FME HighEFF

Centre for an Energy Efficient and Competitive Industry for the Future

Deliverable D2.3_2019.06

Humid gases

Delivery date: 2019-12-20

Organisation name of lead beneficiary for this deliverable:

SINTEF Energy Research

HighEFF- Centre for an Energy Efficient and Competitive Industry for the Future is one of Norway's Centre for Environment-friendly Energy Research (FME). Project co-funded by the Research Council of Norway and Industry partners. Host institution is SINTEF Energi AS.		
Dissemination Level		
PU	Public	x
RE	Restricted to a group specified by the consortium	

Deliverable number:	D2.3_2019.06
ISBN number:	
Deliverable title:	Thermodynamics of humid gases
Work package:	RA 2.3
Deliverable type:	Memo
Lead participant:	SINTEF Energy Research

Quality Assurance, status of deliverable		
Action	Performed by	Date
Verified (WP leader)	Geir Skaugen	2019-12-20
Reviewed (RA leader)	Armin Hafner	2019-12-31
Approved (dependent on nature of deliverable)*)		

*) *The quality assurance and approval of HighEFF deliverables and publications have to follow the established procedure. The procedure can be found in the HighEFF eRoom in the folder "Administrative > Procedures".*

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Abstract

The work on humid gas in 2019 includes the modelling of two heat transfer scenarios with using the thermodynamic model for humid gas implemented in 2018, as reported in D2.3_2018.03. The first scenario is to simulate the heat transfer of humid gas on a cold flat plate, where the calculation of heat transfer is based on the traditional method for dehumidification process. The second scenario is to simulate a tube-in-tube heat exchanger with humid gas flowing in the inner tube and cooling water flowing in the annulus. The method treats the humid gas as normal mixtures and uses the thermodynamic libraries for mixtures.

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1 Introduction

The work on humid gas in 2019 includes the modelling of two heat transfer scenarios with using the thermodynamic model for humid gas implemented in 2018, as reported in D2.3_2018.03. The first scenario is to simulate the heat transfer of humid gas on a cold flat plate, where the calculation of heat transfer is based on the traditional method for dehumidification process. The second scenario is to simulate a tube-in-tube heat exchanger with humid gas flowing in the inner tube and cooling water flowing in the annulus. The method treats the humid gas as normal mixtures and uses the thermodynamic libraries for mixtures. The two problems are described in Section 2 and Section 3 with the modelling methods and results, respectively.

2 Humid gas cooling on a flat plate

In the first model, the problem is simplified as humid gas flowing on a flat surface with constant and uniform temperature, as illustrated by Figure 1.

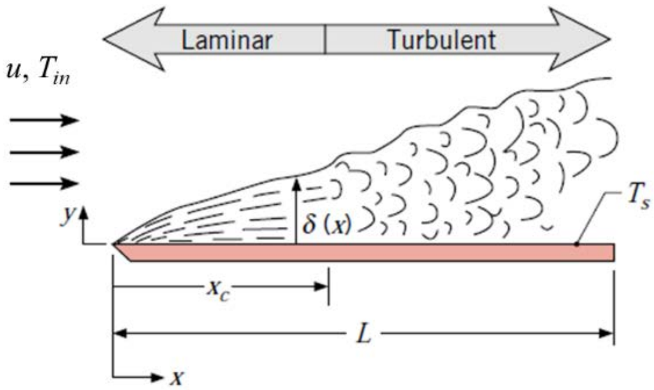


Figure 1 Fluid flows on a flat plate [1]

Table 1 Geometries and inlet conditions

Plate length, *L*	6 m
Plate width, *d*	0.06 m
Surface temperature, T_s	5°C
Free stream velocity, *u*	10 m/s
Inlet stream temperature, T_{in}	85°C

2.1 Modelling and calculation procedures

2.2 Discretization and conservation equations

The problem will be solved with a one-dimensional model. The discretization along the length of the plate is illustrated in Figure 2.

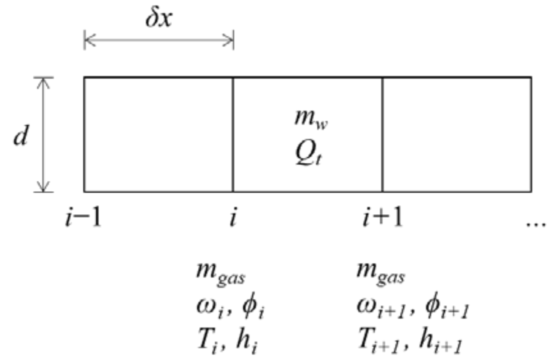


Figure 2 Discretization of the domain

The mass conservation for water (liquid) and energy conservation for humid gas at inlet of each element can be applied to a single element as:

$$m_{dgas,i}\omega_i = m_{dgas,i+1}\omega_{i+1} + m_w \tag{1}$$

$$m_{dgas,i}h_i = m_{dgas,i+1}h_{i+1} + m_w h_w + Q_t \tag{2}$$

where m_{dgas} is the mass flow rate of the dry gas excluding vapor, m_w is the produced liquid (almost water) in each element due to cooling of the humid gas, h_w is the specific enthalpy of the produced liquid, and Q_t is the total heat released during the cooling and dehumidification. The rest variables have been introduced in the previous memo. It is worth noting that the method is actually different from the normal control volume method, which usually applies the conservation equations on the total mass at the inlet ($x = 0$). The mass conversation equation here is based on the balance for the water (or the liquid with very small amount of other components) in each element. This is equivalent to the conversation for the humid gas at inlet of each element ($m_{hgas,i} = m_{dgas}(1 + \omega_i)$), namely the liquid produced in one element is removed from the considered system in the next following element. Another problem that should be noted is that, the mass flow rate of dry gas m_{dgas} could be treated as constant since the produced liquid will be almost pure water. However, if we want to obtain the real composition of the produced liquid, the calculation of the mass flow rate of dry gas should be considered. In the following, m_{dgas} is assumed as constant.

The total heat is made of the sensible heat $\boldsymbol{Q_s}$ and the latent heat $\boldsymbol{Q_l}$ as,

$$Q_t = Q_s + Q_l \tag{3}$$

$$Q_s = m_{gas}c_p(T_i - T_{i+1}) \tag{4}$$

$$Q_l = m_{gas}(\omega_i - \omega_{i+1})h_{fg,w} \tag{5}$$

where $h_{fg,w}$ is the latent heat of the liquid. The specific heat c_p defined in the previous memo is respect to the humid gas with the unit kJ/(kg of dry gas · K).

The total heat also satisfies the relationship that,

$$Q_t = \alpha_t(T_f - T_s)\delta A \tag{6}$$

where α_t is the total heat transfer coefficient (HTC), also known as apparent HTC, T_f is the average fluid temperature in each element, which can be estimated as $(T_i + T_{i+1})/2$, and the area of each element $\delta A = \delta x \cdot d$.

2.3 Cooling and dehumidification process

A cooling and dehumidification process of humid gas in contact with a cooling coil is shown in Figure 3 (a), and it is modified and integrated to our present case Figure 3(b). Here, we first introduce several useful parameters. The *sensible heat factor* (*SHF*) is defined as the ratio of the sensible heat to the total heat, namely Q_s/Q_t. The temperature T_s is the effective surface temperature, also known as *apparatus dew-point* (*ADP*) temperature. A *bypass factor* (*BPF* or *BF*) is sometimes used to express cooling coil efficiency, defined as,

$$BPF = \frac{T_{in} - T_s}{T_{out} - T_s} = \frac{\omega_{in} - \omega_s}{\omega_{out} - \omega_s} \tag{7}$$

where the parameters with subscripts 'out' are corresponding to the status at outlet $x = L$, marked in the left figure. Ideally, the gas is in perfect contact with cooling coil surface, and the outlet temperature T_{out} will be the same as T_s. In fact, T_{out} will be always higher than T_s. We will apply this model to a single element described above, and rewrite the *BPF* as,

$$BPF = \frac{T_i - T_s}{T_{i+1} - T_s} = \frac{\omega_i - \omega_s}{\omega_{i+1} - \omega_s} \tag{8}$$

and the corresponding parameters have been marked in Figure 3(b).

The enthalpy differences are defined as,

$$\Delta h_{tot} = f_h(\omega_i, T_i) - f_h(\omega_s, T_s) \tag{9}$$

$$\Delta h_{dry} = f_h(\omega_i, T_i) - f_h(\omega_i, T_s) \tag{10}$$

with f_h the enthalpy function in terms of temperature and humidity ratio.

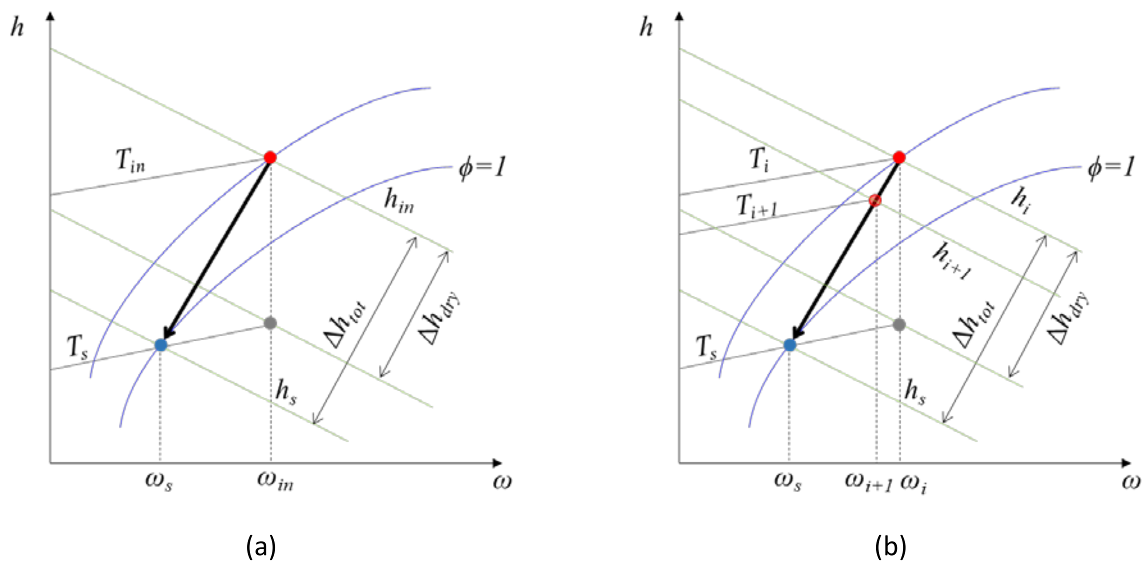


Figure 3 Cooling and dehumidification process (a) general case, (b) integrated to current model

2.4 Heat transfer models

The apparent HTC for the cooling of humid gas (Figure 3(a)) is derived based on the relationship between mass transfer and heat transfer [2], as,

$$\alpha_t = \alpha_{dry} \frac{\Delta h_{tot}}{\Delta h_{dry}} \tag{11}$$

where α_{dry} related to the sensible heat transfer is often modelled with common convective HTC. Here, the Δh_{tot} and Δh_{dry} are corresponding to the ones in Figure 3(a). The differences between using (a) or (b) will be detailed later.

We also apply this relationship to the current model, with α_{dry} estimated with the local heat transfer coefficient for single phase fluid flowing on a constant and uniform flat surface, as shown in Figure 1. The local Nusselt number can be calculated as [1],

$$Nu_x = \frac{\alpha_{dry,x} x}{k} = \begin{cases} 0.332 Re_x^{1/2} Pr^{1/3} & Re_x \le Re_{x,c} \\ 0.0296 Re_x^{4/5} Pr^{1/3} & Re_x > Re_{x,c} \end{cases} \tag{12}$$

where the local Reynolds number $Re_x = \rho u x / \mu$, and $Re_{x,c}$ is the critical Reynolds number to determine the beginning of flow transition from laminar to transition and turbulent flow regimes, and k is the thermal conductivity. For flow over a flat plate, $Re_{x,c}$ varies depending on the surface roughness and the turbulence level of the free stream, while 5×10^5 is often used. This equation is valid for $Pr \ge 0.6$. The location *x* is used the average between two nodes as $(x_i + x_{i+1})/2$.

2.5 Thermodynamic properties

The thermodynamic properties for humid gas are calculated with considering properties of dry gas and vapor separately. The dry gas part is calculated with real EOS. The functions for transport properties have been implemented for dry gas, which are required for Reynolds number and Nusselt number. The calculation of viscosity and thermal conductivity uses "TRAPP" model. The latent heat of water $h_{fg,w}$ is set as a constant value of 2502 kJ/kg, which will be extend to general liquid in the next step.

2.6 Calculation procedures

Based on the above sub-models, the problem can be solved with an iterative manner. The calculation procedures are illustrated as follows, with the parameters at node *i* known from previous. The produced liquid is assumed as pure water for now, thus the mass flow rate of dry gas is constant.

1) Initial guess T_{i+1}
2) Calculate ω_{i+1} based on Eq. (8), then obtain m_w from Eq. (1)
3) Update properties $c_{p,i+1}, h_{i+1}$ (humid gas based), and $k_{i+1}, \mu_{i+1}, \rho_{i+1}$ (dry gas based)
4) Calculate Q_t, Q_s, Q_l from Eq. (3)-(5)
5) Calculate Δh_{tot} and Δh_{dry} based on Eq. (9)(10)
6) Calculate $\alpha_{dry,x}$ from Eq. (12), and α_t from Eq. (11)
7) Check Q_t' obtained from Eq. (6)
8) Compare Q_t and Q_t'
    a. If the relative difference is smaller than 0.01, the iteration for this element is done, and set the parameters at node $i+1$ as the known node of the next following element.
    b. Otherwise, update T_{i+1} and go back to step 2)

The reason to iterate on temperature is that the function for calculating the enthalpy from temperature is available, but not for calculating temperature from enthalpy. In the second scenario, the calculation process is based the enthalpy, other than the temperature.

2.7 Results

Here, humid gases with two different vapor fractions are tested:

- Humid gas 1: N_2 /O_2 /H_2O, 0.74/0.20/0.06 by mole fraction

- Humid gas 2: N_2 /O_2 /H_2O, 0.472/0.128/0.4 by mole fraction

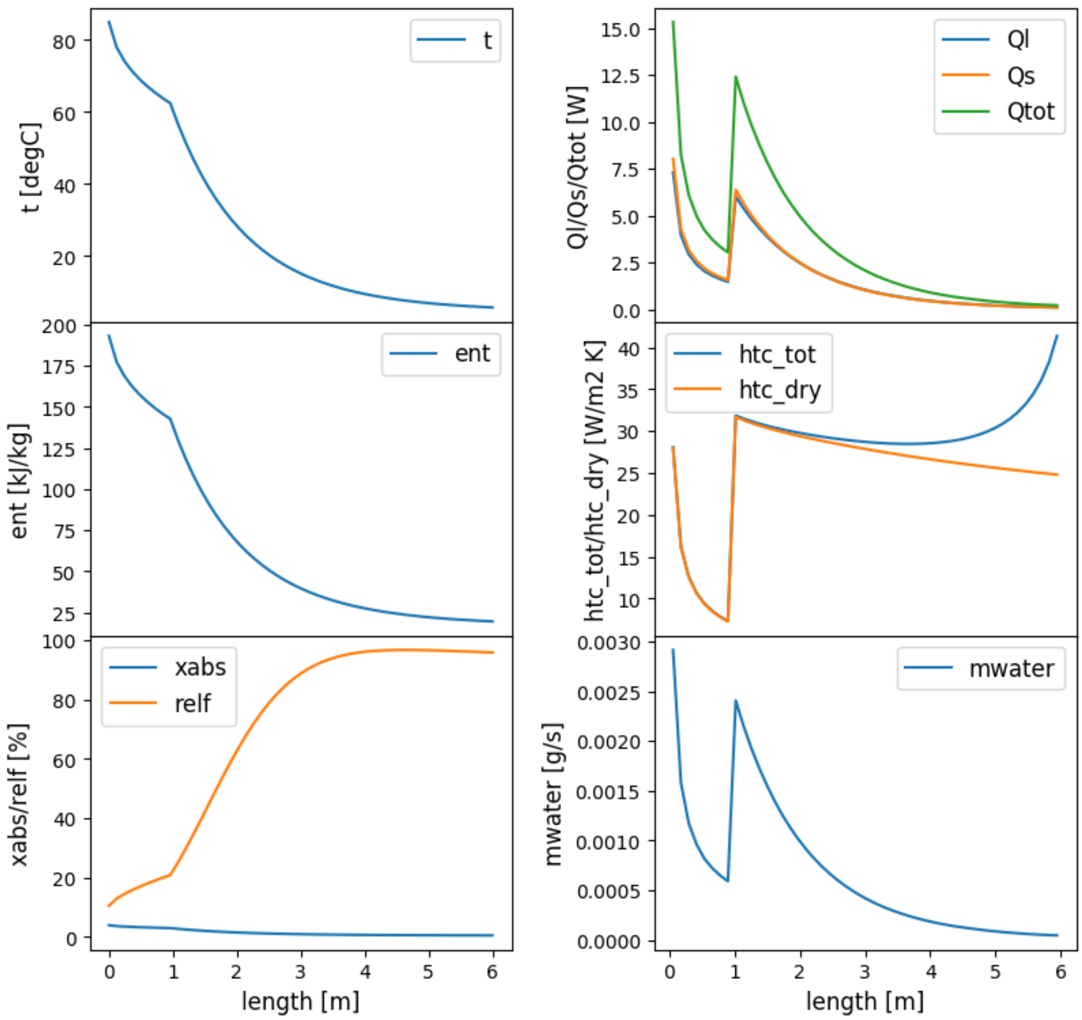


Figure 4 Results for humid gas 1, parameters against plate length

The discontinuity in heat transfer coefficients is caused by the transition of the flow.

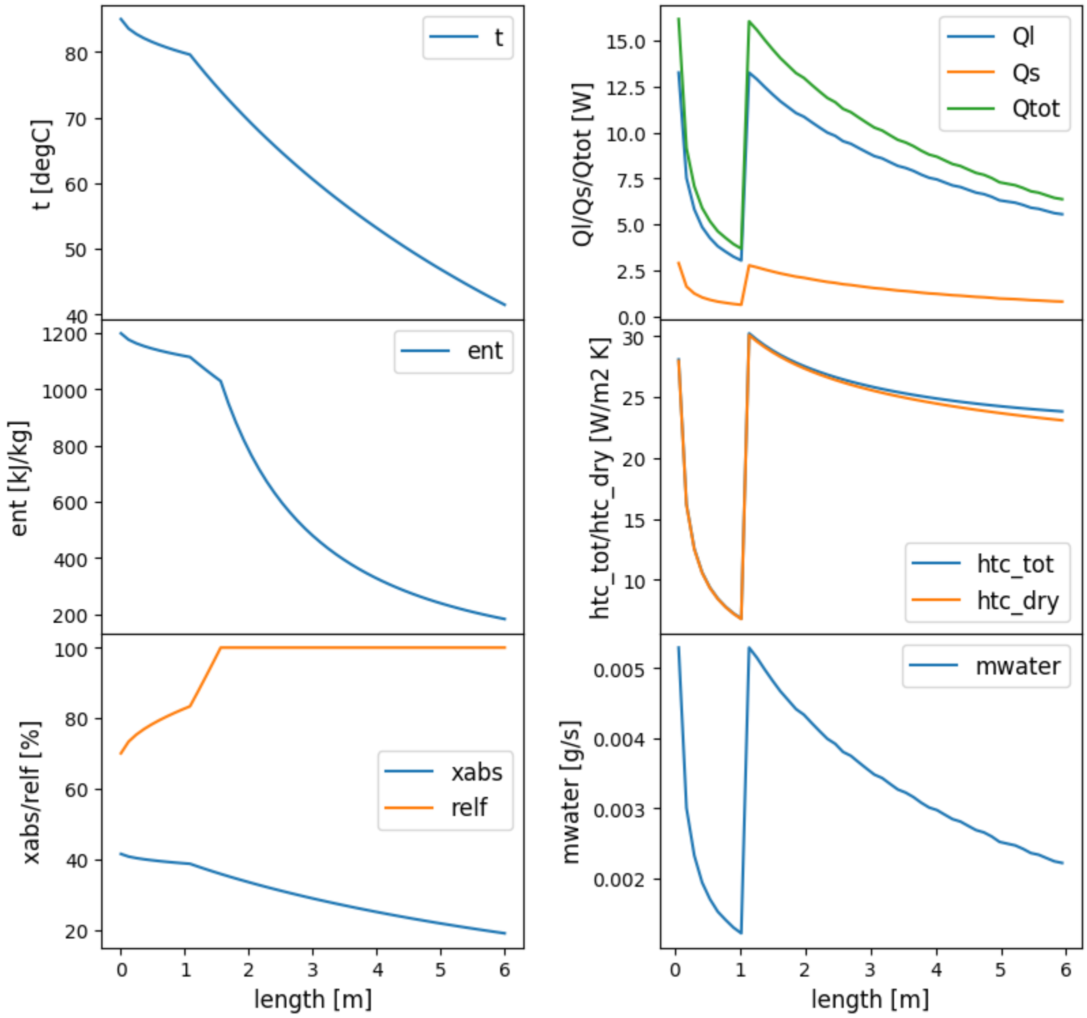


Figure 5 Results for humid gas 2, parameters against plate length

3 Humid gas cooling in a tube-in-tube heat exchanger

3.1 Discretization and conservation equations

The model treats the humid gas as general mixtures and solves the conservation equation of mass, momentum, and energy for the humid gas. The wall resistance is neglected here.

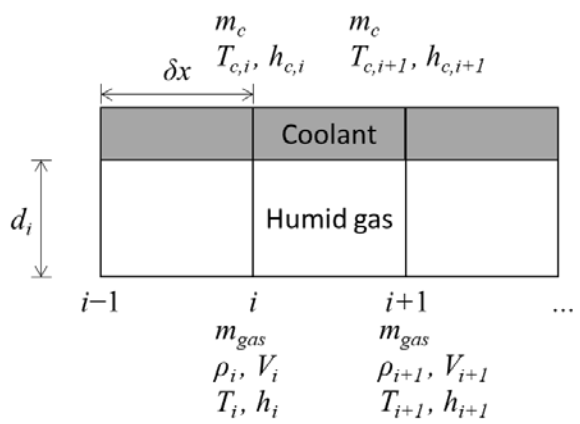


Figure 6 Discretization of the domain

$$\rho_i V_i - \rho_{i+1} V_{i+1} = 0 \tag{13}$$

$$\rho_i V_i (V_{i+1} - V_i) = -(P_{i+1} - P_i) - 4\tau_w \cdot \delta x / d_i \tag{14}$$

$$m_{gas} h_{i+1} - m_{gas} h_i = -Q_t \tag{15}$$

Here m_{gas} is constant, which is equivalent to Eq. (13). For the cooling water, only the energy equation is solved for simplicity.

$$m_c h_{c,i+1} - m_c h_{c,i} = Q_t \tag{16}$$

where the m_c is the mass flow rate of coolant, and h_c is the enthalpy of the coolant.
The heat balance satisfies:

$$Q_t = \alpha_f (T_f - T_w) \pi d_i \cdot \delta x = \alpha_c (T_w - T_c) \pi d_i \cdot \delta x \tag{17}$$

where α_f, α_c are the HTC for the bulk fluid and coolant (cooling water).

The correlation by Friedel[3] is used for the calculation of the two-phase friction factor, and the correlation by Thome [4] is used for the calculation of the two-phase heat transfer of mixtures using the method of Silver, Bell and Ghaly [5] for mixtures and the standard Dittus-Boelter equation for the single phase terms. The thermodynamic library used is TREND with EOS GERG2008[6].

3.2 Calculation procedures

The calculation procedures are as below:

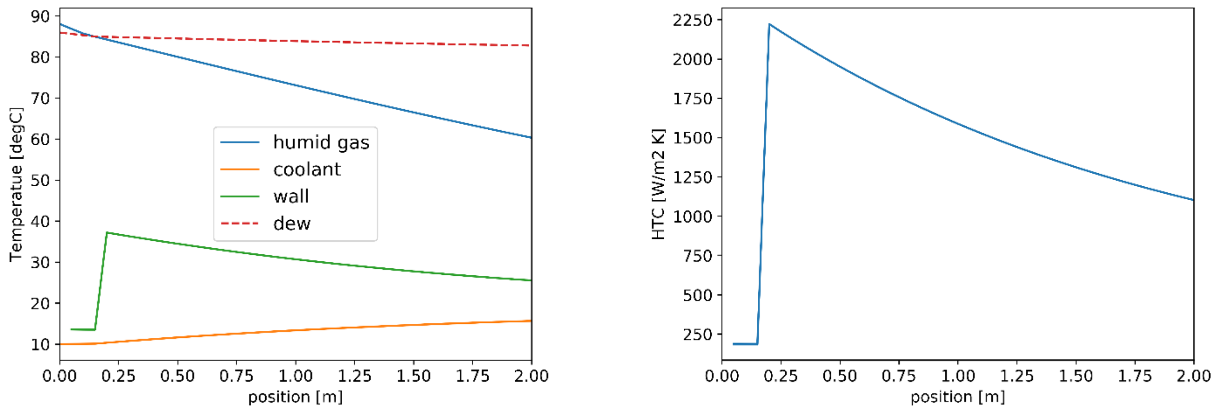
1) Initial guess T_{w+1}, and initialize parameter for node (*i*+1) with the values from node *i*
2) Solve equations for the humid gas by iterations
    a. Calculate properties and temperature based on (P_{i+1}, h_{i+1})
    b. Calculate source terms τ_w, Q_t based on correlations
    c. Update (P_{i+1}, h_{i+1}) based on Eq. (13)-(15)

   d. Finish when the relative difference of h_{i+1} in two iterations is smaller than 1E-5
3) Calculate Q_t' on the coolant side, solve energy equation for coolant Eq. (16)
4) Compare Q_t and Q_t'
   a. If the relative difference is smaller than 1E-3, finish and move to next element
   b. Otherwise, update T_{w+1} based on Eq. (17) with average heat $(Q_t + Q_t')/2$

3.3 Results

A gas of N_2 /O_2 /H_2O, 0.472/0.328/0.2 by mole fraction is used as an example. The inlet pressure is set as 3 bar and the inlet temperature is 88 °C. The diameter of the inner and outer tube are 30 mm and 40 mm, and the length is 2 m. The mass flow rate of the humid gas is 0.05 kg/s while the cooling water is 0.5 kg/s.

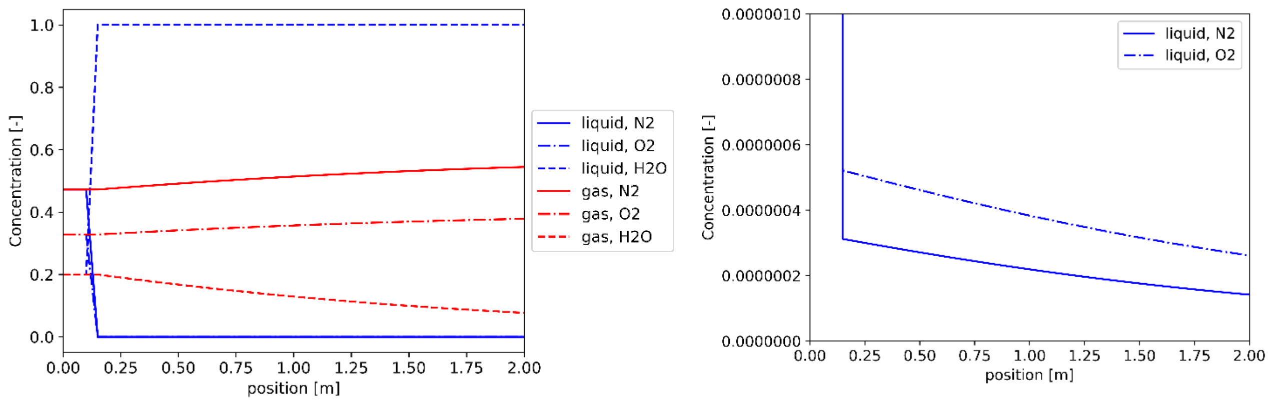
The results are shown in Figure 7. At the inlet, the humid gas is in superheated gas phase. When the temperature reduces and reaches the dew temperature, the flow goes into two phases, with a significant increase in the HTC. The sharp change of the wall temperature is the results without considering the axial heat transfer.



(a) Temperature profiles (b) Heat transfer coefficient of humid gas

Figure 7 temperatures and HTC along the tube

Figure 8 shows the variation of the compositions in the liquid and vapor phase. The produced liquid is almost water, with very small amount of other minor components based on the current EOS.



(a) Composition of vapor and liquid (b) Small component fractions in the liquid film, 2-6 ppm

Figure 8 Variation of composition along the tube

4 Conclusions and future work

In the present work, two heat transfer scenarios for humid gas has been modelled, one is based on the traditional method for dehumidification process and the other is for general mixtures.

It requires further validation of the results based on experimental data, which is not available at present. Regarding the future work on modelling, it is suggested to find a suitable case relevant to the practical applications, such as the flue gases containing Sulphur components. The differences between using the dehumidification method and method of general mixtures could be further investigated.

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