Calculate azimuth of the Sun on June 30 at 12:00 as seen from the AlbaNova building (coordinates: 59.3536 N, 18.0578 E)

Assume that the year has 365 days.

The answer should be in angle degree.

az = 183.437 deg

```
% GIVEN
%
June_day = 21; % changing from 1 to 30 step 1
LT = 12; % local time
LAMBDA = 18.0578; % local longitude
FI = 59.3536;     % local latitude
%
% SOLUTION
% We find the day number Nd
Nd = 3*31+28+30+June_day;
arg1_deg = 360*Nd/365+9.5;
arg2_deg = 2*360*Nd/365+5.4;
arg3_deg = 3*360*Nd/365+105.2;
arg1_rad = arg1_deg*pi/180; % in radians
arg2_rad = arg2_deg*pi/180; % in radians
arg3_rad = arg3_deg*pi/180; % in radians
% Declination of the Sun
d=0.3948-23.2559*cos(arg1_rad)-0.3915*cos(arg2_rad);
d=d-0.1764*cos(arg3_rad);
% Equation of time
arg1_deg = 360*Nd/365+85.9;
arg2_deg = 2*360*Nd/365+108.9;
arg3_deg = 3*360*Nd/365+105.2;
arg1_rad = arg1_deg*pi/180; % in radians
arg2_rad = arg2_deg*pi/180; % in radians
arg3_rad = arg3_deg*pi/180; % in radians
ET=0.0066+7.3525*cos(arg1_rad)+9.9359*cos(arg2_rad);
ET=ET+0.3387*cos(arg3_rad); % in minutes
% Solar time
ST = LT - (4*(15-LAMBDA) + ET)/60; % note units! in h
% Hour angle
w = (12 - ST)*15; % in degrees
% Altitude of the Sun
FI_rad = FI*pi/180;
w_rad = w*pi/180; % in radians
d_rad = d*pi/180; % in radians
arg_psi = cos(w_rad)*cos(FI_rad)*cos(d_rad)+sin(FI_rad)*sin(d_rad);
psi_rad = asin(arg_psi);
psi_deg = psi_rad*180/pi;
% Azimuth
arg_rad = sin(psi_rad)*sin(FI_rad)-sin(d_rad);
arg_rad = arg_rad/cos(psi_rad)/cos(FI_rad);
if ST > 12.00
  az = 180 + acos(arg_rad)*180/pi;
else
  az = 180 - acos(arg_rad)*180/pi;
end
answer = az;
```

Calculate solar time of the Sun on June 3 at 12:00 as seen from the AlbaNova building (coordinates: 59.3536 N, 18.0578 E)

Assume that the year has 365 days.

S_T = 12.2337 h

```
% GIVEN
%
June_day = 21; % changing from 1 to 30 step 1
LT = 12; % local time
LAMBDA = 18.0578; % local longitude
FI = 59.3536;     % local latitude
%
% SOLUTION
% We find the day number Nd
Nd = 3*31+28+30+June_day;
% Equation of time
arg1_deg = 360*Nd/365+85.9;
arg2_deg = 2*360*Nd/365+108.9;
arg3_deg = 3*360*Nd/365+105.2;
arg1_rad = arg1_deg*pi/180; % in radians
arg2_rad = arg2_deg*pi/180; % in radians
arg3_rad = arg3_deg*pi/180; % in radians
ET=0.0066+7.3525*cos(arg1_rad)+9.9359*cos(arg2_rad);
ET=ET+0.3387*cos(arg3_rad); % in minutes
% Solar time
ST = LT - (4*(15-LAMBDA) + ET)/60; % note units! in h
answer = ST;
```

Calculate hour angle of the Sun on June 29 at 12:00 as seen from the AlbaNova building (coordinates: 59.3536 N, 18.0578 E)

Assume that the year has 365 days.

w = -2.261 deg

```
% GIVEN
%
June_day = 21; % changing from 1 to 30 step 1
LT = 12; % local time
LAMBDA = 18.0578; % local longitude
FI = 59.3536;     % local latitude
%
% SOLUTION
% We find the day number Nd
Nd = 3*31+28+30+June_day;
% Equation of time
arg1_deg = 360*Nd/365+85.9;
arg2_deg = 2*360*Nd/365+108.9;
arg3_deg = 3*360*Nd/365+105.2;
arg1_rad = arg1_deg*pi/180; % in radians
arg2_rad = arg2_deg*pi/180; % in radians
arg3_rad = arg3_deg*pi/180; % in radians
ET=0.0066+7.3525*cos(arg1_rad)+9.9359*cos(arg2_rad);
ET=ET+0.3387*cos(arg3_rad); % in minutes
% Solar time
ST = LT - (4*(15-LAMBDA) + ET)/60; % note units! in h
% Hour angle
w = (12 - ST)*15; % in degrees
answer = w;
```

Calculate altitude of the Sun on June 3 at 12:00 as seen from the AlbaNova building (coordinates: 59.3536 N, 18.0578 E)

Assume that the year has 365 days.

The answer should be in angle degree.

ψ_{deg} = 52.91 deg

```
:%
:% GIVEN
:%
June_day = 21; :% changing from 1 to 30 step 1
LT = 12; :% local time
LAMBDA = 18.0578; :% local longitude
FI = 59.3536;     :% local latitude
:%
:% SOLUTION
:% We find the day number Nd
Nd = 3*31+28+30+June_day;
arg1_deg = 360*Nd/365+9.5;
arg2_deg = 2*360*Nd/365+5.4;
arg3_deg = 3*360*Nd/365+105.2;
arg1_rad = arg1_deg*pi/180; :% in radians
arg2_rad = arg2_deg*pi/180; :% in radians
arg3_rad = arg3_deg*pi/180; :% in radians
:% Declination of the Sun
d=0.3948-23.2559*cos(arg1_rad)-0.3915*cos(arg2_rad);
d=d-0.1764*cos(arg3_rad);
:% Equation of time
arg1_deg = 360*Nd/365+85.9;
arg2_deg = 2*360*Nd/365+108.9;
arg3_deg = 3*360*Nd/365+105.2;
arg1_rad = arg1_deg*pi/180; :% in radians
arg2_rad = arg2_deg*pi/180; :% in radians
arg3_rad = arg3_deg*pi/180; :% in radians
ET=0.0066+7.3525*cos(arg1_rad)+9.9359*cos(arg2_rad);
ET=ET+0.3387*cos(arg3_rad); :% in minutes
:% Solar time
ST = LT - (4*(15-LAMBDA) + ET)/60; :% note units! in h
:% Hour angle
w = (12 - ST)*15; :% in degrees
:% Altitude of the Sun
FI_rad = FI*pi/180;
w_rad = w*pi/180; :% in radians
d_rad = d*pi/180; :% in radians
arg_psi = cos(w_rad)*cos(FI_rad)*cos(d_rad)+sin(FI_rad)*sin(d_rad);
psi_rad = asin(arg_psi);
psi_deg = psi_rad*180/pi;
answer = psi_deg;
```

Calculate equation of time of the Sun on June 12.

Give answer in minutes.

Assume that the year has 365 days.

E_T = 0.193 min

```
% GIVEN
%
June_day = 21; % changing from 1 to 30 step 1
%
```

```
% SOLUTION
% We find the day number Nd
Nd = 3*31+28+30+June_day;
% Equation of time
arg1_deg = 360*Nd/365+85.9;
arg2_deg = 2*360*Nd/365+108.9;
arg3_deg = 3*360*Nd/365+105.2;
arg1_rad = arg1_deg*pi/180; % in radians
arg2_rad = arg2_deg*pi/180; % in radians
arg3_rad = arg3_deg*pi/180; % in radians
ET=0.0066+7.3525*cos(arg1_rad)+9.9359*cos(arg2_rad);
ET=ET+0.3387*cos(arg3_rad); % in minutes
answer = ET;
```

Calculate declination of the Sun on June 19.
Give answer in angle degrees.
Assume that the year has 365 days.

d = 23.3182 deg

```
% GIVEN
%
June_day = 21; % changing from 1 to 30 step 1
%
% SOLUTION
% We find the day number Nd
Nd = 3*31+28+30+June_day;
arg1_deg = 360*Nd/365+9.5;
arg2_deg = 2*360*Nd/365+5.4;
arg3_deg = 3*360*Nd/365+105.2;
arg1_rad = arg1_deg*pi/180; % in radians
arg2_rad = arg2_deg*pi/180; % in radians
arg3_rad = arg3_deg*pi/180; % in radians
% Declination of the Sun
d=0.3948-23.2559*cos(arg1_rad)-0.3915*cos(arg2_rad);
d=d-0.1764*cos(arg3_rad);
answer = d;
```

Calculate the Sun's surface temperature (K) knowing that due to fusion, the Sun is losing mass 4.24E9 (kg) during one second.
The Sun's radius is $R=6.955*10^{8}$ m.
Assume a black-body radiation from the Sun's surface.
Stefan-Boltzmann constant is sig = $5.67051*10^{-8}$ $W/m^{2}/K^{4}$
and the speed of light is c = 299792458 m/s.

T = 5666.27 K

```
% GIVEN
%
R=6.955e8;       % not changing
sig=5.67051e-8;  % not changing
c=299792458;     % not changing
dM=4.3e9;        % changing from 4.1e9 to 4.3e9 step 0.02e9
%
% SOLUTION
% Total energy emitted by the Sun per second
q = dM*c**2;
% Heat flux from the Sun's surface
A = 4*pi*R**2;
```

```
q2p = q/A;
% The Sun's surface temperature
T=(q2p/sig)**0.25;
answer = T;
```

Calculate the mean heat flux at Earth orbit resulting from solar irradiation on June 21st.
Assume the Sun constant Gcs = 1366 W/m^2 and that year has 365 days.

I_e = 1326.90633 W/m^2

```
% GIVEN
%
June_day = 21; % changing from 1 to 30 step 1
Gsc = 1366;    % not changing
%
% SOLUTION
% We find variation of Sun irradiation as a
% function of day number Nd
Nd = 3*31+28+30+June_day;
arg_deg = 360*(Nd-3)/365;
arg_rad = arg_deg*pi/180; % in radians
Ie = Gsc*(1+0.033412*cos(arg_rad));
answer = Ie;
```

Electricity generating plant is typically built within 5 years with uniformly distributed expenditures per each year. A new construction schedule is considered in which the plant will be built within 4 years with uniformly distributed expenditures. The sum of expenditures in both cases will be the same.

Calculate the change of capital cost in percent using the original schedule as a reference. Use the following pattern for the answer: answer = (Co-Cn)/Co * 100 (%),

where:
Co - original schedule cost (5-year build)
Cn - new schedule cost (4-tear build).
The interest rate in both cases is the same
and equal to 4%.

Use the end of construction as the reference time point

Answer = 1.99 %

```
% GIVEN
%
i = 0.05; % changing from 0.03 to 0.1, step 0.01
%
% SOLUTION
% We assume annual expenditure in original
% schedule:
Ex_o = 1;
% Then expenditures in new schedule will be:
Ex_n = 5*Ex_o/4;
% Capital cost for original schedule is now:
Co = Ex_o*(1+(1+i)+(1+i)**2+(1+i)**3+(1+i)**4);
% and for the new one
Cn = Ex_n*(1+(1+i)+(1+i)**2+(1+i)**3);
%
answer = (Co-Cn)/Co * 100;
```

Electricity generating plant was initially scheduled to operate N_o = 18 years, but this lifetime was extended to N_n = 30 years.

Calculate the new levelized charge for electricity to cover capital expenditures if according to the initial schedule it would be equal to L_{cap}_o = 38 Euro/MWh

In both cases the invested amount and the number of electricity produced per year are the same and the interest rate is equal to i = 0.05 .

L_{cap}_n = 28.8961 Euro/MWh

```
% GIVEN
%
i = 0.05; % changing from 0.03 to 0.1, step 0.01
Lcap_o = 25; % Changing from 20 to 40, step 2
No=20;    % changing from 10 to 20, step 1
Nn=30;    % changing from 30 to 40, step 1
%
% SOLUTION
% We find new levelized charge for electricity
% to cover capital expeditures directly
% from the relation between the two:
%
i1=1+i;
answer = Lcap_o*i1**(Nn-No) * (i1**No-1)/(i1**Nn-1);
```

For a given finite thermodynamic process from state (1) to state (2), the maximum work is:

- ☐ equal to the internal energy change: $E_{I1} – E_{I2}$
- ☑ Is less than the internal energy change $E_{I1} – E_{I2}$
- ☐ Is greater than the internal energy change $E_{I1} – E_{I2}$
- ☐ none of the above

Control mass system is defined as such a system that:

- ☐ Is opened for mass flow through its boundaries
- ☑ No mass can flow through its boundaries
- ☐ Mass flow through boundaries is controlled
- ☐ none of the above

For an isobaric process from state 1 to 2, according to the Gay-Lussac's law, the volume ratio V_1/V_2 is:

- ☐ constant
- ☑ equal to temperature ratio T_1/T_2
- ☐ equal to temperature ratio T_2/T_1
- ☐ none of the above

The energy efficiency of an ideal Rankine cycle is approximately equal to the ratio of:

- [x] The theoretical turbine power to the added thermal power
- [ ] The turbine internal power to the added thermal power
- [ ] The turbine effective power to the turbine internal power
- [ ] none of the above

The following property is an example of the intensive property:

- [ ] Entropy S [J K^{-1}]
- [x] Temperature T [K]
- [x] Specific entropy s [J kg^{-1} K^{-1}]
- [ ] none of the above

Energy that is available to be used as useful work is called:

- [x] exergy
- [x] Available energy
- [ ] Available work
- [ ] none of the above

A phase transition from solid to gas is called:

- [ ] deposition
- [ ] evaporation
- [x] sublimation
- [ ] none of the above

Energy efficiency of the real Rankine cycle, when its ideal energy efficiency is equal to 0.5 and the turbine internal efficiency is equal to 0.8, is approximately equal to:

- [ ] 0.3
- [x] 0.4
- [ ] 0.35
- [ ] none of the above

The importance of the second law of the thermodynamics stems from the fact that it helps:

- [ ] To search for systems with high thermodynamic efficiency
- [ ] To determine the flow and temperature fields in systems
- [x] To search for entropy generation minimization in systems
- [ ] none of the above

The isentropic (or internal) turbine efficiency is defined as a ratio of:

- [ ] The internal to effective turbine power
- [ ] The effective to internal turbine power
- [x] The internal to theoretical turbine power
- [ ] none of the above

Energy associated with kinetic and potential energy of molecules in a body is called:

- [ ] The total energy
- [x] The internal energy
- [ ] The stagnation energy
- [ ] none of the above

Two Carnot cycles with the same heat extraction temperature $T_{e1} = T_{e2}$, and different heat addition temperatures $T_{a1} = 2*T_{a2}$ have efficiencies η_1 and η_2, where:

- [ ] $\eta_1 = 2*\eta_2 - 1/2$
- [ ] $\eta_1 = \eta_2/2 + 1$
- [x] $\eta_1 = \eta_2/2 + 1/2$
- [ ] none of the above

The maximum work Lmax of a Carnot cycle with efficiency η and where added heat is Qa:

- [ ] Cannot be determined
- [ ] Is equal to $L_{max} = Q_a / \eta$
- [x] Is equal to $L_{max} = Q_a * \eta$
- [ ] none of the above

For a reversible adiabatic process in the control mass system the entropy:

- [ ] Always increases
- [ ] In some cases decreases
- [x] Is unchanged
- [ ] none of the above

The mechanical turbine efficiency is defined as a ratio of:

- [ ] The internal to effective turbine power
- [x] The effective to internal turbine power

- [ ] The internal to theoretical turbine power
- [ ] none of the above

For control volume system, its boundary can be crossed:

- [x] By heat
- [x] By mass
- [x] By both heat and mass
- [ ] none of the above

The following property is an example of the extensive property:

- [ ] Pressure p [Pa]
- [x] Enthalpy I [J]
- [x] Internal energy EI [J]
- [ ] none of the above

Energy transfer into a control mass system can take place by:

- [x] work
- [x] conduction
- [ ] convection
- [ ] none of the above

The fundamental difference between the energy and the exergy of a system is that:

- [ ] The exergy is conserved but the energy is not
- [x] The energy is conserved but the exergy is not
- [ ] They have different physical units
- [ ] none of the above

For a given finite thermodynamic process, the maximum work can be determined by applying:

- [ ] The first principle of thermodynamics only
- [ ] The second principle of thermodynamics only
- [x] Both the first and the second principle of thermodynamics together
- [ ] none of the above

For an isobaric process from state 1 to 2, according to the first principle of thermodynamics, the heat change δ_q is:

- [ ] Equal to internal energy change e_{I2}-e_{I1}
- [ ] Always equal to zero
- [x] Equal to enthalpy change $i_2 – i_1$
- [ ] none of the above

For an isobaric process of an ideal gas from state 1 to 2, with $T_2 = 2.7183*T_1$ the specific entropy will:

- [ ] Not change
- [x] Increase approximately by c_p (specific heat at constant pressure)
- [ ] Increase approximately by c_v (specific heat at constant volume)
- [ ] none of the above

Energy in transit from a system with higher temperature to a system with lower temperature is called:

- [x] The heat
- [ ] The enthalpy
- [ ] The entropy
- [ ] none of the above

The theoretical turbine power is such a power that can be calculated based on:

- [ ] Known mechanical (friction) losses in the turbine
- [x] Assumption of isentropic steam expansion in the turbine
- [ ] Known inlet and outlet enthalpies of the turbine
- [ ] none of the above

For an isothermal process of an ideal gas from state 1 to 2, where $p_2 = p_1/2.7183$, the specific entropy will:

- [x] Increase approximately by R (specific gas constant)
- [ ] Decrease approximately by R (specific gas constant)
- [ ] Not change
- [ ] none of the above

Steam turbine in a thermal power plant operates in a Rankine cycle and has an effective power 100 (MW), mechanical efficiency 0.97 and internal efficiency 0.84. Mass flow rate of cooling water in the condenser is 4,200 (kg/s) and its mean temperature increase is 8 (K). Specific heat of the cooling water is 4,190 (J/kg.K).

Assuming no heat losses in the system and neglecting the needed pumping power calculate the energy efficiency of the ideal Rankine cycle in the power plant.

η_{EIR} = 50.324 %

Solution:
(a) internal power of the turbine

$$N_i = \frac{N_e}{\eta_m};$$

(b) thermal power extracted in the condenser

$$q_e = W_w * c_{pw} * \Delta T_w$$

(c) theoretical power

$$N_{th} = \frac{N_i}{\eta_i}$$

(d) total added thermal power to the cycle

$$q_a = q_e + N_i$$

(e) efficiency of the ideal Rankine cycle

$$\eta_{EIR} = \frac{N_{th}}{q_a};$$

137.SORU

A tank with volume V = 20 (m^3) contains nitrogen (N_2) at pressure p_1 = 150,000 (Pa) and temperature T_1 = 340 (K).

An electrical heater with electrical power N_{el} = 10,000 (W) is installed inside the tank and heats up the gas to temperature T_2 = 850 (K). A fraction (0.2) of the heat provided by the heater is absorbed by the tank's walls.

Assuming:
(1) no heat accumulation in the heater
(2) ideal gas model for nitrogen

(3) Heat capacity ratio κκ = 1.4

Calculate:
(a) the heat absorbed by the gas Q_{12} (MJ)

Q_{12} =

(b) the time needed to heat up the gas t_{12} (s)

t_{12} =

A steam turbine operates at steady-state conditions and delivers effective (shaft) power N_e = 420 MW. Steam mass flow rate through the turbine is W = 499 kg/s. The inlet specific enthalpy and the mean velocity of the steam are i_{in} = 2,780 kJ/kg and U_{in} = 89 m/s, respectively. The corresponding parameters for steam at the outlet are i_{out} = 1,850 kJ/kg and U_{out} = 226 m/s.

Neglect potential energy of the steam streams. Calculate the turbine heat losses to the surroundings.

Q_{loss} = 33.3028 MW_{th}

Solution:

(a) internal power of the turbine

$N_i = W*(i_{in}+(U_{in}^2)/2-i_{out}-(U_{out}^2)/2)$;

(b) thermal power loss

q_{loss} = Ni - Ne;

139. SORU

Steam turbine in a thermal power plant has an effective power 100 (MW) and the mechanical efficiency 0.99. Specific steam usage in relation to the effective power is 8.2E-6 (kg/J of eff. power). Mass flow rate of cooling water in the condenser is 4,700 (kg/s) and its mean temperature increase is 9 (K). Specific heat of the cooling water is 4,190 (J/kg.K). Heat losses in the pipline between the boiler and the turbine are equal to 44,000 (J per kg of flowing steam).

Calculate the energy efficiency of the thermodynamic cycle in the power plant. Neglect the needed pumping power.

η_E = 32.135 %

Solution:

(a) internal power of the turbine

$$N_i = \frac{N_e}{\eta m}$$

(b) thermal power extracted in the condenser

$$q_e = W_w * cp_w * \Delta T_w$$

(c) mass flow of steam in the turbine

$$W = U_{st} * N_e$$

(d) thermal power loss in the pipeline

$$q_{loss} = Q_{loss} * W$$

(e) total added thermal power to the cycle

$$q_a = q_{loss} + q_e + N_i$$

(f) Energy efficiency of the cycle

$$eta_E = \frac{N_i}{q_a}$$

A gear transmission in a windmil has an output power N_{ext} = 1,500,000 (W) and mechanical efficiency η_M = 0.95.

Calculate the amount of heat Q (kW) generated in the gear due to friction.

Q = 78.94 kW

The added power is $N_{add} = N_{ext}$/eta
and from the first law:

Q = N_{add} - N_{ext} = N_{ext}*(1/eta-1)

Q = N_{ext}*(1.0/eta-1.0)

An ideal gas mixture with known kappa = c_p/c_v = 1.582 undergoes in a closed system a frictionless process described with a linear function p(V).

Calculate the amonnt of heat Q_{12} (kJ) supplied to the mixture for known initial pressure p_1 = 700,000 (Pa) and volume V_1 = 0.05 (m^3) and final pressure p_2 = 240,000 (Pa) and volume V_2 = 0.3 (m^3).

Q_{12} = 181.07 kJ

from the first law we have $Q_{12} = E_{I2} - E_{I1} + L_{12}$ where:

- $E_{I1,2}$ is the initial and final internal energy and
- L_{12} is the work done during the process.
- L_{12} = integral from V1 to V2 of p(V)dV. Since p(V) is linear,
- $L12 = \frac{(p_1 + p_2) * (V_2 - V_1)}{2}$ (mean pressure times volume change).

Hence,

$$E_{I2} - E_{I1} = m * c_v * (T_2 - T_1) = m\left(\frac{R}{(1-\kappa)}\right) * \frac{(p_1 * V_1 - p_2 * V_2)}{(m * R)} =$$

$$= \frac{(p_1 * V_1 - p_2 * V_2)}{(1-\kappa)}$$

.

We arrive at:

$$Q_{12} = \frac{(p_1 * V_1 - p_2 * V_2)}{(1-\kappa)} + \frac{(p_1 + p_2) * (V_2 - V_1)}{2}$$

For steady-state flow of incompressible fluid in a pipe, the mass flow rate W (kg/s):

- ☐ Is a function of distance z, that is W=f(z), if the pipe cross section area A is a function of z
- ☐ Is a function of distance z, that is W=f(z), if the fluid pressure p is a function of z
- ☑ Is always constant, that is W = const

- [ ] none of the above

For laminar viscous flow with mean velocity U in a round tube with constant radius R, the wall shear stress is:

- [x] Proportional to U/R
- [ ] Proportional to U^2/R
- [ ] Proportional to U/R^2
- [ ] none of the above

For definition of a local loss coefficient at any obstacle with different downstream and upstream flow areas, it is customary to use as a reference:

- [ ] The upstream flow area
- [ ] The downstream flow area
- [x] The smaller of the two flow areas
- [ ] none of the above

For flow of fluid in a channel with a sudden expansion, assuming the flow direction as a positive direction, the reversible and irreversible pressure changes:

- [ ] Are both positive
- [ ] Are both negative
- [x] They have different signs
- [ ] none of the above

For two-phase saturated mixture flowing in a uniformly heated channel, the integral acceleration multiplier derived from the Homogeneous Equilibrium Model is:

- [ ] never greater than 1
- [ ] always greater than 1
- [ ] always decreases with increasing exit quality
- [x] none of the above

Volumetric flow rates of water and air in a pipe are 1 m3/s and 5 m3/s, respectively. The total mixture volumetric flow rate is:

- [x] exactly 6 m^3/s
- [ ] approximately 1/1000 + 5/1.3 = 3.847m^3/s
- [ ] unknown, since densities of water and air are not given
- [ ] none of the above

The thermodynamic equilibrium quality and the actual quality are:

- [ ] always equal to each other

- [ ] always in a range from zero to one
- [ ] never equal to each other
- [x] none of the above

According to the Homogeneous Equilibrium Model, irreversible pressure loss at a local obstacle for two-phase mixture, when quality x=0.2 and density ratio ρ_f/ρ_g=26, in comparison to saturated liquid flow is:

- [ ] approximately the same
- [ ] unknown due to unknown total mass flux
- [x] exactly 6 times higher
- [ ] none of the above

Void fraction calculated from the Drift Flux Model, in comparison to the void fraction calculated from the Homogeneous Equilibrium Mode is:

- [ ] exactly the same
- [x] exactly the same only when C_0=1 and U_{vj} = 0
- [ ] always less, irrespective of C_0 and U_{vj} values
- [ ] none of the above

Churn two-phase flow occurs when, in comparison to annular two-phase flow, the momentum of gas phase is:

- [ ] approximately the same
- [ ] higher
- [x] lower
- [ ] none of the above

The Fanning friction factor for laminar flow of fluid with Reynolds number Re, in a round tube, is:

- [x] C_f = 16/Re
- [ ] C_f = 32/Re
- [ ] C_f = 64/Re
- [ ] none of the above

In a uniformly heated channel with constant cross-section area and constant heated perimeter, the thermodynamic equilibrium quality is:

- [x] always increasing with distance from inlet
- [ ] always positive
- [x] Always increasing linearly with distance from inlet

☐ none of the above

For a horizontal, frictionless and steady flow of incompressible fluid in a pipe with constant cross section, the pressure will:

☑ Remain constant along the pipe
☐ Decrease in the direction of flow
☐ Increase in the direction of flow
☐ none of the above

For viscous flow of fluid with mean velocity U, density ρ and wall shear stress τw, the Fanning friction factor is defined as:

☐ $Cf=4\tau w/(\rho U^2/2)$
☑ $Cf=\tau w/(\rho U^2/2)$
☐ $Cf=8\tau w/(\rho U^2/2)$
☐ none of the above

According to the Homogeneous Equilibrium Model, the void fraction of saturated water/steam mixture flowing in a channel with known mass fluxes for each phase can be uniquely determined only when:

☑ pressure in the channel is known
☐ channel cross-section area is known
☐ channel orientation against gravity is known
☐ none of the above

For a horizontal, frictionless and steady flow of incompressible fluid in a pipe with an increasing cross-section area in the flow direction, the pressure will:

☐ Remain constant along the pipe
☐ Decrease in the direction of flow
☑ Increase in the direction of flow
☐ none of the above

For viscous flow of fluid with mean velocity U, density ρ and wall shear stress τ_w, the Darcy-Weisbach friction factor is defined as:

☑ $\lambda= Cf=4\tau w/(\rho U^2/2)$
$Cf=\tau w/(\rho U^2/2)$
$Cf=8\tau w/(\rho U^2/2)$
none of the above

For flow of fluid in a channel with a sudden contraction, assuming the flow direction as a positive direction, the reversible and irreversible pressure changes:

- [ ] Are both positive
- [x] Are both negative
- [ ] They can have different signs, depending on mass flow rate
- [ ] none of the above

Volumetric flow rates of water and air in a pipe are 1 m3/s and 5 m3/s, respectively. The mixture actual quality is:

- [ ] exactly 20%
- [ ] approximately 83.3%
- [x] unknown, since densities of water and air are not given
- [ ] none of the above

For two-phase saturated mixture flowing in a uniformly heated channel, the integral gravity multiplier derived from the Homogeneous Equilibrium Model is:

- [x] always increases with increasing pressure
- [x] always decreases with increasing exit quality
- [x] never greater than 1
- [ ] none of the above

162)

Saturated water/steam mixture at temperature T=522.01 K flows downwards through a vertical pipe with diameter 130 mm and length 13 m.

Calculate the total pressure change $\Delta p = p_2 - p_1$ over the pipe length when the mass flow rate of water is 0.7 kg/s and the mass flow rate of steam is 1.64 kg/s.

Assume the same water/steam properties everywhere.
Assume wall roughness 0.02 mm.
Neglect the local inlet and outlet losses.

Δp =2894.782 Pa

-273.15=-273.15;
10=10.;
0.001=1.E-3;
1=1.;
1=1.;

248.86=522.01+-273.15;

130 = range(100,200,10);
0.13=130*0.001;
0.02 = range(0.01,0.10,0.01);
2E-5=0.02*0.001;
13 = range(5,15,1);
13=13*1;
0.7 = range(0.1,2,0.1);
0.7=0.7*1;
1.64 = range(0.01,2,0.01);
1.64=1.64*1;

800.58=800.58;
19.573=19.573;
1.0664E-4=1.0664E-4;
1.7452E-5=1.7452E-5;
0.013273=3.141593*0.13^2/4;
2.34=0.7+1.64;
0.700855=1.64/2.34;
176.294692=2.34/0.013273;
214,912.884096 = 176.294692*0.13/1.0664E-4;: % Liquid-only Reynolds number;
7.810209=-1.8*log((2E-5/0.13/3.7)^1.11+6.9/214,912.884096);
0.004098=1/4/7.810209^2;
19.798268=(1+(800.58/19.573-1)*0.700855)/(1+(1.0664E-4/1.7452E-5-1)*0.700855)^0.25;
-630.00685 = -19.798268 * (4*0.004098*13/0.13) * (176.294692^2/2/800.58);
0.989672=1/(1+(1-0.700855)/0.700855*(19.573/800.58));
0.034524=1-(800.58-19.573)/800.58*0.989672;
3,524.789387 = 9.81*0.034524*800.58*13;
2,894.78254 = -630.00685 + 3,524.789387;

163)

Saturated water/steam mixture at temperature T=453.04 K flows through a horizontal pipe with diameter 140 mm and length 30 m.

Calculate the total pressure change $\Delta p = p_2 - p_1$ over the pipe length when the mass flow rate of water is 1.25 kg/s and the mass flow rate of steam 1 kg/s.

Assume the same water/steam properties everywhere.

Assume wall roughness 0.01 mm.

Neglect the local inlet and outlet losses.

Δp = -2277.6443 Pa

-273.15=-273.15;
10=10.;
0.001=1.E-3;
1=1.;
1=1.;
179.89=453.04+-273.15;
140 = range(100,200,10);
0.14=140*0.001;
0.01 = range(0.01,0.10,0.01);
1E-5=0.01*0.001;
30 = range(10,50,5);
30=30*1;
1.25 = range(1,2,0.25);
1.25=1.25*1;
1 = range(1,2,0.25);
1=1*1;
887.12=887.12;
5.1459=5.1459;
:psat=XSteam('psat_T',T) % saturation pressure, bar;
1.5024E-4=1.5024E-4; :% =XSteam('my_pT',psat,T-0.01) % to get liquid viscosity;
1.5022E-5=1.5022E-5; :% =XSteam('my_pT',psat,T+0.01) % to get vapor viscosity;
0.015394=pi*0.14^2/4;
2.25=1.25+1;
0.444444=1/2.25;
146.162741=2.25/0.015394;
136,200.637247 = 146.162741*0.14/1.5024E-4; :% Liquid-only Reynolds number;
7.646178=-1.8*log((1E-5/0.14/3.7)^1.11+6.9/136,200.637247);

0.004276=1/4/7.646178^2;
51.608448=(1+(887.12/5.1459-1)*0.444444)/(1+(1.5024E-4/1.5022E-5-1)*0.444444)^0.25;
-2,277.644301 = -51.608448 * (4*0.004276*30/0.14) * (146.162741^2/2/887.12);

164)

Water at temperature T=350 K and pressure 0.25 MPa flows upwards through a vertical pipe with diameter 140 mm and length 26 m.

Calculate the total pressure change $\Delta p = p_2 - p_1$ over the pipe length when the mass flow rate of water is 175 kg/s.

Assume the same water properties everywhere.
Assume wall roughness 0.05 mm.
Neglect the local inlet and outlet losses

Δp = -441525.343 Pa

-273.15=-273.15;
10=10.;
0.001=1.E-3;
1=1.;
1=1.;
76.85=350+-273.15;

140 = range(100,500,10);
0.14=140*0.001; :% variation from 100 to 500 mm;
0.05= range(0.01,0.10,0.01);
5E-5=0.05*0.001; :% variation from 0.01 to 0.1 mm;
26 = range(10,50,1);
26=26*1; : % variation from 10 to 50 m;
175 = range(50,250,25);
175=175*1; :% variation from 50 to 250 kg/s;
973.81=973.81;
3.6883E-4=3.6883E-4;
0.015394=pi*0.14^2/4;
11.673954=175/(973.81*0.015394);
4,315,131.19937 = 11.673954*0.14*973.81/3.6883E-4;
7.987678=-1.8*log((5E-5/0.14/3.7)^1.11+6.9/4,315,131.19937);
0.003918=1/4/7.987678^2;
-193,145.36396 = -(4*0.003918*26/0.14) * (175^2/2/973.81/0.015394^2);
-248,379.9786 = -9.81*973.81*26;
-441,525.343 = -193,145.36396 + -248,379.9786;

165)

Water at temperature T=350 K and pressure 0.25 MPa flows
through a horizontal pipe with diameter 190 mm and length 32 m.

Calculate the total pressure change $\Delta p = p_2 - p_1$ over the pipe length when the mass flow rate of water is 200 kg/s.

Assume the same water properties everywhere.

Assume wall roughness 0.1 mm.

Neglect the local inlet and outlet losses.

Δp = -73351.518368 Pa

-273.15=-273.15;
10=10.;

0.001=1.E-3;
1=1.;
1=1.;
76.85=350+-273.15;
2.5=0.25*10;
190 = range(100,500,10);
0.19=190*0.001;   :% variation from 100 to 500 mm;
0.1= range(0.01,0.10,0.01);
1E-4=0.1*0.001;   :% variation from 0.01 to 0.1 mm;
32 = range(10,50,1);
32=32*1;   : % variation from 10 to 50 m;
200 = range(50,250,25);
200=200*1;   :% variation from 50 to 250 kg/s;
973.81=973.81;
3.6883E-4=3.6883E-4;
0.028353=pi*0.19^2/4;
7.243671=200/(973.81*0.028353);
3,633,794.233646 = 7.243671*0.19*973.81/3.6883E-4;
7.659046=-1.8*log((1E-4/0.19/3.7)^1.11+6.9/3,633,794.233646);
0.004262=1/4/7.659046^2;
-73,351.518368 = -(4*0.004262*32/0.19) * (200^2/2/973.81/0.028353^2);

166)

Water steam at temperature T=750 K and pressure 7.0 MPa flows upwards through a vertical pipe with diameter 150 mm and length 49 m.

Calculate the total pressure change Δp = p2-p1 over the pipe length when the mass flow rate of water steam is 25 kg/s.

Assume the same water steam properties everywhere.
Assume wall roughness 0.08 mm.
Neglect the local inlet and outlet losses.

Δp =-268448.5601 Pa

-273.15=-273.15;
10=10.;
0.001=1.E-3;
1=1.;
1=1.;
476.85=750+-273.15;
70=7.0*10;

150 = range(100,500,10);
0.15=150*0.001;   :% variation from 100 to 500 mm;
0.08= range(0.01,0.10,0.01);
8E-5=0.08*0.001;   :% variation from 0.01 to 0.1 mm;
49 = range(10,50,1);
49=49*1;   : % variation from 10 to 50 m;
25 = range(5,25,5);
25=25*1;   :% variation from 5 to 25 kg/s;
21.577=21.577;
2.7767E-5=2.7767E-5;
0.017671=pi*0.15^2/4;
65.565672=25/(21.577*0.017671);
7,642,401.942617 = 65.565672*0.15*21.577/2.7767E-5;
7.661876=-1.8*log((8E-5/0.15/3.7)^1.11+6.9/7,642,401.942617);
0.004259=1/4/7.661876^2;
-258,076.712339 = -(4*0.004259*49/0.15) * (25^2/2/21.577/0.017671^2);

-10,371.84813 = -9.81*21.577*49;
-268,448.5601 = -258,076.712339 + -10,371.84813;

167)

Water at temperature T=370 K and pressure 0.27 MPa flows through a sudden expansion with diameter change from 350 mm to 700 mm.

Calculate the total pressure change $\Delta p = p_2 - p_1$ over the sudden expansion when the volumetric flow rate of water is 0.2 m^3/s.

Assume the same water properties everywhere.

Water-steam properties:

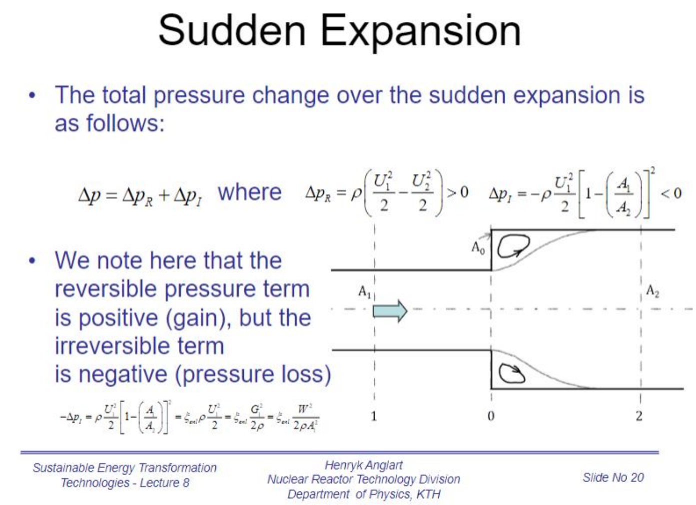
Density of water at temperature T=370 K and pressure 0.27 MPa is 960.68 kg/m^3;

Viscosity of water at temperature T=370 K and pressure 0.27 MPa is 2.91×10^{-4} Pas.

Δp = 778.37361 Pa

```
-273.15=-273.15;
10=10.;
0.001=1.E-3;
1=1.;

96.85=370+-273.15;
2.7=0.27*10;
350 = range(100,450,50);
0.35=350*0.001;   :% variation from 100 to 450 mm;
700 = range(500,750,50);
0.7=700*0.001;   :% variation from 500 to 750 mm;
0.2 = range(0.1,1,0.1);
0.2=0.2*1;    :% variation from 0.1 to 1 m3/s;
960.68=960.68; :% =XSteam('rho_pT',p,T);
0.096211=pi*0.35^2/4;
0.384845=pi*0.7^2/4;
2.078759=0.2/0.096211;
0.51969=0.2/0.384845;
1,945.934107 = 960.68*(2.078759^2-0.51969^2)/2;
0.5625=(1-0.096211/0.384845)^2;
-1,167.560498 = -0.5625*960.68*2.078759^2/2;
778.37361 = 1,945.934107 + -1,167.560498;
```



168)

Water steam at temperature T=750 K and pressure 7.0 MPa flows downwards through a vertical pipe with diameter 310 mm and length 48 m.

Calculate the total pressure change Δp = p2-p1 over the pipe length when the mass flow rate of water steam is 10 kg/s.

Assume the same water steam properties everywhere.
Assume wall roughness 0.05 mm.
Neglect the local inlet and outlet losses.

Water-steam properties:

Density of water steam at temperature T=750 K and pressure 7.0 MPa is 21.577 kg/m^3;

Viscosity of water steam at temperature T=750 K and pressure 7.0 MPa is 2.777x10^{-5} Pas.

Δp =9287.130384 Pa

```
-273.15=-273.15;
10=10.;
0.001=1.E-3;
1=1.;
1=1.;
476.85=750+-273.15;
70=7.0*10;

310 = range(100,500,10);
0.31=310*0.001;    :% variation from 100 to 500 mm;
0.05= range(0.01,0.10,0.01);
5E-5=0.05*0.001;   :% variation from 0.01 to 0.1 mm;
48 = range(10,50,1);
48=48*1;     : % variation from 10 to 50 m;
10 = range(5,25,5);
10=10*1;   :% variation from 5 to 25 kg/s;
21.577=21.577;
2.7767E-5=2.7767E-5;
0.075477=pi*0.31^2/4;
6.140386=10/(21.577*0.075477);
1,479,174.750933 = 6.140386*0.31*21.577/2.7767E-5;
8.493708=-1.8*log((5E-5/0.31/3.7)^1.11+6.9/1,479,174.750933);
0.003465=1/4/8.493708^2;
-873.047416 = -( 4*0.003465*48/0.31) * (10^2/2/21.577/0.075477^2);
10,160.17776 = 9.81*21.577*48;
9,287.130384 = -873.047416 + 10,160.17776;
```

169)

Saturated water/steam mixture at pressure 3 MPa flows downwards through a vertical uniformly heated pipe with diameter 170 mm and length 15 m.

Calculate the total pressure change Δp = p_2 - p_1 over the pipe length when the mass flow rate of saturated water at the inlet (x_{in}=0) is 5 kg/s and the exit therodynamic equilibrium quality is x_{ex}=0.4.

Assume the same water/steam properties everywhere.
Assume wall roughness 0.06 mm.

Use Haaland's Correlation for friction factor.
Neglect the local inlet and outlet losses.

Water-steam properties:

Density of saturated water at 3 MPa is 821.8949 kg/m[3];

Density of saturated steam at 3 MPa is 15.0006 kg/m[3];

Viscosity of saturated water at 3 MPa is $1.1395x10^{-4}$ Pas;

Viscosity of saturated steam at 3 MPa is $1.6903x10^{-5}$ Pas.

Δp = 15211.35648 Pa

```
-273.15=-273.15;
10=10.;
0.001=1.E-3;
1=1.;
1=1.;

170 = range(100,200,10);
0.17=170*0.001;
0.06 = range(0.01,0.10,0.01);
6E-5=0.06*0.001;
15 = range(5,15,5);
15=15*1;
5 = range(1,5,0.25);
5=5*1;
821.8949=821.8949;
1.1395E-4=1.1395E-4;
0.022698=pi*0.17^2/4;
5=5;
220.283628=5/0.022698;
328,637.268627 = 220.283628*0.17/1.1395E-4; :% Liquid-only Reynolds number;
7.660973=-1.8*log((6E-5/0.17/3.7)^1.11+6.9/328,637.268627);
0.00426=1.0/4.0/7.660973^2;
9.5=9.5; :% found from figure (b);
-421.61492 = -9.5 * ( 4*0.00426*15/0.17) * (220.283628^2/2/821.8949);
0.14=0.14; :% found from figure (c);
16,931.856835 = 9.81*0.14*821.8949*15;
22=22.0; :% found from figure (a);
-1,298.885404 = -22 * (220.283628^2/821.8949);
15,211.35648 = -421.61492 + 16,931.856835 + -1,298.885404;
```

170)

Saturated water/steam mixture at temperature T=471.45 K flows upwards through a vertical pipe with diameter 190 mm and length 11 m.

Calculate the total pressure change $\Delta p = p_2 - p_1$ over the pipe length when the mass flow rate of water is 0.2 kg/s and the mass flow rate of steam is 0.27 kg/s.

Assume the same water/steam properties everywhere.
Assume wall roughness 0.09 mm.
Neglect the local inlet and outlet losses.

Δp = -1427.437299 Pa

```
-273.15=-273.15;
10=10.;
0.001=1.E-3;
1=1.;
1=1.;
```

198.3=471.45+-273.15;
190 = range(100,200,10);
0.19=190*0.001;
0.09 = range(0.01,0.10,0.01);
9E-5=0.09*0.001;
11 = range(5,15,1);
11=11*1;
0.2 = range(0.1,2,0.1);
0.2=0.2*1;
0.27 = range(0.01,2,0.01);
0.27=0.27*1;
866.64=866.64;
7.5936=7.5936;
1.3554E-4=1.3554E-4;
1.5657E-5=1.5657E-5;
0.028353=3.141593*0.19^2/4;
0.47=0.2+0.27;
0.574468=0.27/0.47;
16.576805=0.47/0.028353;
23,237.368814 = 16.576805*0.19/1.3554E-4;: % Liquid-only Reynolds number;
6.23261=-1.8*log((9E-5/0.19/3.7)^1.11+6.9/23,237.368814);
0.006436=1/4/6.23261^2;
43.290858=(1+(866.64/7.5936-1)*0.574468)/(1+(1.3554E-4/1.5657E-5-1)*0.574468)^0.25;
-10.228879 = -43.290858 * (4*0.006436*11/0.19) * (16.576805^2/2/866.64);
0.993551=1/(1+(1-0.574468)/0.574468*(7.5936/866.64));
0.015154=1-(866.64-7.5936)/866.64*0.993551;
-1,417.20842 = -9.81*0.015154*866.64*11;
-1,427.437299 = -10.228879 + -1,417.20842;

171)

Water steam at temperature T = 770 K and pressure 7.2 MPa flows through a sudden expansion with diameter change from 450 mm to 500 mm.

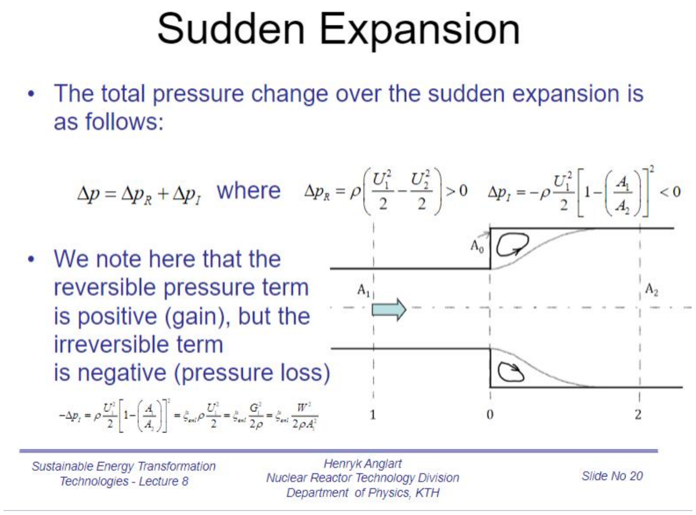
Calculate the total pressure change $\Delta p = p_2 - p_1$ over the sudden expansion when the volumetric flow rate of water steam is 0.3 m^3/s.

Assume the same water steam properties everywhere.

Δp = 11.775808 Pa

-273.15=-273.15;
10=10.;
0.001=1.E-3;
1=1.;
496.85=770+-273.15;
72=7.2*10;
450 = range(100,450,50);
0.45=450*0.001;
500 = range(500,750,50);
0.5=500*0.001;
0.3 = range(0.10,1.00,0.10);
0.3=0.3*1;
21.505=21.505;
0.159043=pi*0.45^2/4;
0.19635=pi*0.5^2/4;
1.886281=0.3/0.159043;
1.527887=0.3/0.19635;
13.156921 = 21.505*(1.886281^2-1.527887^2)/2;
0.0361=(1-0.159043/0.19635)^2;

-1.381113 = -0.0361*21.505*1.886281^2/2;
11.775808 = 13.156921 + -1.381113;



172)

Water at temperature T=350 K and pressure 0.25 MPa flows downwards through a vertical pipe with diameter 480 mm and length 43 m.

Calculate the total pressure change $\Delta p = p_2 - p_1$ over the pipe length when the mass flow rate of water is 175 kg/s.

Assume the same water properties everywhere.
Assume wall roughness 0.07 mm.
Neglect the local inlet and outlet losses

Δp = 410190.116692 Pa

-273.15=-273.15;
10=10.;
0.001=1.E-3;
1=1.;
1=1.;
76.85=350+-273.15;

480 = range(100,500,10);
0.48=480*0.001; :% variation from 100 to 500 mm;
0.07= range(0.01,0.10,0.01);
7E-5=0.07*0.001; :% variation from 0.01 to 0.1 mm;
43 = range(10,50,1);
43=43*1; : % variation from 10 to 50 m;
175 = range(50,250,25);
175=175*1; :% variation from 50 to 250 kg/s;
973.81=973.81;
3.6883E-4=3.6883E-4;
0.180956=pi*0.48^2/4;
0.993097=175/(973.81*0.180956);
1,258,579.589387 = 0.993097*0.48*973.81/3.6883E-4;
8.523335=-1.8*log((7E-5/0.48/3.7)^1.11+6.9/1,258,579.589387);
0.003441=1/4/8.523335^2;
-592.155308 = -(4*0.003441*43/0.48) * (175^2/2/973.81/0.180956^2);
410,782.2723 = 9.81*973.81*43;
410,190.116692 = -592.155308 + 410,782.2723;

173)

Water at temperature T=390 K and pressure 0.29 MPa flows through a sudden contraction with diameter change from 700 mm to 200 mm.

Calculate the total pressure change $\Delta p = p_2 - p_1$ over the sudden contraction when the volumetric flow rate of water is 0.05 m^3/s.

Assume the same water properties everywhere.

Water-steam properties:

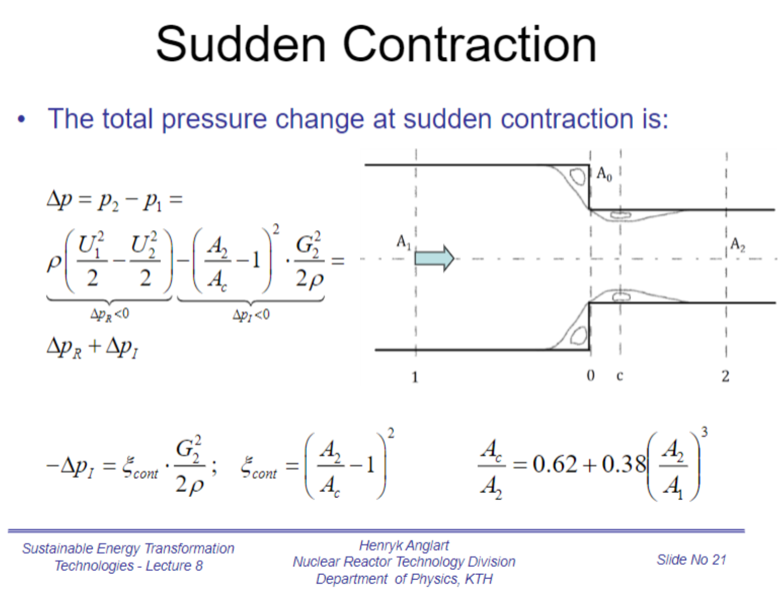
Density of water at temperature T=390 K and pressure 0.29 MPa is 945.68 kg/m^3;

Viscosity of water at temperature T=390 K and pressure 0.29 MPa is 2.39×10^{-4} Pas.

Δp = -1638.870068 Pa

```
-273.15=-273.15;
10=10.;
0.001=1.E-3;
1=1.;
116.85=390+-273.15;
2.9=0.29*10;
200 = range(100,450,50);
0.2=200*0.001;
700 = range(500,750,50);
0.7=700*0.001;
0.05 = range(0.05,0.20,0.05);
0.05=0.05*1;

945.68=945.68;
0.384845=pi*0.7^2/4;
0.031416=pi*0.2^2/4;
0.129922=0.05/0.384845;
1.591549=0.05/0.031416;
-1,189.735996 = 945.68*(0.129922^2-1.591549^2)/2;
0.620207 = 0.62 + 0.38*(0.031416/0.384845)^3;
0.374992=(1/0.620207-1)^2;
-449.134068 = -0.374992*945.68*1.591549^2/2;
-1,638.870068 = -1,189.735996 + -449.134068;
```



174)

Saturated water/steam mixture at temperature T=494.95 K flows upwards through a vertical uniformly heated pipe with diameter 200 mm and length 5 m.

Calculate the total pressure change Δp = p_2 - p_1 over the pipe length when the mass flow rate of saturated water at the inlet is 2 kg/s and the exit therodynamic equilibrium quality is x_{ex}=0.45.

Assume the same water/steam properties everywhere.
Assume wall roughness 0.04 mm.
Neglect the local inlet and outlet losses.

Δp = -3857.115891 Pa

-273.15=-273.15;
10=10.;
0.001=1.E-3;
1=1.;
1=1.;
221.8=494.95+-273.15;
200 = range(100,200,10);
0.2=200*0.001;
0.04 = range(0.01,0.10,0.01);
4E-5=0.04*0.001;
5 = range(5,15,5);
5=5*1;
2 = range(1,5,0.25);
2=2*1;
837.91=837.91;
1.2049E-4=1.2049E-4;
0.031416=pi*0.2^2/4;
2=2;
63.66197=2/0.031416;
105,671.790522 = 63.66197*0.2/1.2049E-4; :% Liquid-only Reynolds number;
7.339671=-1.8*log((4E-5/0.2/3.7)^1.11+6.9/105,671.790522);
0.004641=1/4/7.339671^2;
15.5=15.5; :% found from figure (b);
-17.396055 = -15.5 * (4*0.004641*5/0.2) * (63.66197^2/2/837.91);
0.089=0.089; :% found from figure (c);
-3,657.85421 = -9.81*0.089*837.91*5;
37.6=37.6; :% found from figure (a);
-181.865626 = -37.6 * (63.66197^2/837.91);
-3,857.115891 = -17.396055 + -3,657.85421 + -181.865626;

175)

Saturated water/steam mixture at temperature T=537.09 K flows through a horizontal uniformly heated pipe with diameter 160 mm and length 35 m.

Calculate the total pressure change Δp = p_2 - p_1 over the pipe length when the mass flow rate of saturated water at the inlet is 4.5 kg/s and exit thermodynamic equilibrium quality is x_{ex}=0.65.

Assume the same water/steam properties everywhere.
Assume wall roughness 0.1 mm.
Neglect the local inlet and outlet losses.

Δp = -2344.315723 Pa

-273.15=-273.15;
10=10.;

0.001=1.E-3;
1=1.;
1=1.;

263.94=537.09+-273.15;

160 = range(100,200,10);
0.16=160*0.001;
0.1 = range(0.01,0.10,0.01);
1E-4=0.1*0.001;
35 = range(10,50,5);
35=35*1;
4.5 = range(2,5,0.25);
4.5=4.5*1;

777.36=777.36;

1.0001E-4=1.0001E-4; : =XSteam('my_pT',psat,T-0.01) to get liquid viscosity;
0.020106=pi*0.16^2/4;
4.5=4.5;
223.811672=4.5/0.020106;
358,062.868913 = 223.811672*0.16/1.0001E-4; : Liquid-only Reynolds number;
7.33405=-1.8*log((1E-4/0.16/3.7)^1.11+6.9/358,062.868913);
0.004648=1/4/7.33405^2;
8.4=8.4; : found from figure for given xex and pressure;
19.3=19.3; : found from figure for given xex and pressure;
-2,344.315723 = -(8.4 * (4*0.004648*35/0.16) + 2*19.3) * (223.811672^2/2/777.36);

176)

Water steam at temperature T=790 K and pressure 7.5 MPa flows through a sudden contraction with diameter change from 550 mm to 450 mm.

Calculate the total pressure change $\Delta p = p_2 - p_1$ over the sudden contraction when the volumetric flow rate of water is 0.2 m^3/s.

Assume the same water properties everywhere.

Δp =-11.749429 Pa

-273.15=-273.15;
10=10.;
0.001=1.E-3;
1=1.;

516.85=790+-273.15;
75=7.5*10;
450 = range(100,450,50);
0.45=450*0.001;
550 = range(500,750,50);
0.55=550*0.001;
0.2 = range(0.05,0.20,0.05);
0.2=0.2*1;

21.75=21.75;
0.237583=pi*0.55^2/4;
0.159043=pi*0.45^2/4;
0.841811=0.2/0.237583;
1.257521=0.2/0.159043;
-9.490739 = 21.75*(0.841811^2-1.257521^2)/2;
0.733994 = 0.62 + 0.38*(0.159043/0.237583)^3;
0.13134=(1/0.733994-1)^2;
-2.25869 = -0.13134*21.75*1.257521^2/2;
-11.749429 = -9.490739 + -2.25869;

177)

Water steam at temperature T=750 K and pressure 7.0 MPa flows through a horizontal pipe with diameter 120 mm and length 39 m.

Calculate the total pressure change Δp = p2-p1 over the pipe length when the mass flow rate of water steam is 25 kg/s.

Assume the same water steam properties everywhere.
Assume wall roughness 0.02 mm.
Neglect the local inlet and outlet losses.

Δp = -492137.180128 Pa

```
-273.15=-273.15;
10=10.;
0.001=1.E-3;
1=1.;
1=1.;
476.85=750+-273.15;
70=7.0*10;

120 = range(100,500,10);
0.12=120*0.001;   :% variation from 100 to 500 mm;
0.02= range(0.01,0.10,0.01);
2E-5=0.02*0.001;   :% variation from 0.01 to 0.1 mm;
39 = range(10,50,1);
39=39*1;    : % variation from 10 to 50 m;
25 = range(5,25,5);
25=25*1;   :% variation from 5 to 25 kg/s;
21.577=21.577;
2.7767E-5=2.7767E-5;
0.01131=pi*0.12^2/4;
102.446402=25/(21.577*0.01131);
9,553,006.15531 = 102.446402*0.12*21.577/2.7767E-5;
8.647204=-1.8*log((2E-5/0.12/3.7)^1.11+6.9/9,553,006.15531);
0.003343=1/4/8.647204^2;
-492,137.180128 = -( 4*0.003343*39/0.12) * (25^2/2/21.577/0.01131^2);
```

Poisson differential equation can be applied to describe temperature distributions

- [ ] For transient cases
- [x] For steady-state cases with internal heat source
- [ ] For transient cases with variable properties
- [ ] none of the above

Dryout occurs predominantly in

- [ ] Bubbly flow regime
- [x] Annular flow regime
- [ ] Single-phase flow
- [ ] none of the above

Correlations relevant to forced convection heat transfer have usually the following form: (Nu – Nusselt Number, Re – Reynolds number, Pr – Prandtl number, Ra – Rayleigh number)

- [x] Nu = f(Re, Pr, …)
- [ ] Re = f(Pr, Ra, …)
- [ ] Nu = f(Ra, …)
- [ ] none of the above

According to the Levitan-Lantsman correlation, for two heated channels with the same mass flux and pressure but different internal diameters, dryout will first occur in the pipe with

- [ ] Greater diameter
- [ ] Smaller diameter
- [ ] Either greater or smaller diameter, depending on the mass flux
- [x] none of the above

Natural convection heat transfer is when fluid flow is

- [ ] laminar
- [x] Driven by buoyancy forces
- [ ] Driven by a pump
- [ ] none of the above

Onset of nucleate boiling (ONB) point in a heated channel is such a point where

- [x] Nucleate boiling appears
- [ ] Bulk temperature becomes equal to the local saturation temperature
- [ ] Wall heat flux is greater than the local critical heat flux
- [ ] none of the above

Wall superheat is defined as a difference between

- [ ] Bulk and wall temperature
- [ ] Bulk and saturation temperature
- [x] Wall and saturation temperature
- [ ] none of the above

Temperature distribution for steady-state conduction in an infinite hollow cylinder with constant material properties is given by

- [x] Logarithmic function
- [ ] Sine function
- [ ] Parabolic function
- [ ] none of the above

For an infinite cylinder with nuclear fission heating, the temperature at the centerline, in comparison with a case with uniform heating, is:

- [ ] Always greater
- [x] Always less
- [ ] Either greater or less, depending on the linear power
- [ ] none of the above

In post-CHF (critical heat flux) heat transfer regime, the heat transfer coefficient, in comparison to the convective boiling heat transfer regime, is:

- [ ] Much greater
- [ ] About the same
- [x] Significantly smaller
- [ ] none of the above

Newton's equation of cooling gives a relationship between

- [ ] Temperature gradient in solid and wall surface temperature
- [ ] Wall heat flux and shear stress
- [x] Wall heat flux and a temperature difference between wall surface and fluid bulk
- [ ] none of the above

Fourier law is concerned with a relationship between

- [ ] Pressure drop and mass flow rate
- [ ] Heat flux and pressure gradient
- [x] Heat flux and temperature gradient
- [ ] none of the above

For steady-state heat conduction in an infinite hollow cylinder, the heat flux on the inner surface, in comparison to the heat flux on the outer surface, is:

- [x] Always greater
- [ ] Always less
- [ ] The same
- [ ] none of the above

The lump thermal capacity model is a good approximation of exact behaviour when thermal conductivity of the body is:

- [ ] Very small
- [ ] In a range from 5 to 10 W/(m.K)
- [x] Very large
- [ ] none of the above

Temperature distribution for steady-state conduction in an infinite cylinder with constant material properties and uniform internal heat sources is given by

- [ ] Logarithmic function
- [x] Parabolic function
- [ ] Linear function
- [ ] none of the above

Inlet subcooling is defined as a difference between

- [ ] Inlet and outlet temperature
- [x] Saturation and inlet temperature
- [ ] Inlet and saturation temperature
- [ ] none of the above

For a specific solid body, with increasing heat transfer coefficient on the body surface, the corresponding Biot number (Bi):

- [x] Increases
- [ ] Decreases
- [ ] Does not change
- [ ] none of the above

The SI unit of thermal resistance is

- [ ] Watt per Kelvin
- [x] Kelvin per watt
- [ ] Kelvin per meter
- [ ] none of the above

For a pipe covered with an insulation layer with a critical thickness, the thermal losses, in comparison to uninsulated pipe, are

- [ ] greater
- [ ] Minimum possible
- [x] Maximum possible
- [ ] none of the above

Departure from Nucleate Boiling (DNB) occurs predominantly when the equilibrium thermodynamic quality is

- [x] Negative or slightly above zero
- [ ] Significantly greater than zero
- [ ] Close to one
- [ ] none of the above

198)

Water at pressure p = 2.5 MPa and average inlet temperature T_{in} = 380 K flows in a uniformly heated pipe with inner diameter 130 (mm).

Assuming laminar, convective heat transfer, calculate the inner wall surface temperature at the pipe exit, T_{wex} (K), knowing that the avearge exit water, temperature is T_{ex} = 390 K, the pipe length is 8 (m) and the water mass flow rate in the pipe is 0.04 (kg/s).

Neglect pressure changes in the pipe.

T_{wex} = 412.5291 K

```
mm = 1.e-3;
m  = 1;
MPa = 10;
Kelvin = -273.15;
```

INPUT DATA

```
p    = 2.5*MPa;    %   constant pressure
Tin  = 380+Kelvin; % constant inlet temperature
Tex  = 390+Kelvin; % constant exit temperature
di   = 100*mm;  %    changing from 100 to 150 mm
```

```
L    = 8*m;     %    changing from 3 to 8 m
W    = 0.04;    %    changing from 0.04 to 0.06 kg/s
```

SOLUTION

```
iin = XSteam('h_pT',p,Tin)*1000;
iex = XSteam('h_pT',p,Tex)*1000;
my = XSteam('my_pT',p,Tin);
tcex = XSteam('tc_pT',p,Tex); % thermal cond. at ex.
Nu = 4.364;
hex = tcex*Nu/di; % heat transfer coeff. at exit
q = (iex-iin)*W;

q2p = q/(L*pi*di); % heat flux
Twex = Tex + q2p/hex; % Found wall temp. (K)
Answer = Twex;
```

199)

Water at pressure p = 11.98 MPa and average inlet subcooling dT_{subi} = 10.3 K flows in a uniformly heated pipe with inner diameter 10 (mm).

Calculate the heat flux value q_{2p} (W/m^2) for which the exit quality will be equal to the critical quality, knowing that the water mass flux in the pipe is 2,475 (kg/m^2.s) and the pipe length is 3 (m).

Neglect pressure changes in the pipe.

Use the Levitan-Lantsman correlation for dryout.

Use steam and water saturation properties in the whole pipe.

Water-steam properties:

Saturated temperature of water-steam at 11.98 MPa is 324.55C;

Density of saturated water at 11.98 MPa is 655.5 kg/m^3;

Density of saturated steam at 11.98 MPa is 69.9 kg/m^3;

Viscosity of saturated water at 11.98 MPa is 7.67x10^{-5} Pas;

Viscosity of saturated steam at 11.98 MPa is 2.12x10^{-5} Pas;

Thermal conductivity of saturated water at 11.98 MPa is 0.497 W/(mK);

Thermal conductivity of saturated steam at 11.98 MPa is 0.091 W/(mK);

Specific heat of saturated water at 11.98 MPa is 6804.6 J/(kgK);

Specific heat of saturated steam at 11.98 MPa is 8799.2 J/(kgK);

Specific enthalpy of saturated water at 11.98 MPa is 1.4905x10^6 J/kg;

Specific enthalpy of saturated steam at 11.98 MPa is 2.6860x10^6 J/kg;

Specific enthalpy of water at 11.98 MPa and 314.25C is 1.4242x10^6 J/kg.

q_{2p} =8991422.022 W/m^2

```
mm = 1.e-3;
m  = 1;
MPa = 10;
kW_m2 = 1e3;
%
% INPUT DATA
%
```

```
p    = 11.98*MPa;   %   constant pressure
dTsubi= 10.3;       % constant inlet subcooling
di   = 10*mm;     %  changing from 8 to 12 mm
G    = 2493.3;    %  changing from 2400 to 2550 kg/m^2.s
L    = 7*m;       %  changing from 3 to 6 m
%
% SOLUTION
% Find properties
  Tsat = 324.55; % =XSteam('Tsat_p',p);
  Tin  = Tsat - dTsubi;
  IV   = 2686017; % =XSteam('hV_p',p)*1000;
  IL   = 1490517; % =XSteam('hL_p',p)*1000;
  IFG  = IV-IL;
  IIN  = 1424182; % =XSteam('h_pT',p,Tin)*1000;
% Flow area
  A = pi*di^2/4;
%
  P = p/98;
  xcr = (0.39+P*(1.57+P*(-2.04+0.68*P)))*(G/1000)^(-0.5);
  if (di ~= 0.008)
    xcr = xcr*(0.008/di)^0.15;
  endif
  IEX = IL + xcr*IFG;
  q = G*A*(IEX-IIN);
  q2p = q/(L*pi*di);

  Answer = q2p;
```

200)

Water at pressure p = 11.98 MPa and average inlet subcooling dT_{subi} = 10.3 K flows in a uniformly heated pipe with inner diameter 11 (mm).

Assuming turbulent convective heat transfer, calculate the inner wall surface temperature at the pipe exit, Twex (K), knowing that the avearge exit water temperature is T_{ex} = 596.15 K, the pipe length is 1 (m) and the water mass flux in the pipe is 2,410 (kg/m^2.s).

Neglect pressure changes in the pipe.

Use the Dittus-Boelter correlation.

T_{wex} = 609.191459 K

```
mm = 1.e-3;
m  = 1;
MPa = 10;
Kelvin = -273.15;
%
% INPUT DATA
%
p    = 11.98*MPa;   %   constant pressure
dTsubi= 10.3;       % constant inlet subcooling
Tex  = 596.15+Kelvin; % constant exit temperature
di   = 10*mm;   %    changing from 8 to 12 mm
L    = 1*m;     %    changing from 0.5 to 1.0 m
G    = 2493.3;  %    changing from 2400 to 2550 kg/m^2.s
%
% SOLUTION
%
Tsat=324.55; % =XSteam('Tsat_p',p);
Tin = Tsat-dTsubi;
rho=660.39; % =XSteam('rho_pT',p,Tex);
iin=1424182; % = XSteam('h_pT',p,Tin)*1000;
```

```
iex=1480065; % = XSteam('h_pT',p,Tex)*1000;
A = pi*di^2/4;  % Pipe flow area
U = G/rho;  % Mean flow velocity
my=7.7397e-5; % = XSteam('my_pT',p,Tex); % Dynamic viscosity
Re = rho*U*di/my;  % Reynolds number
tcex=0.50093; % = XSteam('tc_pT',p,Tex); % thermal cond. at ex.
cpex=6678.2; % = XSteam('cp_pT',p,Tex)*1000; % thermal capacity
Pr = cpex*my/tcex;  % Prandtl number
Nu = 0.023*Re**0.8*Pr**0.4;  % Dittus-Boelter correlation
hex = tcex*Nu/di; % heat transfer coeff. at exit
q = (iex-iin)*G*A; % energy balance
q2p = q/(L*pi*di); % heat flux
Twex = Tex + q2p/hex; % Found wall temp. (K)
Answer = Twex;
```

201)

A pipe with outer diameter 0.031 (m), inner diameter 0.018 (m) and length 23 (m) is made of steel with heat thermal conductivity 36 (W/m.K). Water/steam mixture flows inside the pipe and exhaust gases flow outside the pipe. The inner surface temperature of the pipe is 465 (K) and the outside surface temperature is 575 (K).

Calculate the heat flux q_{2p} (W/m^2) to water/steam mixture from the pipe wall inner surface.

q_{2p} = 809395.685 W/m^2

Find thermal resistance

$$R_{th} = \frac{\log \frac{D_{out}}{D_{in}}}{(2 \cdot \pi \cdot L \cdot H_{con})}$$

Find heat flow rate

$$q = \frac{(T_{out} - T_{in})}{R_{th}}$$

Find heat flux on the inner surface

$$q_{2p} = \frac{q}{(\pi \cdot D_{in} \cdot L)}$$

202)

A pipeline with outer wall diameter 250 (mm), wall thickness 17 (mm) and length 70 (m) is made of material with thermal conductivity 50 (W/(m.K)). The pipeline is insulated with a layer with thickness 15 (cm) and thermal conductivity 0.09 (W/(m.K)). A fluid with mean temperature 660 (K) flows inside the pipeline and heat transfer coefficient on the inside is 510 (W/(m^2.K)). The air temperature outside the pipeline is 300 (K) and the heat transfer coefficient is 13 (W/(m^2.K)).

Calculate the total thermal losses q_{loss} (W) of the pipeline.

q_{loss} = 17473.6415 W

Find thermal resistance

```
ri = d/2 - wth;                    % inner pipe wall radius
rwo= d/2;                          % outer pipe wall radius
ro = rwo + ith;                    % outer insulation radius
Rthi = 1.0/hi/ri;                  % inner resistance
Rthw = log(rwo/ri)/wthc;           % wall resistance
Rthil = log(ro/rwo)/ithc;          % insulation resistance
Rtho = 1.0/ho/ro;                  % outer resistance
Rth = (Rthi+Rthw+Rthil+Rtho)/(2*pi*L);
```

Find heat flow rate (loss)

```
q = (Tfi-Tfo)/Rth;
```

203)

A pipe with outer diameter 0.03 (m), inner diameter 0.019 (m) and length 28 (m) is made of steel with heat thermal conductivity 41 (W/m.K). Water/steam mixture flows inside the pipe and exhaust gases flow outside the pipe. The inner surface temperature of the pipe is 495 (K) and the outside surface temperature is 575 (K).

Calculate the heat flux q_{2p} (W/m^2) from exhaust gases to the pipe wall outer surface.

q_{2p} = 478735.947 W/m^2

Find thermal resistance

$$R_{th} = \frac{\log \frac{D_{out}}{D_{in}}}{(2 \cdot \pi \cdot L \cdot H_{con})}$$

Find heat flow rate

$$q = \frac{(T_{out} - T_{in})}{R_{th}}$$

Find heat flux on the outer surface

$$q_{2p} = \frac{q}{(\pi \cdot D_{out} \cdot L)}$$

204)

Water at pressure p = 11.98 MPa and average inlet subcooling dT_{subi} = 10.3 K flows in a uniformly heated pipe with inner diameter 9 (mm).

Calculate the wall temperature at inner surface T_w (K) at distance 4 (m) from the inlet, knowing that the heat flux is 360 (kW/m^2) and the water mass flux in the pipe is 2,475 (kg/m^2.s).

Neglect pressure changes in the pipe.

Use the Chen correlation for convective boiling heat transfer. Use the following approximation to calculate saturated pressure ps (bar) as a function of the wall temperature Tw (C): ps = 113.37 + 1.5145*(Tw - 320.36)

HINT: iterate wall temperature until convergence.

Tw = 603.78464 K

```
mm = 1.e-3;
m  = 1;
MPa = 10;
kW_m2 = 1e3;
%
% INPUT DATA
%
p    = 11.98*MPa;   %   constant pressure
dTsubi= 10.3;       % constant inlet subcooling
di   = 10*mm;     %  changing from 8 to 12 mm
q2p  = 348*kW_m2; %  changing from 330 to 360 kW/m^2
G    = 2493.3;    %  changing from 2400 to 2550 kg/m^2.s
L    = 5*m;       %  changing from 3 to 6 m
%
% SOLUTION
% Find properties
  Tsat = 3.245506605846990e+02;  %XSteam('Tsat_p',p);
  Tin  = Tsat - dTsubi;
  RHOL = 6.555002172822931e+02;  %XSteam('rhoL_p',p);
  RHOV = 69.925776024162350;     %XSteam('rhoV_p',p);
  VISL = 7.666935902283314e-05;  %XSteam('my_pT',p,Tsat-0.01);
  VISV = 2.117364057284995e-05;  %XSteam('my_pT',p,Tsat+0.01);
  CPL  = 6.804592996635221e+03;  %XSteam('cpL_p',p)*1000;
  CONL = 0.496816348037341;      %XSteam('tcL_p',p);
  CONV = 0.090794755477667;      %XSteam('tcV_p',p);
  SIG  = 0.008871693100245;      %XSteam('st_p',p);
  IV   = 2.686016748218022e+06;  %XSteam('hV_p',p)*1000;
  IL   = 1.490516858836635e+06;  %XSteam('hL_p',p)*1000;
  IFG  = IV-IL;
  IIN  = 1.424181972369010e+06;  %XSteam('h_pT',p,Tin)*1000;
% Energy balance
  A = pi*di^2/4;
  ILOC = IIN + q2p*L*pi*di/G/A;
  x = (ILOC-IL)/IFG;
%
  DTguess = 5;
  eps = 100;
  Xtt = ((1-x)/x)^0.9*(RHOV/RHOL)^0.5*(VISL/VISV)^0.1;
  Rel = G*(1-x)*di/VISL;
  if 1/Xtt <= 0.1
    F = 1;
  else
    F = 2.35*(0.213+1/Xtt)^0.736;
  end
  S = 1/(1+2.56e-6*F^1.463*Rel^1.17);
  Prl = CPL*VISL/CONL;
  hmac = 0.023*CONL*Rel^0.8*Prl^0.4*F/di;
  iter = 1;
  ConstHmic = 0.00122*CONL^0.79*CPL^0.45*RHOL^0.49*S/SIG^0.5/VISL^0.29/IFG^0.24/RHOV^0.24;
  while iter <=100 & eps>=0.01
    DTsup = DTguess;
    Tw   = Tsat + DTsup;
%    PsTw = XSteam('psat_T',Tw);  % exact
    PsTw = 113.37 + 1.5145*(Tw - 320.36); % Approximation
    hmic = ConstHmic*DTsup^(0.24)*((PsTw-p)*1e5)^(0.75);
    htc = hmic + hmac;
```

```
    DTnew = q2p/htc;
    DTguess = 0.85*DTnew + 0.15*DTsup;
    eps = abs(DTnew-DTsup);
    iter= iter+1;
  end
  Answer = Tsat + DTguess + 273.15;  % wall temperature (K)
```

205)

Water at pressure p = 11.98 MPa and average inlet subcooling dT_{subi} = 10.3 K flows in a uniformly heated pipe with inner diameter 9 (mm).

Calculate the distance from the inlet to the ONB point, zONB (m) knowing that the heat flux is 340 (kW/m^2) and the water mass flux in the pipe is 2,500 (kg/m^2.s).

Neglect pressure changes in the pipe and, use all water properties as at inlet.

Use the Dittus-Boelter and the Thom et al. correlations.

Water-steam properties:

Saturated temperature of water-steam at 11.98 MPa is 324.55C;

Density of water at 11.98 MPa and 314.25C is 685.38 kg/m^3;

Viscosity of water at 11.98 MPa and 314.25C is 8.13x10^{-5} Pas;

Thermal conductivity of water at 11.98 MPa and 314.25C is 0.52 W/(mK);

Specific heat of water at 11.98 MPa and 314.25C is 6136 J/(kgK).

zONB = 0.217468 m

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206)

Water at pressure p = 11.98 MPa and average inlet subcooling dT_{subi} = 10.3 K flows in a uniformly heated pipe with inner diameter 10 (mm).

Calculate the critical heat flux value q_{2pcr} (W/m^2) at distance 4 (m) from the inlet knowing that the water mass flux in the pipe is 2,550 (kg/m^2.s) and the heat flux is 370 (kW/m^2).

Neglect pressure changes in the pipe.

Use the Levitan-Lantsman correlation for DNB.

Use steam and water saturation properties in the whole pipe.

Water-steam properties:

Saturated temperature of water-steam at 11.98 MPa is 324.55C;

Density of saturated water at 11.98 MPa is 655.5 kg/m^3;

Density of saturated steam at 11.98 MPa is 69.9 kg/m^3;

Viscosity of saturated water at 11.98 MPa is 7.67x10^{-5} Pas;

Viscosity of saturated steam at 11.98 MPa is 2.12x10^{-5} Pas;

Thermal conductivity of saturated water at 11.98 MPa is 0.497 W/(mK);

Thermal conductivity of saturated steam at 11.98 MPa is 0.091 W/(mK);

Specific heat of saturated water at 11.98 MPa is 6804.6 J/(kgK);

Specific heat of saturated steam at 11.98 MPa is 8799.2 J/(kgK);

Specific enthalpy of saturated water at 11.98 MPa is 1.4905x10^6 J/kg;

Specific enthalpy of saturated steam at 11.98 MPa is 2.6860x10^6 J/kg;

Specific enthalpy of water at 11.98 MPa and 314.25C is 1.4242x10^6 J/kg.

q_{2pcr} = 2088211.43 W/m^2

```
mm = 1.e-3;
m  = 1;
MPa = 10;
kW_m2 = 1e3;
%
% INPUT DATA
%
p    = 11.98*MPa;   %   constant pressure
dTsubi= 10.3;       % constant inlet subcooling
di   = 10*mm;     %  changing from 7 to 12 mm
G    = 2493.3;    %  changing from 2400 to 2550 kg/m^2.s
L    = 3*m;       %  changing from 1 to 4 m
q2p  = 350*kW_m2; %  changing from 320 to 370 kW/m^2
%
% SOLUTION
% Find properties
  Tsat = 324.55; % =XSteam('Tsat_p',p);
  Tin  = Tsat - dTsubi;
  IV   = 2686017; % =XSteam('hV_p',p)*1000;
  IL   = 1490517; % =XSteam('hL_p',p)*1000;
  IFG  = IV-IL;
  IIN  = 1424182; % =XSteam('h_pT',p,Tin)*1000;
% Energy balance
  A = pi*di^2/4;
  ILOC = IIN + q2p*L*pi*di/G/A;
  x = (ILOC-IL)/IFG;
%
  P = p/98;
  ex = 1.2*((P-1)/4.-x);
  q2pcr = (10.3+P*(-7.8+1.6*P))*(G/1000)^(ex)*exp(-1.5*x);
  if (di ~= 0.008)
    q2pcr = q2pcr*sqrt(0.008/di);
  endif
  Answer = q2pcr*1e6;
```

207)

A plane wall with thickness 2.5 (m) and area 13 (m^2) is made of material with heat thermal conductivity 1.6 (W/m.K). The air on one side of the wall has temperature 300 (K) and heat transfer coefficient from the air to the wall surface is 3 (W/m^2 K). On the other side of the wall the air temperature is 250(K) and the heat transfer coefficient is 19 (W/m^2 K).

Calculate the wall surface temperature T_{s1}(K) on side 1 (facing air with temperature T_1).

T_{s1} = 291.44 K

Find thermal resistance

$$R_{th} = \frac{(\frac{1.0}{h_1} + \frac{L}{Hcon} + \frac{1.0}{h_2})}{A}$$

Find heat flow rate

$$q = \frac{(T_1 - T_2)}{R_{th}}$$

Find heat flux

$$q_{2p} = \frac{q}{A}$$

Wall surface temperature on side 1

$$T_{s1} = T_1 - \frac{q_{2p}}{h1}$$

208)

A plane wall with thickness 27 (m) and area 20 (m^2) is made of material with heat thermal conductivity 1.2 (W/m.K). The air on one side of the wall has temperature 290 (K) and heat transfer coefficient from the air to the wall surface is 4 (W/m^2 K). On the other side of the wall there is an insulation layer with thickness 0.2 (m) made of styrofoam with thermal conductivity 0.02 (W/m.K). The air temperature outside the insulated wall is 270(K) and the heat transfer coefficient is 20 (W/m^2 K).

Calculate the temperature T_{cs} (K) of the contact surface between the wall and the insulation.

T_{cs} = 276.057 K

Find thermal resistance

$$R_{th} = \frac{(\frac{1.0}{h_1} + \frac{L}{H_{con}} + \frac{L_{ins}}{H_{cins}} + \frac{1.0}{h_2})}{A}$$

Find heat flow rate

$$q = \frac{(T_1 - T_2)}{R_{th}}$$

Find heat flux

$$q_{2p} = \frac{q}{A}$$

Find the temperature at the contact surface

$$T_{cs} = T_1 - \frac{q_{2p}}{h_1} - q_{2p} * \frac{L}{H_{con}}$$

209)

A copper pipe with outer diameter 7 (mm) and wall thickness 1.5 (mm) contains flowing hot water with temperature 360 (K). The pipe is insulated with material that has thermal conductivity 0.08 W/(m.K). The inner heat transfer coefficient is 505 W/(m^2.K) and the outer heat transfer coefficient (the same for uninsulated and insulated pipe) is 3 W/(m^2.K).

Ambient temperature is 270 (K).

Calculate the change of heat loss from the pipe per unit length q (W/m) when an uninsulated copper pipe is covered with an insulation layer that has the critical thickness.

Note that this change should be negative if the loss increases for insulated pipe ($q_{changeofheatloss}$qchangeofheatloss = $q_{uninsulated}$quninsulated - $q_{insulated}$qinsulated).

Copper thermal conductivity is 395 W/(m.K).

$q_{changeofheatloss}$qchangeofheatloss =

210)

A pipe with outer diameter 0.034 (m), inner diameter 0.024 (m) and length 12 (m) is made of steel with heat conductivity 45 (W/m.K). Water/steam mixture flows inside the pipe and exhaust gases flow on outside the pipe. The inner surface temperature of the pipe is 495 (K) and the outside wall surface temperature is 520 (K).

Calculate the total heat flow rate q (W) transferred from exhaust gases to the water/steam mixture.

q = 243529.6336 W

Find thermal resistance

$$R_{th} = \frac{\log \frac{D_{out}}{D_{in}}}{(2 \cdot \pi \cdot L \cdot H_{con})}$$

Find heat flow rate

$$q = \frac{(T_{out} - T_{in})}{R_{th}}$$

211)

A plane wall with thickness 2.6 (m) and area 16 (m^2) is made of material with heat thermal conductivity 1 (W/m.K). The air on one side of the wall has temperature 295 (K) and heat transfer coefficient from the air to the wall surface is 3 (W/m^2 K). On the other side of the wall there is an insulation layer with thickness 0.15 (m) made of styrofoam with thermal conductivity 0.04 (W/m.K). The air temperature outside the insulated wall is 265(K) and the heat transfer coefficient is 14 (W/m^2 K).

Calculate the rate of heat q (W) transfered through the insulated wall from side (1) to side (2).

q = 71.06098 W

Find thermal resistance

$$R_{th} = \frac{\left(\frac{1.0}{h_1} + \frac{L}{H_{con}} + \frac{L_{ins}}{H_{cins}} + \frac{1.0}{h_2}\right)}{A}$$

Find heat flow rate

$$q = \frac{(T_1 - T_2)}{R_{th}}$$

212)

A plane wall with thickness 1 (m) and area 11 (m^2) is made of material with heat thermal conductivity 1 (W/m.K). The air on one side of the wall has temperature 300 (K) and heat transfer coefficient from the air to the wall surface is 10 (W/m^2 K). On the other side of the wall the air temperature is 280(K) and the heat transfer coefficient is 15 (W/m^2 K).

Calculate the rate of heat q (W) transfered through the wall from side (1) to side (2).

q = 188.68 W

Find thermal resistance

$$R_{th} = \frac{(\frac{1.0}{h_1} + \frac{L}{H_{con}} + \frac{1.0}{h_2})}{A}$$

Find heat flow rate

$$q = \frac{(T_1 - T_2)}{R_{th}}$$