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The evaporation of water into air of different humidities and the inversion temperature phenomenon

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Abstract

A theoretical study of the evaporation of water into its own vapour, air and a mixture thereof is presented. The equations describing the coupled heat and mass transfer are set up and solved to find the evaporation rates. The calculated results are in good agreement with published experimental data. The inversion temperature, above which the evaporation rate into pure superheated steam is higher than into dry air, is calculated for constant mass and volumetric flows and compared with published data. Refined definitions of the inversion temperature are introduced which enable a more precise description of the phenomenon and explain the variation in published data. © 2000 Elsevier Science Ltd. All rights reserved.

Keywords: Inversion temperature; Superheated steam; Drying; Evaporation

1. Introduction

The evaporation of a liquid into a gas flow consisting of its own vapour, air or a mixture thereof is of importance in many chemical engineering processes. Drying is one of these applications. At very high moisture contents of the material to be dried the drying process is mainly governed by unhindered evaporation, which makes the understanding of evaporation a prerequisite for the design and optimisation of the drying process. In most cases the drying medium is a mixture of air and the vapour of a pure evaporating liquid. This significantly simplifies the problem, because the gas can be considered as a binary mixture. In this paper, the equations for coupled heat and mass transfer describing this process are set up and solved for a range of gas temperatures and compositions.

The calculations are carried out at atmospheric pressure using air as the inert gas and water as the evaporating liquid, which represents a common case in drying. The results of these calculations are the evaporation rates. With the knowledge of these, the so-called inversion temperature can be determined. The inversion temperature has most commonly been described as the gas temperature at which the evaporation rates into completely dry air and pure superheated steam are equal. This definition implicitly assumes the amount of evaporation to be so small that no change in gas temperature and composition is incurred by the process. Therefore, it

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Nomenclature

A	area of interface (m^2)
c_p	molar heat capacity (J/mol/K)
c_t	molar density (mol/m^3)
d	diameter of the wetted wall column (m)
D	diffusion coefficient (m/s)
h	molar enthalpy (J/mol)
l	length from the beginning of the plate (m)
$\dot{n}$	molar flux ($mol/m^2/s$)
$\dot{N}$	molar flow (mol/s)
Nu	Nusselt number
p_{sat}	saturation pressure (Pa)
Pr	Prandtl number
$\dot{q}$	heat flux (W/m^2)
Re	Reynolds number
Sc	Schmidt number
Sh	Sherwood number
T	temperature (K)
y	mole fraction (mol/mol)

Greek symbols

α	heat transfer coefficient ($W/m^2/K$)
β	mass transfer coefficient (m/s)
Δh_v	latent heat of vaporisation (J/mol)
Φ_W	Ackermann correction factor

Subscripts

a	air
conv	convection only
evap	evaporated
G	bulk flow gaseous phase
i	component i
I	interface
in	at inlet
L	liquid
out	at outlet
total	total (convection and radiation)
v	vapour, vaporisation

only applies to evaporation from infinitely small surface areas into completely dry air or pure superheated steam. Due to these limitations, new and more precise definitions for the inversion temperature are introduced in this paper.

First, the temperature at which the evaporation rates from an infinitely small surface area into the two gas flows with vapour mole fractions a and b is equal, is defined as the local inversion temperature $\text{LIT}_{a,b}$. In every real process and experiment the amount of evaporation is not infinitely small and the changes in gas condition cannot be neglected. Therefore, the introduction of the apparent inversion temperature is suggested here for these applications. This apparent inversion temperature is the gas temperature at the beginning of the evaporation area for which the average evaporation rates into the two gas flows with the inlet vapour mole fractions a and b is equal, thus giving $\text{AIT}_{a,b}$. Any apparent inversion temperature has to be accompanied by detailed information about the geometry and gas flow conditions investigated to make an evaluation possible.

By applying the equations for coupled heat and mass transfer and using the new definitions of the inversion temperatures this work comprehensively explains the inversion temperature phenomenon.

After early research on the drying of different materials with air, humid air and superheated steam yielded contradictory results concerning the change in drying rate with the different gas phase compositions, some investigations concentrated on the evaporation of water into air at different temperatures and humidities. Yoshida and Hyodo [1] carried out extensive experiments using a wetted wall column and were the first to find an inversion temperature. They initially assumed that at this temperature the evaporation rate of water becomes independent of the humidity of the gas for a constant gas mass flow at the inlet. Since then, a number of publications have dealt with the theoretical and experimental determination of this inversion temperature, revealing a dependence of the inversion temperature on the gas compositions being compared [2–8]. Table 1 summarises some of the results of these investigations.

Chow and Chung were the first to theoretically investigate the evaporation of water for a laminar [2] and turbulent [3] gas stream over a flat plate using equations for heat and mass transfer, and derived an iterative similarity solution to the problem. They described what is now the most common explanation for the existence of the inversion temperature. The wet bulb depression becoming smaller with increasing steam content leads to a smaller heat flux and consequently to a smaller evaporation rate. At higher temperatures this effect becomes small compared to the total temperature difference and is compensated by the changing thermodynamic properties for heat transfer in steam. Their approach has been taken up by various authors [4–7]. Sheikoleslami and Watkinson [8] and Faber et al. [11] addressed the problem by simply using an equilibrium approach, where the assumption of the gas being saturated at the outlet makes the description of the mass transfer obsolete. However,

Table 1
Inversion temperatures from various investigations

Reference	Theory/experiment	Description	$T_{\text{inversion}}$ (°C)
Here	Theory	Wetted wall column, turbulent	199
[1]	Experiment	Wetted wall column, turbulent	160–176
[2]	Theory	Flat plate, laminar	250
[3]	Theory	Flat plate, turbulent	190
Here	Theory	Flat plate, turbulent	200
[4]	Experiment	Flat plate, turbulent	170–220
[5]	Theory	Moving flat surface and gas, laminar	230–260
[6]	Theory	Moving flat surface, laminar	225
[7]	Theory	Wedge, turbulent	200
[9] in [10]	Experiment	Impinging jets, laminar	218–230
[10]	Experiment	Impinging jets, turbulent	175
[8]	Theory	Equilibrium approach	164
[11]	Both	Fluidised bed, equilibrium approach	160

this approach is impractical for most drying applications other than fluidised bed dryers.

Although most publications determine the inversion temperature on the basis of constant mass flow, a comparison on the basis of constant volumetric flow, i.e. constant gas velocity, is of theoretical and practical interest as well. Faber et al. [11] report an inversion temperature of 406°C for constant volumetric flow which is considerably higher than their value of 160°C found for constant mass flow.

2. Theory

The evaporation of a liquid into a gas flow of a mixture of its own vapour and an inert gas is a problem of coupled heat and mass transfer. The problem encountered in this case is easy to describe since it only deals with a pure liquid and a gas that can be considered as a binary mixture.

Fig. 1 shows an increment of a liquid surface exposed to a gas flow. As long as this increment is sufficiently small the gas temperature and composition can be assumed to be constant within this increment in the direction of the gas flow. Applying the film theory, the coupled heat and mass transfer can be described. The resulting set of equations consists of two transfer and two balance equations. Heat is transferred between the gas flow and the liquid. Because mass transfer takes place at the same time due to evaporation, the heat transfer coefficient α_G has to be modified by the analytically derived Ackermann correction [12]. The resulting heat transfer equation is

$$\dot{q}_G = \alpha_G \frac{\Phi_W}{\exp(\Phi_W) - 1}(T_I - T_G) \quad \text{and} \qquad (1)$$
$$\Phi_W = \frac{\dot{n}_v \cdot c_{pv}}{\alpha_G}$$

where the heat transfer coefficient α_G can be calculated by the appropriate empirical Nusselt equation for the desired geometry.

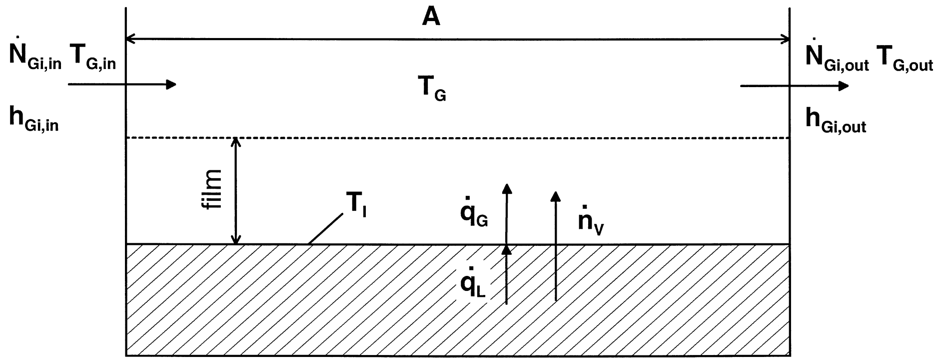

Fig. 1. Increment of a liquid surface exposed to a gas flow.

The mass transfer for a binary gas mixture with one non-transferring component can be described by

$$\dot{n}_\mathrm{v} = c_\mathrm{t} \cdot \beta_\mathrm{G} \cdot \ln\left(\frac{1 - y_\mathrm{vI}}{1 - y_\mathrm{vG}}\right) \tag{2}$$

where β_G is the mass transfer coefficient to be calculated from the Sherwood equation, which is analogous to the Nusselt equation.

The mass balance for the gaseous phase gives

$$\begin{aligned} &\dot{N}_\mathrm{Gv,\,in} + \dot{n}_\mathrm{v} \cdot A = \dot{N}_\mathrm{Gv,\,out} \quad \text{and} \\ &\dot{N}_\mathrm{Ga,\,in} = \dot{N}_\mathrm{Ga,\,out} = \text{const.} \end{aligned} \tag{3}$$

with air as the non-transferring component.

The heat balance for the gaseous phase is then described by

$$\begin{aligned} \dot{q}_\mathrm{G} \cdot A = {} & \left(\dot{N}_\mathrm{Ga,\,in} \cdot c_\mathrm{pa} + \dot{N}_\mathrm{Gv,\,in} \cdot c_\mathrm{pv}\right)\big(T_\mathrm{G,\,out} \\ & - T_\mathrm{G,\,in}\big) + \dot{n}_\mathrm{v} \cdot c_\mathrm{pv}\big(T_\mathrm{G,\,out} - T_\mathrm{I}\big)A \end{aligned} \tag{4}$$

where the molar enthalpies of the gas components have been substituted by the product of a mean molar heat capacity and the temperature difference. This simplification is widely used and does not significantly affect the calculated results. It is assumed that heat is transferred from the gas to the liquid only by convection and that there is no heat flux between wall and liquid. Furthermore, the liquid phase is assumed to be at the temperature of evaporation. With these assumptions, the heat balance at the interface becomes

$$\dot{q}_\mathrm{L} = 0 = \dot{n}_\mathrm{v} \cdot \Delta h_\mathrm{v}(T_\mathrm{I}) + \dot{q}_\mathrm{G}. \tag{5}$$

The interface has to be at equilibrium which gives the saturation pressure p_sat, and thus, the interface gas composition y_vI at the interface temperature. All thermodynamic properties of the vapour and the inert gas are functions of temperature and pressure. Assuming an ideal gas mixture, its properties can be calculated from the pure components with knowledge of the partial pressures. The unknown variables are the outlet temperature of the gas $T_\mathrm{G,\,out}$, the interface temperature T_I, the evaporative flux $\dot{n}_\mathrm{v}$ and the heat flux $\dot{q}_\mathrm{G}$. These parameters can be obtained by solving the set of four non-linear equations of heat (1) and mass (2) transfer and the heat balances (4) and (5).

For pure superheated steam there is no mass transfer resistance and the evaporation is solely controlled by the heat transfer to the interface. Eq. (2) does not apply and the interface temperature is determined by the pressure of the system.

Up to now the phenomenon of an inversion temperature has mainly been explained by the temperature depression between the interface and the bulk gas flow, as well as the change of thermodynamic properties with increasing temperature [2–7,10]. Consequently, only the heat transfer was taken into consideration. Although these findings are undoubtedly valid, a more comprehensive explanation of the inversion temperature phenomenon is possible by considering the coupled heat and mass transfer.

In air the process of evaporation is governed by simultaneous heat and mass transfer. The boundary layer incorporates a resistance for both transport processes. At low temperatures the process is mainly heat transfer controlled, since the concentration difference across the boundary layer is large as compared to a rather small temperature difference. With increasing gas temperature the temperature difference, and consequently, the heat transfer rate increases. The concentration difference increases as well because of the combined increase in temperature and saturation pressure at the interface. While the increase of the temperature difference is unlimited, the concentration difference reaches a maximum when the interface saturation pressure reaches the pressure of the system. Consequently, the whole process of evaporation becomes mass transfer controlled and the mass transfer resistance is the governing factor.

In pure superheated steam the mass transfer resistance does not exist, since the gas consists only of the evaporating species. This explains why the evaporation into pure superheated vapour is greater than into dry air at higher temperatures. The disadvantages of the higher interfacial temperature and less heat transfer potential in superheated steam are outweighed by the absence of a mass transfer resistance.

3. Results and discussion

A computer program was written to solve the problem described. The thermodynamic properties needed for air and water are calculated with equations from Daubert and Danner [13], the diffusion coefficient is estimated using the method of Fuller et al. [14] recommended in Ref. [15]. The pressure for all of the calculations presented here is atmospheric pressure. More detailed information about the calculations and the computer code can be found in Ref. [16].

3.1. General calculations

For the calculation of evaporation rates a single increment with the geometry of a wetted wall column is considered. The liquid phase is assumed to be stationary. The inner diameter of the column is 0.029 m, thus resembling the set-up used by Yoshida and Hyodo [1]. The required heat and mass transfer coefficients for turbulent flow in a circular duct are calculated with the correlation of Krupiczka and Rotkegel [17]:

$$Nu = \frac{\alpha_G \cdot d}{\lambda_G} = 0.0279 \cdot Re^{0.791} \cdot Pr^{0.44} \quad (6)$$

$$Sh = \frac{\beta_G \cdot d}{D_{av}} = 0.0279 \cdot Re^{0.791} \cdot Sc^{0.44} \quad (7)$$

Because the gas temperature and composition are assumed to be constant within the calculated increment in the direction of the gas flow, the length of this increment has to be sufficiently small. As shown later, a value of 10^{-3} m turned out to meet this requirement.

The evaporation rates for four different steam concentrations are plotted against the gas temperature for constant gas mass flow at the inlet in Fig. 2. There is no gas temperature for which the evaporation rate is independent of the gas composition. This is in contrast to the statement of Yoshida and Hyodo [1] and supports the findings of other investigators [2–8]. The local inversion temperature $LIT_{0,1}$ can be found at the intersection of the evaporation rate curves for completely dry air and pure superheated steam. For the calculated geometry $LIT_{0,1}$ is 198.6°C and calculations with mass flow rates of 9100 and 27,300 kg/m^2/h show that it is independent of the mass flow.

In addition calculations are carried out for turbulent flow over a flat plate resembling the experiments by Haji and Chow [4]. For these calculations the empirical correlations are extracted from Ref. [18].

$$Nu = \frac{\alpha_G \cdot l}{\lambda_G} = 0.036 \cdot Re^{0.8} \cdot Pr^{0.43} \quad (8)$$

$$Sh = \frac{\beta_G \cdot l}{D_{av}} = 0.036 \cdot Re^{0.8} \cdot Sc^{0.43} \quad (9)$$

The local inversion temperature $LIT_{0,1}$ found for one increment of the flat plate is 198.4°C. In this case, the Reynolds number incorporates the distance of the increment from the beginning of the flat plate. Calculations showed that the chosen distance affected the result for the local inversion temperature only in the second decimal place.

Even though the correlations for the transfer coefficients for the chosen geometries have a similar structure and do not exhibit large differences in the constants, they still resemble fundamental geometries frequently encountered in real processes. The influence of the geometry on the inversion temperature does not seem to be very large, even though the evaporation rates found at this inversion temperature may be significantly different.

The evaporation rates for four different steam concentrations are plotted against the gas temperature for constant gas velocity at the inlet in Fig. 3. The inversion temperature found is significantly higher. The calculations on the basis of constant gas mass flow and on the basis of constant volumetric gas flow (Figs. 2 and 3) are correlated by the density of the gases. Due to the different densities of air and water vapour the gas mass flow of air is higher at a constant velocity of the gases.

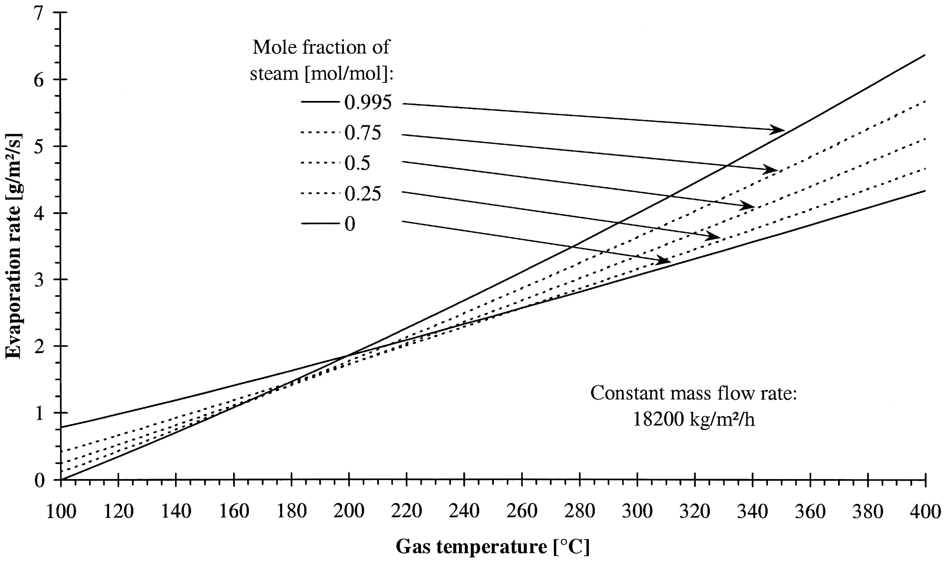

Fig. 2. Evaporation rates versus gas temperature at constant gas mass flow rate.

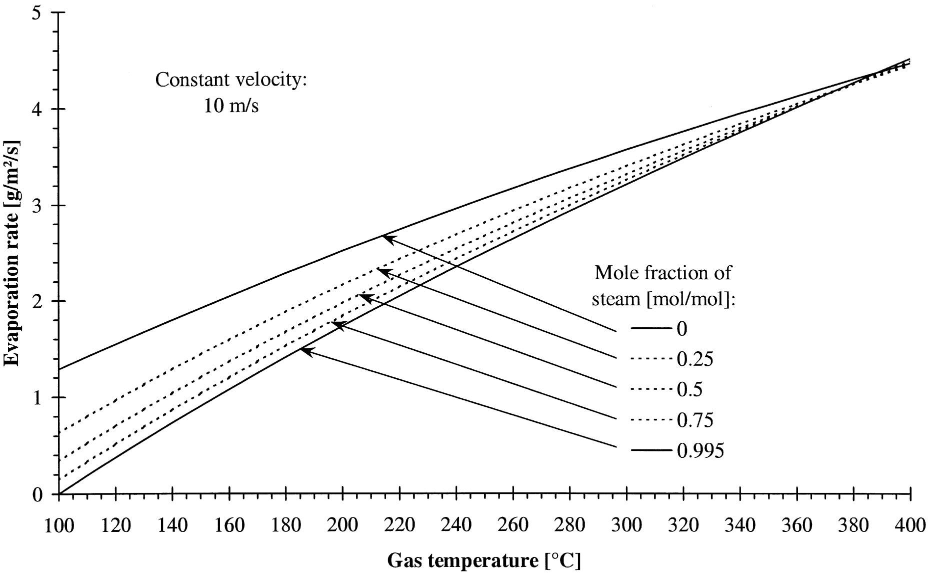

Fig. 3. Evaporation rates versus gas temperature at constant gas velocity.

3.2. Validation of the calculated evaporation rates

Despite the fact that the calculated local inversion temperatures fit in very well with the literature data in Table 1, the calculated evaporation rates do not necessarily have to be correct. To evaluate the quality of the model presented in this investigation the calculated evaporation rates have to be compared with experimental data. This has been done for the investigations of Yoshida and Hyodo [1] and Haji and Chow [4].

The wetted wall column of Yoshida and Hyodo [1] has a diameter of 0.029 m and a length of 1 m. Although their column had a counterflow of the phases, the liquid phase is assumed to be stationary in the calculations of this investigation. The flat plate of Haji and Chow [4] is 0.0508 m wide and 0.3048 m long and is enclosed in a duct 0.1062 m in width and 0.0254 m in height. The calculations for both geometries are carried out by dividing the interface into 1000 increments of identical area along the gas flow. Fig. 4 shows all the calculated evaporation rates plotted against the experimental findings. For the wetted wall column the values corrected for radiative heat transfer given by Yoshida and Hyodo [1] are plotted as well. For the flat plate only two values for the evaporation rate are explicitly given in the original publication [4], all other values had to be extracted from graphs that showed a line fitted to all the experimental data.

The calculated evaporation rates are in excellent agreement with the experimental findings. The results show that the model presented in this investigation is able to describe the real process of evaporation of a liquid into a gas flow of its own vapour and a non-transferring gas. The majority of the calculated evaporation rates tend to slightly underpredict the experimental data. This may be explained by the assumptions underlying the calculations, namely the assumption of a perfectly insulated liquid-wall interface, the neglected radiative heat transfer by steam and — for the wetted wall column — the neglected movement of the liquid phase. The influence of the radiative heat transfer can be estimated from the data corrected for purely convective heat transfer given by Yoshida and Hyodo [1]. As expected the agreement with the calculated results is even better.

The largest underprediction is found for the evaporation rates into dry air measured by Haji and Chow [4]. They claimed a strong influence of conductive heat transfer from the walls to the liquid on these measurements and introduced a correction. The correction is not included in this paper so that only the measured data are shown. The corrected values, which are slightly smaller than the evaporation rates calculated in this investigation, fitted their calculated results perfectly. However, the authors consider a comparison to the originally measured data to be more meaningful, especially since no details of the correction procedure were given.

3.3. Determination and definition of the local and apparent inversion temperatures

With such a good prediction of the evaporation

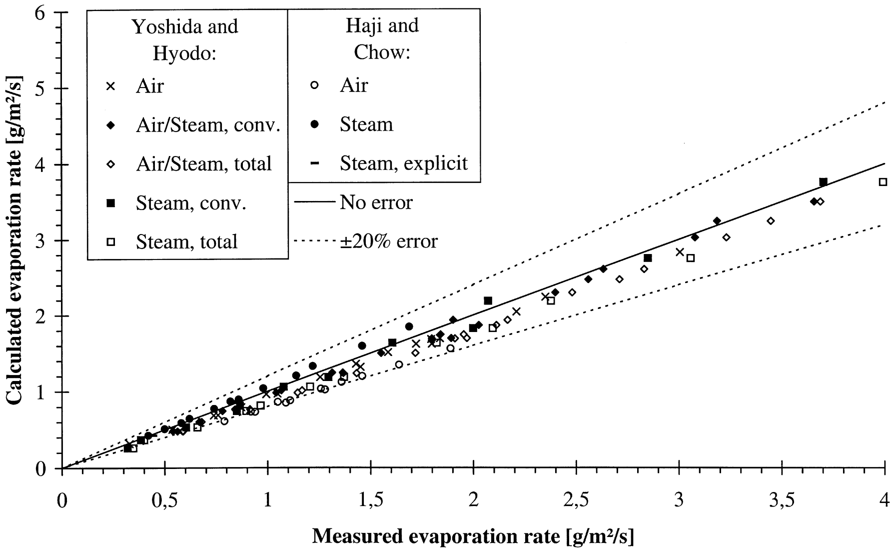

Fig. 4. Comparison between calculated and experimentally determined evaporation rates.

rates it would be expected that the inversion temperatures found in the general calculations above would agree with the experimental findings as well. For the flat plate of Haji and Chow [4], the inversion temperature found is indeed within the experimentally determined range. However, this is partly due to the fact that the range cited is relatively large because of data scatter. For the wetted wall column of Yoshida and Hyodo [1], the agreement is far less satisfactory.

This discrepancy highlights the problem which leads to the review and refinement of the inversion temperature definition. The original definition of the inversion temperature assumes the changes in gas flow, composition and temperature in the direction of flow to be negligibly small. Consequently, the amount of evaporation, and thus the surface area, have to be infinitely small. For this reason, the term local inversion temperature is introduced for this temperature, which can only be found by calculation. However, any experimental investigation has to deal with finite surface areas for evaporation, the size of which is determined by the accuracy and resolution of the measurement technique employed. The gas temperature and composition will change in the direction of the gas flow, and therefore, the definition of the local inversion temperature is not sufficient to describe the situation. Only an apparent inversion temperature can be measured. This apparent inversion temperature has to be defined at the inlet condition of the gas flow, because this is the only point where air can actually be completely dry. Values for the apparent inversion temperature have to be accompanied by detailed information about the geometry of the system and the gas flow to be evaluated.

To illustrate the change of the apparent inversion temperature with the size of the surface area for evaporation, calculations were carried out for wetted wall columns of increasing lengths. The inlet gas flows were completely dry air or pure superheated steam in all of these calculations. Three different gas mass flows as well as three different gas velocities have been calculated. The results are summarised in Fig. 5.

The apparent inversion temperatures cover a wide range of the temperature scale. The limits of this range are determined on one extreme by infinitely small lengths without significant changes of the gas condition in the direction of the flow (i.e., the local inversion temperature) and on the other by an infinitely long column with the maximum change in gas composition possible in the direction of the flow. This maximum is defined by a complete saturation of the gas at the outlet, and therefore, resembles the results of the equilibrium approach [8,11]. In between these two theoretically defined limits, the gas mass flow or velocity, respectively, has an influence on the observed inversion temperature since it affects the changes in gas temperature and composition along the column. With a higher gas flow, the humidification of the dry air will be less pronounced, and consequently, the apparent inversion temperature will be higher than that for smaller gas mass flows. However, this effect is small compared to the overall drop in the apparent inver-

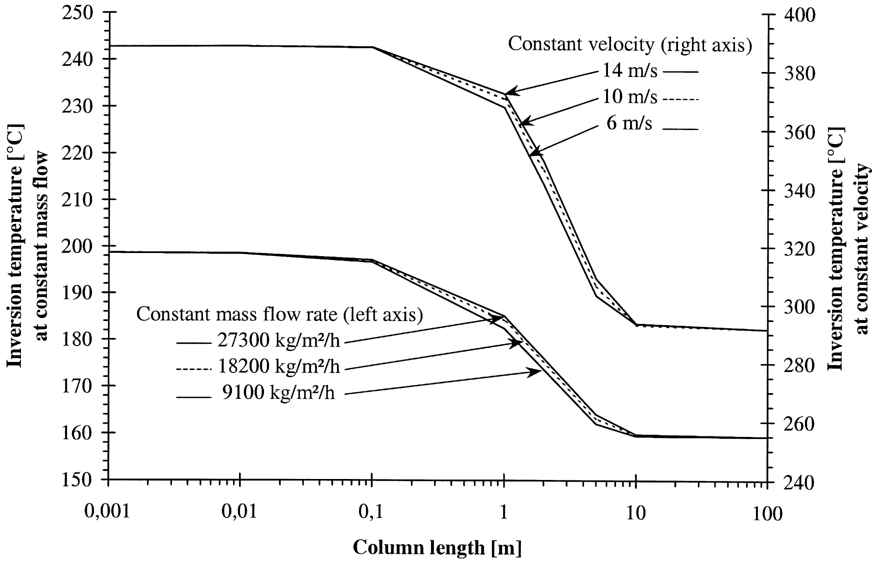

Fig. 5. Variation of the apparent inversion temperature for wetted wall columns of different lengths.

sion temperature with increasing column lengths, and thus, gas condition changes. This has implications for real evaporation processes, such as drying in the constant rate period. A finite layer of material to be dried leads to lower apparent inversion temperatures compared with the local inversion temperature for completely dry air and superheated steam of the geometrical system under consideration.

The data of Yoshida and Hyodo [1] were reanalysed with the new definition for the apparent inversion temperature, which yielded different results from those given in the original publication. This is due to the fact that they determined their inversion temperatures as the arithmetic mean gas temperatures within the column for which the mean evaporation rates were equal, as opposed to the determination on the basis of gas inlet temperatures, as is proposed for the apparent inversion temperature in this paper. Fig. 6 shows the measured and calculated evaporation rates plotted against the gas inlet temperature. Due to the underprediction of the measured evaporation rates, as discussed above, the calculated apparent inversion temperatures are higher than those experimentally determined. The difference seems large because of the small angle of intersection of the two curves. It is interesting to note that in this presentation of Yoshida and Hyodo's [1] results the theoretically found trend of a higher apparent inversion temperature with a higher gas mass flow is confirmed by the experimental data.

4. Summary and conclusion

The evaporation of liquids into their own vapour, air and mixtures thereof is a problem of coupled heat and mass transfer. The governing equations can be set up if the Nusselt and Sherwood equation for the heat and mass transfer coefficients in the desired geometrical system are known. The set of non-linear equations can be solved giving the interface temperature, evaporation rate and gas outlet conditions for a small calculated increment. The inversion temperature can be found as the intersection of the evaporation rate curves for dry air and superheated steam when plotted over the gas inlet temperature. New definitions of the inversion temperature are introduced to allow a distinction between theoretically found values meeting the assumptions of the original definition and experimentally found values that are influenced by the finite amount of evaporation encountered in real processes.

The evaporation rates calculated from the model presented in this investigation are in excellent agreement with literature data. For the simulation of two independent experimental investigations, the calculation results are within ±20% of the experimental data. The fact that the calculations in this investigation do not involve any parameters fitted to the experimental data they are compared with

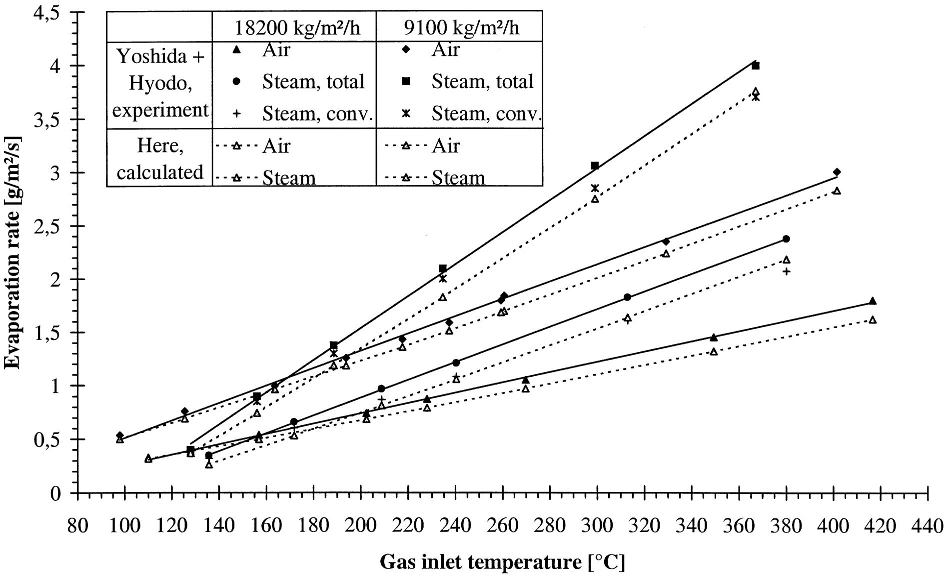

Fig. 6. Evaporation rates in the wetted wall column versus the gas inlet temperature.

clearly adds to the credibility of the model presented here. Deviations are generally small and can be explained by the simplifying assumptions made.

Good agreement is also found between the apparent inversion temperatures determined experimentally and those found by simulation of these experiments. These apparent inversion temperatures may differ significantly from the theoretically defined local inversion temperature of the system. This explains some of the confusion about the inversion temperatures reported previously in the literature. Furthermore, the analysis shows that for the constant rate of real drying processes the apparent inversion temperature found may be significantly lower than the theoretically defined value.

The model presented here obviously yields good and reliable results for the evaporation rates. The results comprehensively describe the inversion temperature phenomenon. The model has some more inherent advantages. The generic calculation procedure makes its application to different geometries and chemical components easy. The incremental procedure makes the description of transfer from finite surface areas possible without loss of accuracy. The theoretical approach employed is widely used in different applications [19]. It is based on the principles of multicomponent mass transfer and the model can be extended to include special applications, such as the evaporation of multicomponent liquids [20] by changing to a matrix form.

The authors, therefore, believe that this model will be a useful and versatile tool for the prediction of evaporation rates and the inversion temperature from theory. It will thus help in the design of drying processes and estimation of the feasibility of different process options.

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