pc}}{A_c \cdot n_{ci} \cdot \rho_c} = 1.543\ \mathrm{m/s}$$

$$Re_c = \frac{d_n \cdot v_c \cdot \rho_c}{\mu_c} = 3.339 \cdot 10^3$$

For turbulent flow, the equation (5.6) should be used:

$$\alpha_c = 0.027 \cdot \frac{\lambda_c}{d_n} \cdot {Re_c}^{0.8} \cdot \left(\frac{c_{pc} \cdot \mu_c}{\lambda_c}\right)^{\frac{1}{3}} = 7451 \frac{\mathrm{W}}{\mathrm{m^2 K}}$$

The heat transfer coefficient on the outer surface of the tubes:
The calculation of the outer wall temperature:

$$\alpha_0 = 970 \frac{\mathrm{W}}{\mathrm{m^2 K}}$$

$$t_w = t_c + \frac{\alpha_0}{\alpha_0 \cdot \frac{d_z}{d_n} + \alpha_0} \cdot (t_p - t_c) = 47.429\ °C$$

The physical properties of n - propanol at average temperature are evaluated:

$t_{hs} = \frac{t_p + t_w}{2} = 82.64\ °C$
$\mu_p = 0.000602\mathrm{Pas}$
$\lambda_p = 0.415 \frac{\mathrm{W}}{\mathrm{m \cdot K}}$
$\rho_p = 746.6\ \mathrm{kg/m^3}$

The heat transfer coefficient of condensing vapours on the outer surface can be determined by equation (5.9):

$q_{mc} = \frac{q_{mp}}{n_{ci}^{\frac{2}{3}}} = 0.08897$ kg/s

$$\alpha_p := \left(\frac{\mu_p^2}{\lambda_p^3 \, \rho_p^2 \cdot g}\right)^{-\frac{1}{3}} \cdot 1.5 \cdot \left(\frac{4 \cdot q_{mk}}{\mu_p \cdot L}\right)^{-\frac{1}{3}} = 862.034 \, \frac{W}{m^2 \cdot K}$$

The heat transfer coefficient:

$R_i = 0.00054 \, \frac{\mathrm{m^2 \cdot K}}{\mathrm{W}}$

$$k_{izr} = \left(\frac{1}{\alpha_p} + R_i \frac{d_z}{d_n} + \frac{1}{\alpha_c} \frac{d_z}{d_n}\right)^{-1} = 506.197 \, \frac{\mathrm{W}}{\mathrm{m^2 K}}$$

Pressure drop:

Pipe bundle:

$Re_c = 3.339 \cdot 10^4$

λ_t Is determined according to figure 5.9

$\lambda_t := 0.0295$ $\qquad \Phi_t := 1$ $\qquad \xi := 4$

Equations (5.14) and (5.19) are used:

$$\Delta p_{pc} = \frac{\rho_c}{2} \cdot v_c^2 \cdot \left(\frac{\lambda_t \cdot L \cdot n_{pc}}{d_n \cdot \Phi_t} + n_{pc} \cdot \zeta\right) = 4.047 \cdot 10^4 \text{ Pa}$$

Shell:

$n_{pr} = \frac{L}{d_{pl}} = 2.911$

$n_{pri} = 3$

$l_{pr} = \frac{L}{n_{pri}} = 0.813$ m

The free surface of the tube bundle:

$$A_s = \frac{d_{pl} \cdot l_{pr} \cdot (c - d_z)}{c} = 0.1704 \, m^2$$

Equivalent diameter for a triangular schedule:

$$d_e = \frac{3.44 \cdot c^2 - d_z^2 \cdot \pi}{d_z \cdot \pi} = 0.01803 \, m$$

Characteristics of n-propanol at 118°C:

$\mu_{pp} = 10^{-5}$ Pas	
$\rho_{pp} = 3.815$ kg/m^3	

$v_{pp} = \frac{q_{mp}}{A_S \cdot \rho_{pp}} = 11.535\ m/s\ 0$

$$Re_p = \frac{d_e \cdot v_{pp} \cdot \rho_{pp}}{\mu_{pp}} = 7.936 \cdot 10^4$$

λ_{tp} is determined from figure 5.10 for 25% of free surface:

$\lambda_{tp} := 0.209$

Equation (5.20) is used:

$$\Delta p_{pp} = \frac{\rho_p p}{2} \cdot v_{pp}^2 \cdot \lambda_{tp} \left(\frac{d_{pl} \cdot (n_{pri} + 1)}{d_e \cdot \Phi_t} \right) = 9.862 \cdot 10^3 \text{ Pa}$$

The calculated heat transfer coefficient is lower than assumed. Therefore, the surface area of the heat exchanger and the number of tubes (the calculation was repeated with a new k) are increased. The coefficient of heat transfer can also be increased by increasing the flow of cold fluid, but in this case the pressure drop is higher.

4 COMPRESSORS

4.1 TWO-STAGE PISTON COMPRESSOR

A mixture of hydrogen and hydrocarbons with an average molecular weight of 2.925 kg/kmol is compressed from $14.5\cdot10^5$ Pa to $130\cdot10^5$ Pa. The temperature of the gas mixture on the suction side of the compressor is 35°C. The mixture flow rate is 5000 m^3/h under normal conditions (0°C, 1 bar). Compressibility ratio is 1.4.

$p_T = 130 \cdot 10^5\,\text{Pa}$
$p_S = 14.5 \cdot 10^5\,\text{Pa}$

Chekin ratio calculation:

$$r_T := \frac{p_T}{p_S} = 8.966$$

The maximum pressure ratio per stage is 5. The calculated pressure ratio is too large; therefore, two stages are necessary:

$$r_K := \sqrt[2]{r_T} = 2.994$$

Checking the temperature rise (temperature on pressure side must not exceed 250°C).
- after the first stage gas is cooled to the inlet temperature:

$T_T = T_S \cdot r_K^{\frac{\kappa-1}{\kappa}} = 421.545\ \text{K}$
$T_T = 148.395°\text{C}$

Approximate calculation of the pressure on the pressure side:
- first stage:

$p_{s1} = p_s$
$p'_{T1} = r_k \cdot p_{s1} = 4341659\ \text{Pa}$

-second stage:

$p_{s2} = p'_{T1}$
$p'_{T2} = r_k \cdot p_{s2} = 13000000\ \text{Pa}$

Pressure drop between stages:

$$\Delta p_{1\_2} = 0.1 \cdot \left(\frac{p'_{T1}}{\text{Pa}}\right)^{0.926} \cdot \text{Pa} = 140110\ \text{Pa}$$

Pressure drop in the compensation vessel (approximately 1% of the pressure):
- first stage:

$\Delta p_{s1} = 0.01 \cdot p_{s1} = 1.45 \cdot 10^4\ \text{Pa}$

-between first and second stage:

$\Delta p'_{1\_2} = 0.01 \cdot p'_{T1} = 4.342 \cdot 10^4\ \text{Pa}$

-second stage:
$\Delta p_{T2} = 0.01 \cdot p'_{T2} = 1.3 \cdot 10^5$ Pa

Pressure on the suction side:
-first stage:
$p_{s1d} = p_{s1} - \Delta p_{s1} = 1.436 \cdot 10^6$ Pa

-second stage
$p_{s2d} = p'_{T1} \quad = 4.342 \cdot 10^6$ Pa

The temperature on the suction side of each stage is 35°C, because the mixture must be cooled.

Pressure on pressure side:
-first stage:
$p_{T1d} = p'_{T1} + \Delta p_{1_2} + \Delta p'_{1_2} = 4.525 \cdot 10^6$Pa

-second stage:
$p_{T2d} = p'_{T2} + \Delta p_{T2} = 1.313 \cdot 10^7$Pa

Actual pressure ratios:

$r_{K1d} = \frac{p_{T1d}}{p_{s1d}} = 3.152$ $\qquad r_{K2d} = \frac{p_{T2d}}{p_{s2d}} = 3.024$

Gas mixture temperature calculation at the discharge side of the adiabatic compression:
$T_{T1} = T_s \cdot r_{K1d}^{\frac{\kappa-1}{\kappa}} = 154.638°\mathrm{C}$ $\qquad T_{T2} = T_s \cdot r_{K2d}^{\frac{\kappa-1}{\kappa}} = 149.596°\mathrm{C}$

Compressor power determination for adiabatic compression:
-first stage:
$q_{v1} = q_{v0} \cdot \frac{p_0}{p_{s1d}} \cdot \frac{T_S}{T_0} = 0.109 \frac{\mathrm{m}^3}{\mathrm{s}}$ $\qquad P_{ad1} = q_{v1} \cdot p_{s1d} \cdot \frac{\kappa}{\kappa-1} \cdot \left(r_{K1d}^{\frac{\kappa-1}{\kappa}} - 1\right) = 212.9$ kW

-second stage:
$q_{v2} = q_{v0} \cdot \frac{p_0}{p_{s2d}} \cdot \frac{T_S}{T_0} = 0.03609 \frac{\mathrm{m}^3}{\mathrm{s}}$ $\qquad P_{ad2} = q_{v2} \cdot p_{s2d} \cdot \frac{\kappa}{\kappa-1} \cdot \left(r_{K2d}^{\frac{\kappa-1}{\kappa}} - 1\right) = 203.9$ kW

Required shaft power:
According to table 6.1, the efficiency is determined to be $\boldsymbol{\eta := 0.81}$
-first stage:
$$P_1 = \frac{P_{ad1}}{\eta} = 262.9\mathrm{kW}$$

-second stage:
$$P_2 = \frac{P_{ad2}}{\eta} = 251.8\mathrm{kW}$$

Total power for both stages:
$P = P_1 + P_2 = 514.6\text{kW}$

Charge rate – useless space:
$\varepsilon_0 := 0.15$

-first stage:

$$\lambda_1 = 1 - \varepsilon_0 \cdot \left(r_{K1d}{}^{\frac{1}{\kappa}} - 1\right) = 0.809$$

-second stage:

$$\lambda_2 = 1 - \varepsilon_0 \cdot \left(r_{K2d}{}^{\frac{1}{\kappa}} - 1\right) = 0.819$$

Required ejection volume of the cylinder:
$V_{g1} = \frac{q_{v1}}{\lambda_1} = 0.1349\,\frac{\text{m}^3}{\text{s}}$ $\qquad V_{g2} = \frac{q_{v2}}{\lambda_2} = 0.04405\,\frac{\text{m}^3}{\text{s}}$

Determination of the piston diameter (inner diameter of the cylinder) – Table 6
Opposite positioned cylinders are taken:

L_n=279 mm $\quad n$ = 400 min^{-1} $\; d_\text{v}$ = 125 mm

Piston diameter:
-first stage:

$$d_1 = \sqrt{\frac{2 \cdot V_{g1}}{L_n \cdot n \cdot \pi} + \frac{d_v^2}{2}} = 0.232\text{ m}$$

-second stage:

$$d_2 = \sqrt{\frac{2 \cdot V_{g2}}{L_n \cdot n \cdot \pi} + \frac{d_v^2}{2}} = 0.151\text{ m}$$

Average piston velocity:
$v = 2 \cdot L_n \cdot n = 3.72\ \frac{\text{m}}{\text{s}}$

The average piston velocity should not exceed 4 m/s (in some cases 4.3 m/s) for pistons with lubrication, and 3.5 m/s for pistons without lubrication.

4.2 CENTRIFUGAL COMPRESSOR

Determine the basic parameters of a centrifugal compressor. Data:

- flow rate of the hydrocarbon mixture at a set temperature and pressure at the suction side is 5500 m^3/h
- the temperature at the suction side is 35°C
- the pressure on the suction side is 2 MPa
- the gas density on the suction side is 4.279 kg/m^3
- pressure on the pressure side 3 is MPa
- the average molecular weight is 5.524 kg/kmol
- the average specific heat coefficient is 31.98 J/mol K

T_s =(35+273.15)K
q_v = 5500 m^3/h
M= 5.524 kg/kmol
ρ_s = 4.279 kg/m^3
p_s = 2 MPa
p_T = 3 MPa
c_p = 31.98 J/(molK)

Exponent determination for adiabatic compression process:

$$\kappa = \frac{c_p}{c_p - R} = 1.351$$

Calculation of sound speed propagation in the gas:

$$v_z = \sqrt{\kappa \cdot \frac{p_s}{\rho_s}} = 794.74 \text{ m/s}$$

Pressure ratio:

$$r_k = \frac{p_T}{p_S} = 1.5$$

The operating rotor diameter for gas flow is evaluated according to table 6.3:

$\mu' := 0.495$

Polytrophic operation rate (Figure 6.2):

$\mu_{pol} := 0.733$

Adiabatic operation rate:

$$\mu_{ad} = \frac{r_K^{\frac{\kappa-1}{\kappa}} - 1}{r_K^{\frac{\kappa-1}{\kappa \cdot \mu_{pol}}} - 1} = 0.7188$$

The rotor diameter is determined (Table 6.3):

d_r = 450 mm

Gas temperature on pressure side:

$$T_T = T_s \cdot r_K^{\frac{\kappa-1}{\kappa \cdot \mu_{pol}}} = 355.8\ \text{K}$$

Polytrophic exponent:

$$n = \frac{1}{1 - \frac{\kappa - 1}{\kappa \cdot \mu_{pol}}} = 1.5496$$

Polytrophic work of compressor:

$$W_{pol} = \frac{n}{n-1} \cdot \frac{R \cdot T_s}{M} \cdot \left(r_K^{\frac{n-1}{n}} - 1 \right) = 202.3 \cdot 10^3 \text{J/kg}$$

Adiabatic work of compressor:

$$W_{ad} = \frac{\kappa}{\kappa - 1} \cdot \frac{R \cdot T_s}{M} \cdot \left(r_K^{\frac{\kappa-1}{\kappa}} - 1 \right) = 198.3 \cdot 10^3 \text{J/kg}$$

Required power for gas compression calculated from polytrophic work:

$$P = \frac{q_v \cdot \rho_s \cdot W_{pol}}{\mu_{pol}} = 1804\ \text{kW}$$

The number of rotors (figure 6.3):
It is chosen:
$n_r = 6$

Rotor speed:

$$v_R = \sqrt{\frac{W_{pol}}{n_r \cdot \mu'}} = 260.96\ \text{m/s}$$

Rotor angular speed:

$$n_0 = \frac{v_R}{\pi \cdot d_r} = 185\ \text{s}^{-1}$$

Typical values:

number of rotors	$n_r = 6$
rotor diameter	$d_r = 0.45$ m
angular speed	$n_0 = 185$ s^{-1}
shaft power required	$P = 1804$ kW

5 MIXING

5.1 STATIC MIXTURE DIMENSIONING FOR LAMINAR FLUID FLOW

We wish to achieve a coefficient of variation of $\sigma / \bar{x} = 0{,}05$ when mixing two viscous fluids ($\eta_1 = \eta_2 = 1.2\ \mathrm{Pa\ s}$) with volume flow of $q_{v_1} = 1.94 \cdot 10^{-4}\ \mathrm{m^3/s}$ and $q_{v_2} = 8.86 \cdot 10^{-4}\ \mathrm{m^3/s}$. The density of the mixed flow is $1100\ \mathrm{kg/m^3}$. A nominal diameter pipe DN 50 ($d = 0.0526\ \mathrm{m}$) is used for a static mixer.

The exact static mixture type is selected:

- The entire flow velocity is determined by equation (10.2):

$$v = \frac{4 \cdot q_v}{\pi \cdot d^2} = \frac{4 \cdot 1.08 \cdot 10^{-3}\mathrm{m^3/s}}{\pi \cdot (0.0526\ \mathrm{m})^2} = 0.5\ \mathrm{m/s}$$

- the Reynolds number is evaluated by equation (10.1):

$$\mathrm{Re} = \frac{\rho \cdot v \cdot d}{\eta} = \frac{1100\ \mathrm{kg/m^3} \cdot\ 0.5\ \mathrm{m/s} \cdot 0.0526\ \mathrm{m}}{1.2\ \mathrm{Pas}} = 24$$

The flow is determined as laminar according to the Reynolds number. The Sulzer SMX or Kenics model with 12 mixing elements (table 10.2) could also be used.

Dimensioning of Sulzer SMX static mixture:

- the average volumetric concentration of the mixture for a fluid with a reduced volumetric flow rate is determined according to equation (10.6):

$$\bar{x} = \frac{q_{v_1}}{q_{v_1} + q_{v_2}} = \frac{1.94 \cdot 10^{-4}\mathrm{m^3/s}}{1.94 \cdot 10^{-4}\mathrm{m^3/s} + 8.86 \cdot 10^{-4}\mathrm{m^3/s}} = 0.18$$

- according to the diagram (figure 10.14), the number of SMX mixing elements is determined to achieve a variation coefficient $\sigma / \bar{x} = 0.05$ at $\bar{x} = 0.18$, and the length of the static mixer is evaluated (equation 10.8):

$$n_E \approx 9$$

$$l = n_E \cdot l_E = 9 \cdot 0.0526\ \mathrm{m} = 0.47\ \mathrm{m}$$

- the pressure drop is evaluated according to equation (10.18):

$Ne \cdot Re \approx 1200$ (table 10.7)

$$\Delta p_{SM} = \frac{4}{\pi} \cdot Ne \cdot \mathrm{Re}\ \cdot \frac{\eta \cdot q_v}{d^3} \cdot \frac{l}{d} =$$

$$= \frac{4}{\pi} \cdot 1200 \cdot \frac{1.2\ \mathrm{Pas} \cdot 1.08 \cdot 10^{-3}\mathrm{m^3/s}}{(0.0526\ \mathrm{m})^3} \cdot \frac{0.47\ \mathrm{m}}{0.0526\ \mathrm{m}} = 121{,}577\ \mathrm{Pa}$$

- the theoretical power is determined according to equation (10.22):

$$P_t = \Delta p_{SM} \cdot q_v = 121577\ \mathrm{Pa} \cdot 1.08 \cdot 10^{-3}\mathrm{m^3/s} = 131\ \mathrm{W}$$

Dimensioning of Kenics static mixture:
If the Reynolds number is in the range 10 < Re < 100, 12 mixing elements are used (table 10.2).

- the length of the static mixer is determined according to equation (10.10):
  $$l = n_E \cdot l_E = 12 \cdot 1.5 \cdot 0.0526\ \text{m} = 0.95\ \text{m}$$

- the pressure drop for an empty pipe is determined with equations (10.11) and (10.12):
  $$\Delta p = \lambda \cdot \frac{\rho \cdot v^2}{2} \cdot \frac{l}{d} = \frac{64}{24} \cdot \frac{1100\ \text{kg/m}^3 \cdot (0.5\ \text{m/s})^2}{2} \cdot \frac{0.95\ \text{m}}{0.0526\ \text{m}} = 6622\ \text{Pa}$$

- the pressure drop for the static mixer is determined with equation (10.20):
  $K'_{OL} = 0.068$ (table 10.8)
  $K_{OL} = 5.70$ (table 10.8)
  $A = 19$ (figure 10.16)
  $$\Delta p_{SM} = K \cdot \Delta p = (K'_{OL} \cdot A + K_{OL}) \cdot \Delta p = (0.068 \cdot 19 + 5.70) \cdot 6622\ \text{Pa} = 46{,}301\ \text{Pa}$$

- the theoretical power is determined according to equation (10.22):
  $$P_t = \Delta p_{SM} \cdot q_v = 46301\ \text{Pa} \cdot 1.08 \cdot 10^{-3}\text{m}^3/\text{s} = 50\ \text{W}$$

5.2 STATIC MIXTURE DIMENSIONING FOR TURBULENT FLUID FLOW

It is necessary to carry out the mixing of two water streams $q_{v_1} = 3.65 \cdot 10^{-3}\,\mathrm{m^3/s}$ and $q_{v_2} = 6.21 \cdot 10^{-3}\,\mathrm{m^3/s}$with the aim to equalize the pH value. A DN 100 static mixer ($d = 0.1024\ \mathrm{m}$) is used for this purpose

$$\rho_1 = \rho_2 = 1000\ \mathrm{kg/m^3}$$

$$\eta_1 = \eta_2 = 0.001\ \mathrm{Pas}$$

The exact static mixture type is selected:

- The entire flow velocity is determined - equation (10.2):

$$v = \frac{4 \cdot q_v}{\pi \cdot d^2} = \frac{4 \cdot 9.86 \cdot 10^{-3}\,\mathrm{m^3/s}}{\pi \cdot (0.1024\ \mathrm{m})^2} = 1.2\ \mathrm{m/s}$$

- the Reynolds number is evaluated - equation (10.1):

$$\mathrm{Re} = \frac{\rho \cdot v \cdot d}{\eta} = \frac{1000\ \mathrm{kg/m^3} \cdot 1.2\ \mathrm{m/s} \cdot 0.1024\ \mathrm{m}}{0.001\ \mathrm{Pas}} = 122{,}880$$

The flow is determined to be turbulent according to the Reynolds number. The Sulzer SMV or Kenics model with 1 mixing elements (table 10.2) could be used.

Dimensioning of Sulzer SMV static mixture:

- the average volumetric concentration of the mixture for the fluid with a reduced volumetric flow rate is determined, according to the equation (10.6):

$$\bar{x} = \frac{q_{v_1}}{q_{v_1} + q_{v_2}} = \frac{3.65 \cdot 10^{-3}\,\mathrm{m^3/s}}{3.65 \cdot 10^{-3}\,\mathrm{m^3/s} + 6.21 \cdot 10^{-3}\,\mathrm{m^3/s}} = 0.37$$

According to the diagram (figure 10.15), the two SMV mixing elements are determined (n_e = 2) to achieve the variation coefficient $\sigma / \bar{x} = 0.05$ at $\bar{x} = 0.37$

- the length of the static mixer is evaluated (equation 10.9):

$$l = n_E \cdot l_E = 2 \cdot 0.1024\ \mathrm{m} = 0.20\ \mathrm{m}$$

- the pressure drop is evaluated according to equation (10.19):

$Ne \approx 2$ (table 10.7)

$$\Delta p_{SM} = \frac{16}{\pi^2} \cdot Ne \cdot \frac{\rho \cdot q_v^2}{d^4} \cdot \frac{l}{d} =$$

$$= \frac{16}{\pi^2} \cdot 2 \cdot \frac{1000\ \mathrm{kg/m^3} \cdot (9.86 \cdot 10^{-3}\,\mathrm{m^3/s})^2}{(0.1024\ \mathrm{m})^4} \cdot \frac{0.20\ \mathrm{m}}{0.1024\ \mathrm{m}} = 5599\ \mathrm{Pa}$$

- the theoretical power is determined according to equation (10.22):

$$P_t = \Delta p_{SM} \cdot q_v = 5599 \text{ Pa} \cdot 9.86 \cdot 10^{-3} \text{m}^3/\text{s} = 55 \text{ W}$$

Dimensioning of Kenics static mixture:
If the Reynolds number is *Re*> 5000, 2 mixing elements are used (table 10.2).

- the length of the static mixer is determined according to equation (10.10):

$$l = n_E \cdot l_E = 2 \cdot 1.5 \cdot 0.1024 \text{ m} = 0.31 \text{ m}$$

- the pressure drop for the empty pipe is determined with equation (10.11)

$\lambda = 0.017$ *(Moody diagram)*

$$\Delta p = \lambda \cdot \frac{\rho \cdot v^2}{2} \cdot \frac{l}{d} = 0.017 \cdot \frac{1000 \text{ kg/m}^3 \cdot (1.2 \text{ m/s})^2}{2} \cdot \frac{0.31 \text{ m}}{0.1024 \text{ m}} = 37 \text{ Pa}$$

- the pressure drop for the static mixer is determined with equation (10.20):

$K_{OT} = 26.9$ (table 10.8)

$B = 2.2$ (figure 10.17)

$$\Delta p_{SM} = K \cdot \Delta p = K_{OT} \cdot B \cdot \Delta p = 26.9 \cdot 2.2 \cdot 37 \text{ Pa} = 2190 \text{ Pa}$$

- the theoretical power is determined according to equation (10.22):

$$P_t = \Delta p_{SM} \cdot q_v = 2190 \text{ Pa} \cdot 9.86 \cdot 10^{-3} \text{m}^3/\text{s} = 22 \text{ W}$$

6 COMPRESSION HEAT PUMPS

6.1 EXAMPLE 1

Define the cooling capacity for 1 kg of refrigerant R12, if the temperature of refrigerant vaporisation is -20°C and the condensation temperature is 30°C.

T_{izp} = -20°C
T_{kon} = 30°C

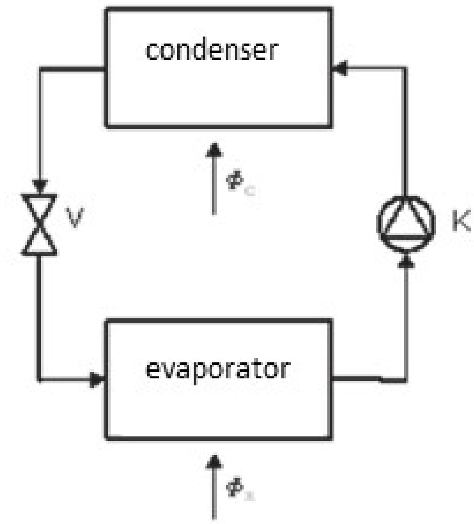


Figure 6.1: Compression heat pump

- in 1 the saturated gas is at -20°C. The enthalpy is:
  h_1 = 343.48 kJ/kg
  v_1 = 0.10934 m^3/kg specific volume

- there is no change in enthalpy through the expansion valve
  $h_1 = h_3$

- in 3 the saturated liquid is at 30°C. The enthalpy is
  h_3 = 228.62 kJ/kg

Cooling capacity

$$q_0 = h_1 - h_2$$
$$q_0 = 114.86 \text{ kJ/kg}$$

Theoretical cooling capacity:

$$q_{v1} = \frac{q_0}{v_1}$$
$$q_{v1} = 1050 \text{ kJ/m}^3$$

6.2 EXAMPLE 2

Define the COP for heating the heat pump, if it uses refrigerant R12. The vaporisation temperature is 0°C and the condensation temperature is 50°C. The enthalpy at point 2 is 376 kJ/kg.

T_u = 0°C
T_{kon} = 50°C
h_2 = 376 kJ/kg

- in 1 the saturated gas is at 0°C. The enthalpy is:
  h_1 = 352.54 kJ/kg

- in 3 the saturated liquid is at 50°C. The enthalpy is:
  h_3 = 248.96 kJ/kg

The heating capacity for 1 kg of refrigerant is defined by the energy balance through the condenser:

$$q_{0c} = h_2 - h_3$$

Compressor theoretical work per kilogram of refrigerant:

$$W = h_2 - h_1$$

Heating number:

$$\varepsilon_c = \frac{\Phi_c}{P} = \frac{q_{oc}}{W}$$

$$\varepsilon_c = 5.4$$

6.3 EXAMPLE 3

The working fluid evaporates in the heat pump at 0°C and condenses at 50°C. The heat is taken from a lake with the water temperature of 10°C. The water is 3°C cooler when it returns to the lake. The water flow is 0.5 kg/s. A heat pump consists of an internal heat exchanger between condensate and steam. The condensate is cooled by 10°C. In the condenser, the refrigerant vapour is not subcooled (only condensed). The isentropic thermodynamic efficiency of the compressor is η = 80%. R12 refrigerant (diflorodiklorometan) is used. Heat pump (heat flux capacitor) h_2` = 389 kJ / kg

T_{upar} = 0°C	ΔT_{vode} = 3°C	η = 80 %
T_{kond} = 50°C	$q_{m,v}$ = 0.5 kg/s	$t_I = 7°C$
T_{vode} = 10°C	ΔT_{kond} = 10°C	$t_{II} = 10°C$

- in 1 the saturated gas is at 0°C. The enthalpy is
  h_1 = 352.54 kJ/kg

- in 3 the saturated liquid is at 50°C. The enthalpy is:
  h_3 = 248.96 kJ/kg

- subcooled liquid
  c_p = 1.095 kJ/kgK

  $h_3^{\prime} = h_3 - c_p \cdot \Delta t$ = 238.01 kJ/kg $= h_4$

Mass flow of refrigerant can be calculated from the energy balance of the evaporator:
$\Delta h_u = h_1 - h_4$
$q_{m,F} \cdot \Delta h_u = q_{m,water} \cdot c_{p,water} \cdot (t_I - t_{II})$
$q_{m.F} = 0.05501$ kg/s

$h_1^{`}$ is determined from energy balance:
$q_{m.F} \cdot (h_3 - h'_3) = q_{m,F} \cdot (h_1 - h'_1)$
$h'_1 = 363.49$ kJ/kg

$$\eta_k = \frac{h_2^{`} - h_1^{`}}{h_2 - h_1^{`}}$$
$h_2 = 395.4$ kJ/kg

$\Phi_c = q_{m,F} \cdot (h_2 - h_3)$
$\Phi_c = 8.056$ kW

7 VENTILATION

7.1 EXAMPLE 1

The volume flow through the channel on Figure is 600 m³/h. The inlet pressure is 200 Pa. Determine the static and dynamic pressure.

ρ_{air} =1.2 kg/m³

μ = 1.85 10^{-5} Pas

λ= 0.1206 $Re^{-0,1505}$

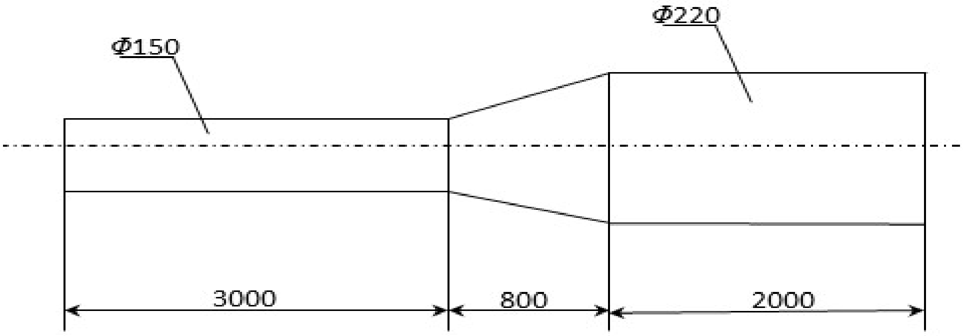


Figure 7.1: Example 1

No	q_v m/s	A m²	Chanel cm×cm	v m/s	p_{dy} Pa	p_{st} Pa	p_s ∑ζ Pa	l m	R Pa/m	$R{\cdot}l$ Pa	Z Pa	Δp Pa

7.2 EXAMPLE 2

Determine the static and dynamic pressure for the following two tubular elements. The static pressure at point 1 is 200 Pa, q_1=400 m^3/h, q_3=150 m^3/h, ρ_{air}= 1.2 kg/m^3

a)

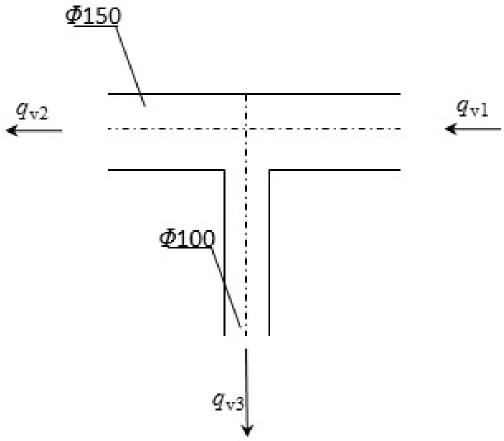


Figure 7.2: Example 2a

No	q_v m/s	A m²	Chanel cm×cm	v m/s	p_{dy} Pa	p_{st} Pa	p_{st} $\Sigma\zeta$ Pa	l m	R Pa/m	$R\cdot l$ Pa	Z Pa	Δp Pa

b)

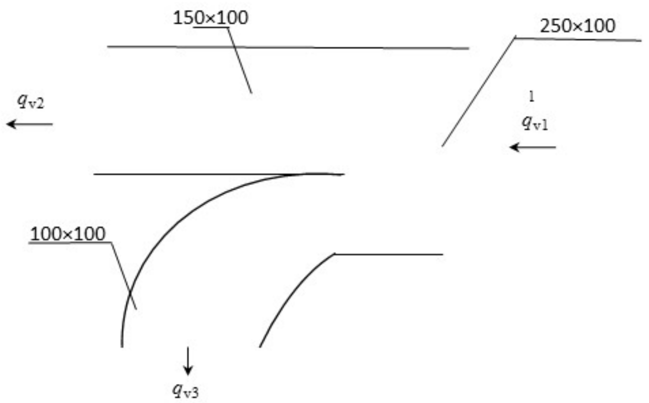


Figure 7.3: Example 2b

No	q_v m/s	A m²	Chanel cm×cm	v m/s	p_{dy} Pa	p_{st} Pa	$\sum\zeta$ Pa	l m	R Pa/m	$R \cdot l$ Pa	Z Pa	Δp Pa

7.3 EXAMPLE 3

The static pressures are given for the pipe section in Figure:
p_1 = 108 Pa, p_2 = 99 Pa, p_3= 80 Pa
q_1=1000 m^3/h, q_3=400 m^3/h
ρ_{air}= 1.2 kg/m^3
Determine the local resistance coefficients.

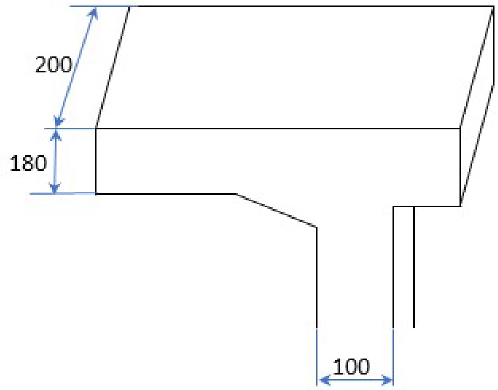


Figure 7.2: Example 3

No		q_v m/s	A m^2	Chanel cm×cm	v m/s	p_{dy} Pa	$\Sigma\zeta$ Pa	p_{sit} l Pam		R Pa/m	$R{\cdot}l$ Pa	Z Pa	Δp Pa