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TECHNICAL NOTES

Mixed convection heat and mass transfer in vertical annuli with asymmetric heating

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1. INTRODUCTION

PHASE change connected with evaporation and condensation is important in many engineering applications and natural environments. Outstanding examples include windy-day evaporation and condensation of mist and fog, distillation of a volatile component from a mixture with involatiles, double-diffusive convection in ocean flows, the protection of system components from high temperature gas streams in supersonic aircraft and combustion chambers, the process of evaporative cooling for waste heat disposal and cooling of microelectronic equipment. This note aims to investigate the role of latent heat transport, in association with the evaporation of the liquid water film and the condensation of water vapour along the wetted annular duct walls, in mixed convection flows influenced by the combined buoyancy forces of thermal and mass diffusion.

Mixed convection heat transfer in vertical channel flows affected by the thermal buoyancy force alone has been studied in detail [1–3]. The effects of mass diffusion on natural convection heat transfer have been examined for external flows [4–6] and internal flows [7–10]. Concerning forced convection flows, combined heat and mass transfer in laminar flows was well investigated [11–13]. Recently, mixed convection heat and mass transfer in vertical tubes and between parallel plate channels, which are two limits of the annular duct, has received most attention [14, 15]. Despite the fact that mixed convection heat and mass transfer in vertical annuli is relatively important in engineering applications, it has not received enough attention.

In the present note, the geometry of the problem under consideration is a vertical concentric annulus with inner and outer radii being R_1 and R_2, respectively. The walls are wetted by thin liquid water films. The inner and outer walls are, respectively, kept at uniform but different temperature levels, T_1 and T_2, higher than the ambient temperature, T_0. The flow of moist air in the channel, initially stationary, is initiated by a mechanical device as well as by the combined buoyancy forces due to differences in temperature and in concentration of water vapour between the liquid films and the ambient. Attention is paid to investigating how the uneven annular duct wall temperatures affect the latent heat transport in conjunction with the evaporation of the liquid film or condensation of water vapour on the wetted walls.

2. ANALYSIS

With the Boussinesq and boundary-layer approximations, the mixed convection heat and mass transfer in a vertical annular duct can be described by the basic equations in dimensionless form as:

continuity equation

$$\frac{\partial(\eta U)}{\partial X}+\frac{\partial(\eta V)}{\partial \eta}=0; \tag{1}$$

axial-momentum equation

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial \eta}=-\frac{\mathrm{d}P}{\mathrm{d}X}+\frac{1}{\eta}\frac{\partial}{\partial \eta}\left(\eta\frac{\partial U}{\partial \eta}\right)+\frac{Gr_T\cdot\theta+Gr_M\cdot W}{4(1-N)^2 Re}; \tag{2}$$

energy equation

$$U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial \eta}=\frac{1}{Pr}\frac{1}{\eta}\frac{\partial}{\partial \eta}\left(\eta\frac{\partial \theta}{\partial \eta}\right)+\frac{A}{Sc}\frac{\partial \theta}{\partial \eta}\frac{\partial W}{\partial \eta}; \tag{3}$$

concentration equation of water vapour

$$U\frac{\partial W}{\partial X}+V\frac{\partial W}{\partial \eta}=\frac{1}{Sc}\frac{1}{\eta}\frac{\partial}{\partial \eta}\left(\eta\frac{\partial W}{\partial \eta}\right). \tag{4}$$

Equations (1)–(4) are subjected to the following boundary conditions:

$$X=0: \quad U=1, \theta=0, W=0, P=0 \tag{5a}$$

$$\eta=N: \quad U=0, V=V_1(X), \theta=r_T, \quad W=(w_1-w_0)/(w_r-w_0) \tag{5b}$$

$$\eta=1: \quad U=0, V=V_2(X), \theta=1, W=(w_2-w_0)/(w_r-w_0) \tag{5c}$$

where

$$V_i=-\frac{w_r-w_0}{1-w_i}\frac{1}{Sc}\left.\frac{\partial W}{\partial \eta}\right|_i, \quad i=1,2. \tag{6}$$

Since the air–water interfaces are assumed to be in thermodynamic equilibrium, the interfacial mass fractions of water vapour on the wetted walls can be evaluated by the

NOMENCLATURE

A	$[(c_{pv}-c_{pa})/c_p]\cdot(w_r-w_0)$
c_p	specific heat
D	mass diffusivity
D_h	hydraulic diameter
g	gravitational acceleration
Gr_M	Grashof number (mass transfer), $g(M_a/M_v-1)(w_r-w_0)D_h^3/v^2$
Gr_T	Grashof number (heat transfer), $g(T_2-T_0)D_h^3/(T_0 v^2)$
h_{fg}	latent heat of vaporization
h_M	local mass transfer coefficient
k	thermal conductivity
M	molecular weight
N	radii ratio, R_1/R_2
P	dimensionless motion pressure
Pr	Prandtl number, v/α
p_1, p_2	partial pressures of water vapour on inner and outer wetted walls, respectively
q''	interfacial energy flux flowing into air stream
Q_l	dimensionless wall heat flux (latent heat), equation (11c)
Q_s	dimensionless wall heat flux (sensible heat), equation (11b)
Q_x	dimensionless wall heat flux (total), equation (10)
Q_2	total heat transfer rate on the outer wetted wall
Q_2''	total heat transfer rate without mass transfer
r	radial coordinate
Re	inlet Reynolds number of the moist air
R_1, R_2	inner and outer radii, respectively
S_1	parameter, $\rho D h_{fg1}(w_1-w_0)/[k(T_1-T_0)]$
Sc	Schmidt number, v/D
T	temperature
U	dimensionless axial velocity
V	dimensionless radial velocity
V_1, V_2	dimensionless interfacial velocities of the moist mixture on the inner and outer wetted walls, respectively
W	dimensionless mass fraction of water vapour, $(w-w_0)/(w_r-w_0)$
w_r	saturated mass fraction of water vapour at T_2 and p_0
X	dimensionless axial coordinate
Y	dimensionless radial coordinate, $(r-R_1)/(R_2-R_1)$.

Greek symbols

α	thermal diffusivity
η	dimensionless radial coordinate, r/R_2
θ	dimensionless temperature, $(T-T_0)/(T_2-T_0)$
v	kinematic viscosity
ρ	density
ϕ	relative humidity of air in the ambient.

Subscripts

a	air
b	bulk quantity
r	reference condition
v	water vapour
0	inlet condition
1	condition at inner wall; i.e. at $r = R_1$
2	condition at outer wall; i.e. at $r = R_2$.

equations [8, 9]

$$w_i = p_i M_v/[p_i M_v + (p-p_i)M_a], \quad i = 1,2 \tag{7}$$

where p_1 and p_2 are respectively the partial pressures of water vapour on the inner and outer wetted walls.

In the nondimensionalization process, the following dimensionless variables are introduced:

$$X = 2x(1-N)/(R_2\, Re), \quad \eta = r/R_2$$
$$U = u/u_0, \quad V = vR_2/v$$
$$\theta = (T-T_0)/(T_2-T_0), \quad W = (w-w_0)/(w_r-w_0)$$
$$P = (p-p_0)/\rho u_0^2, \quad N = R_1/R_2$$
$$Gr_T = g(T_2-T_0)D_h^3/(v^2 T_0)$$
$$Gr_M = g(M_a/M_v-1)(w_r-w_0)D_h^3/v^2$$
$$Re = u_0 D_h/v, \quad D_h = 2(R_2-R_1)$$
$$r_T = (T_1-T_0)/(T_2-T_0), \quad Y = (r-R_1)/(R_2-R_1). \tag{8}$$

Energy transport between the wetted walls and the moist air in the presence of mass transfer depends on two related factors: the fluid temperature gradient along the wetted wall, resulting in a sensible heat transfer, and the rate of mass transfer, resulting in a latent heat transfer [16, 17]. The total energy fluxes from the wetted walls into the gas stream can then be expressed as

$$q_1'' = q_{s1}'' + q_{l1}'' = -k\left.\frac{\partial T}{\partial r}\right|_{r=R_1} - \frac{\rho D h_{fg}}{1-w_1}\left.\frac{\partial w}{\partial r}\right|_{r=R_1} \tag{9a}$$

and

$$q_2'' = q_{s2}'' + q_{l2}'' = k\left.\frac{\partial T}{\partial r}\right|_{r=R_2} - \frac{\rho D h_{fg}}{1-w_2}\left.\frac{\partial w}{\partial r}\right|_{r=R_2}. \tag{9b}$$

The dimensionless wall heat flux is defined as follows:

$$Q_{xi} = \frac{q_i''}{k(T_i-T_0)/D_h}, \quad i = 1,2. \tag{10}$$

Combining equations (9) and (10) yields

$$Q_{xi} = Q_{si} + Q_{li}, \quad i = 1,2 \tag{11a}$$

where

$$Q_{s1} = -2(1-N)\frac{1}{r_T}\left.\frac{\partial\theta}{\partial\eta}\right|_{\eta=N}, \quad Q_{s2} = 2(1-N)\left.\frac{\partial\theta}{\partial\eta}\right|_{\eta=1} \tag{11b}$$

and

$$Q_{l1} = \frac{-2(1-N)S_1}{1-w_1}\left.\frac{\partial W}{\partial\eta}\right|_{\eta=N}, \quad Q_{l2} = \frac{2(1-N)S_2}{1-w_2}\left.\frac{\partial W}{\partial\eta}\right|_{\eta=1}. \tag{11c}$$

Similarly, the local Sherwood numbers at the air–water interfaces are defined as

$$Sh_1 = \frac{h_M\cdot D_h}{D} = -2(1-N)\frac{w_r-w_0}{w_1-w_0}\left.\frac{\partial W}{\partial\eta}\right|_{\eta=N} \tag{12a}$$

and

$$Sh_2 = \frac{h_M\cdot D_h}{D} = 2(1-N)\frac{w_r-w_0}{w_2-w_0}\left.\frac{\partial W}{\partial\eta}\right|_{\eta=1}. \tag{12b}$$

In this note, the thermophysical properties of the mixture are taken to be constant and are evaluated by the one-third rule. This special way of computing the properties is found to be appropriate for the study of combined heat and mass transfer problems [11, 18]. The complete details on the evaluation are available in ref. [9].

3. RESULTS AND DISCUSSION

To check the adequacy of the numerical scheme employed in the present study, the results for the limiting case of laminar mixed convection heat transfer in a vertical annular duct were first obtained. Excellent agreement between the present predictions and those of El-Shaarawi and Sarhan [1] was found. Additionally, to produce the grid independent numerical results, several different arrangements of grid points in the x- and r-directions were tested. It is found that the differences in Q_{x1} for the computation by using 101×81 and 201×161 grids are always less than 2%. Accordingly, the results presented in this note are based on a 101×81 grid.

In the present note, calculations are particularly performed for the moist air in the annular duct, a situation widely found in engineering systems. The following conditions are selected in the computations: unsaturated moist air with a relative humidity of 50% at 20°C and 1 atm enters the long vertical annuli with outer radius being 3.0 cm from the bottom by the combined action of a certain external force as well as the buoyancy forces of heat and mass diffusion. All the non-dimensional parameters can then be evaluated. Results obtained for several cases are presented in Table 1.

To demonstrate the relative contributions of heat transfer through sensible and latent heat exchanges in the flow, three kinds of dimensionless wall heat fluxes along the inner wetted wall are presented in Fig. 1. Careful scrutiny of Figs. 1(a) and (b) indicates that in the initial portion of the annular duct, both Q_{s1} and $Q_{\ell 1}$ are positive. But as the moist air goes downstream, Q_{s1} and $Q_{\ell 1}$ change sign and become negative, except for case I—a symmetric heating case. In addition, the axial locations after which Q_{s1} and $Q_{\ell 1}$ become negative are closer to the channel entrance for the system with a higher T_2. The change of sign in $Q_{\ell 1}$ is undoubtedly associated with the direction of mass diffusion along the inner wetted wall. Near the channel entrance, because of the low concentration of water vapour in the air stream, the liquid film evaporates and generates water vapour into the air stream from the inner wetted wall. Therefore, the direction of latent heat transfer is from the inner wetted wall to the air stream, giving a positive $Q_{\ell 1}$. But as the moist air moves downstream, due to the strong evaporation from the outer wetted wall which is kept at a higher temperature, the mass fraction of water vapour in the flow could be over w_1 after a certain axial location. Thereafter, the condensation of water vapour occurs on the inner wetted wall, which, in turn, results in a negative distribution of $Q_{\ell 1}$. In Fig. 1(c), Q_{x1} is the sum of Q_{s1} and $Q_{\ell 1}$.

The effects of radii ratio N on the distributions of Q_{x1} and Q_{x2} are shown in Fig. 2. It is clear in Fig. 2(a) that the axial location at which Q_{x1} changes sign is closer to the channel entrance for the system with a larger radii ratio N. Moreover, a higher Q_{x1} is experienced for the system with smaller N in the initial portion of the channel. But as the flow moves in the downstream direction, the reverse is true. In Fig. 2(b), the dimensionless wall heat flux along the outer wetted wall is always positive. This implies that the heat transfer exchange is from the outer wetted wall to the gas stream.

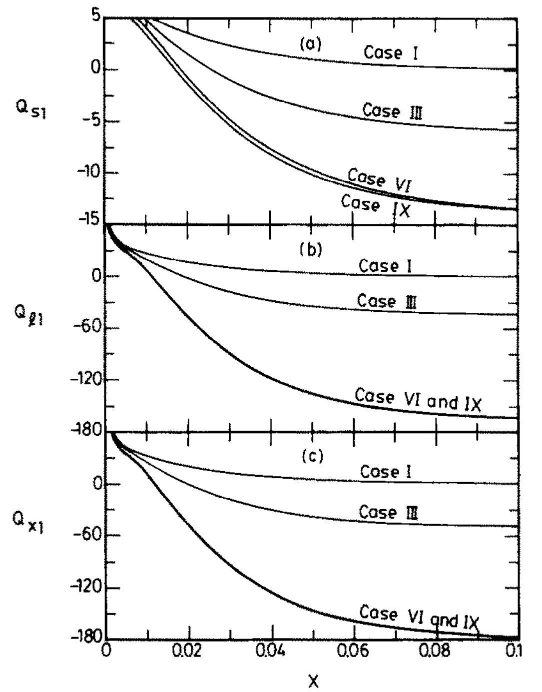

FIG. 1. Local dimensionless wall heat fluxes along the inner wetted wall: (a) sensible; (b) latent heat; (c) overall. Case I: $T_1 = 30°C$, $T_2 = 30°C$, $N = 0.5$, $Re = 2000$; Case III: $T_2 = 50°C$, $N = 0.5$, $Re = 2000$; Case VI: $T_2 = 70°C$, $N = 0.5$, $Re = 2000$; Case IX: $T_2 = 70°C$, $N = 0.5$, $Re = 500$.

The axial distributions of Sherwood number along both wetted walls are shown in Fig. 3. Because the outer wall temperature is maintained at a higher level than the inner wall temperature, the water vapour is always evaporated into the air stream from the outer wetted wall, and hence Sh_2 is positive. But on the inner wetted wall, Sh_1 is negative in the downstream region. This, as just discussed, implies that the condensation of water vapour occurs on the inner wetted wall in the downstream region.

To illustrate the effectiveness of latent heat transfer through mass diffusion, the heat transfer rate from the outer wetted wall is compared with the results for the situation in which both walls are not wetted. The results for Q_2/Q_2'' are presented in Fig. 4. Q_2'' represents the actual heat transfer rate on the outer wall under the same condition for each case except no water films on the channel walls. It is clearly seen that the capacity of energy transport through mass diffusion

Table 1. Values of major parameters for various cases

Case	T_1	T_2	N	Re	Gr_T	Gr_M	Pr	Sc	S_1	S_2
I	30	30	0.5	2000	37 182.55	12 771.31	0.706	0.594	5.391	5.391
II	30	50	0.2	2000	426 467.10	183 021.34	0.703	0.591	19.943	6.498
III	30	50	0.5	2000	104 117.95	44 682.94	0.703	0.591	19.943	6.498
IV	30	50	0.8	2000	6663.55	2859.71	0.703	0.591	19.943	6.498
V	30	70	0.2	2000	673 626.64	503 061.18	0.700	0.583	56.185	10.730
VI	30	70	0.5	2000	164 459.63	122 817.67	0.700	0.583	56.185	10.730
VII	30	70	0.8	2000	10 525.41	7860.33	0.700	0.583	56.185	10.730
VIII	30	50	0.5	500	104 117.95	44 682.94	0.703	0.591	19.943	6.498
IX	30	70	0.5	500	164 459.63	122 817.67	0.700	0.583	56.185	10.730

Units for parameters: T in °C, Φ in %, R_2 in m. $T_0 = 20°C$, $\Phi = 50\%$, $R_2 = 0.03$ m.

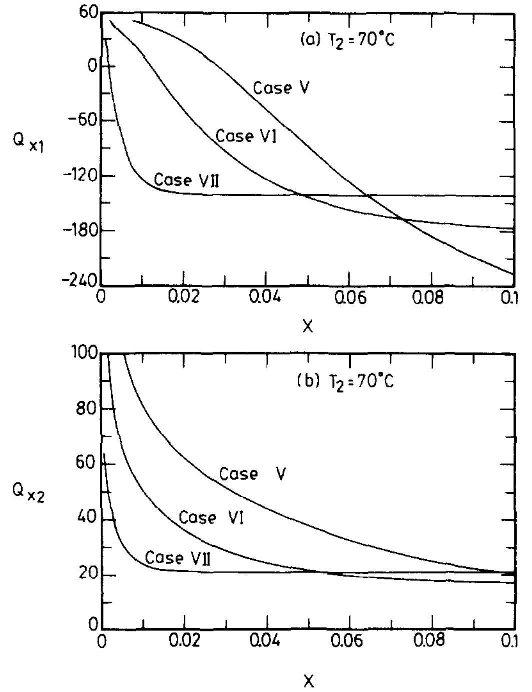

FIG. 2. Effects of radii ratio N on the distributions of dimensionless wall heat fluxes along both wetted walls. Case V: $T_1 = 300°C$, $T_2 = 70°C$, $N = 0.2$, $Re = 2000$; Case VI: $T_2 = 70°C$, $N = 0.5$, $Re = 2000$; Case VII: $T_2 = 70°C$, $N = 0.8$, $Re = 2000$.

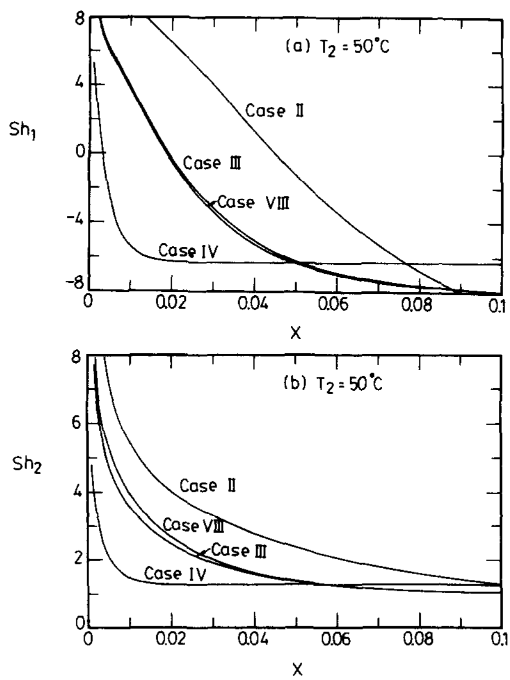

FIG. 3. Local Sherwood number distributions along both wetted walls. Case II: $T_1 = 30°C$, $T_2 = 50°C$, $N = 0.2$, $Re = 2000$; Case III: $T_2 = 50°C$, $N = 0.5$, $Re = 2000$; Case IV: $T_2 = 50°C$, $N = 0.8$, $Re = 2000$; Case VIII: $T_2 = 50°C$, $N = 0.5$, $Re = 500$.

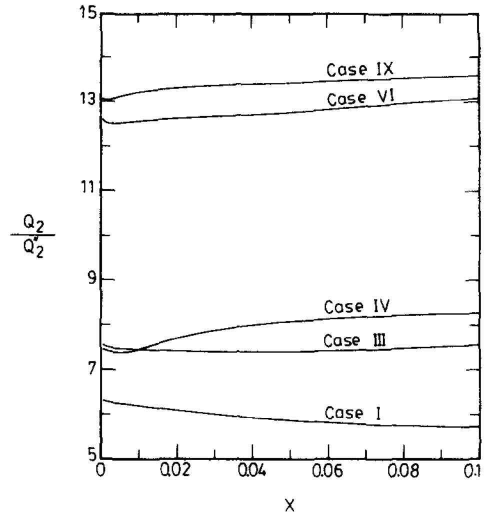

FIG. 4. Effects of system conditions on the total heat transfer rate. Case I: $T_1 = 30°C$, $T_2 = 30°C$, $N = 0.5$, $Re = 2000$; Case III: $T_2 = 50°C$, $N = 0.5$, $Re = 2000$; Case IV: $T_2 = 50°C$, $N = 0.8$, $Re = 2000$; Case VI: $T_2 = 70°C$, $N = 0.5$, $Re = 2000$; Case IX: $T_2 = 70°C$, $N = 0.5$, $Re = 500$.

is tremendous by noting that Q_2/Q_2'' can be as large as 13 for case IX ($T_2 = 70°C$ and $Re = 500$).

4. CONCLUSIONS

The nature of laminar mixed convection heat and mass transfer in vertical concentric annuli has been studied for an air–water vapour mixture system. The effects of wetted wall temperatures, Reynolds number of the flow and radii ratio on the heat and mass transfer were investigated in great detail. A brief summary of the major results is given below.

(1) Heat transfer in the flow is dominated by the transport of latent heat in conjunction with the vaporization or condensation of water vapour along the wetted walls.

(2) Because of a larger amount of water vapour evaporated into the air stream for the system with a higher T_2, the axial location where the incipience of the condensation on the inner wetted wall is closer to the channel entrance.

(3) In the initial portion of the channel, higher dimensionless wall heat flux Q_x and Sherwood number Sh along both wetted walls are experienced for the system with smaller radii ratio N.

(4) The presence of the mass diffusion effect causes pronounced heat transfer enhancement.

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On the analysis of counter-flow cooling towers

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INTRODUCTION

UNTIL recently the performance of counter-flow cooling towers was most commonly analyzed by applying the so-called Merkel model. This model is based on equations developed in a paper published in German by Merkel in 1925 [1]. This work was largely neglected until the paper was translated into English by Nottage in 1941 [2]. Since then the model has been widely applied. The equations express an energy balance and describe simultaneous mass and heat transfer coupled through the Lewis relation. However, in the interest of tractability the equations were simplified by omitting a term and as a result do not account for the mass of water lost by evaporation. Given the inlet water and air conditions the Merkel equations predict the enthalpy (hence wet-bulb temperature) of the outlet air, but not its humidity. The equations also predict the required number of transfer units (*NTU*) to accomplish the process. The *NTU* expresses the relationship which must exist between the mass/heat transfer coefficient and the tower volume to make the process possible. The Merkel equations are readily solved numerically using the modified Euler procedure, which can easily be executed in a spreadsheet format on a PC.

Recently a computer program called VERA2D [3] was developed for the Electric Power Research Institute (EPRI) to provide a two-dimensional model for determining cooling tower performance, both counter-flow and cross-flow. VERA2D also corrects an error in the derivation of the Merkel equations. The present paper retains the assumption of a one-dimensional model but corrects the Merkel equations so that the mass of water lost by evaporation is properly accounted for. Consequently with the model developed here the enthalpy and humidity of the air exiting the tower are determined. Corrected values of *NTU* are also evaluated. Now, a set of differential equations must be solved rather than just one. However, again using the modified Euler procedure the solution is readily executed in a spreadsheet format on a PC. It is found that the Merkel equations underestimate the required *NTU* by an amount which can be significant. In addition to improving the prediction of the required *NTU*, this model predicts the state of the outlet air, not just its enthalpy. It is necessary to know the state of this air if the effect on the environment of the cooling tower operation is to be determined; by, for example, entropy or exergy calculations.

BASIC THEORY

Consider a vertical counter-flow cooling tower in which liquid water enters the top at a mass flow rate L_1 and a temperature t_1, and leaves the bottom at a mass flow rate L_2 (less than L_1 due to evaporation) and a temperature t_2. Air enters the bottom of the tower at a mass flow rate $G(1+X_3)$ (where G is the mass flow rate of dry air and X is the absolute humidity; i.e. mass of water vapor per unit mass of dry air), a dry-bulb temperature T_3 and a wet-bulb teperature T_3^*, and leaves at the top at a temperature T_4 and an absolute humidity X_4.

L_1, t_1, t_2, G, T_3, and T_3^* (hence X_3) will be considered given as describing a particular cooling task; i.e. L_1 cooled from t_1 to t_2 (with some evaporation) by a dry air flow G with atmospheric conditions described by T_3 and T_3^*. The problem is to determine T_4, X_4 (and hence the liquid water loss due to evaporation), and the relationship between the mass and heat transfer coefficients and the tower volume required to perform the operation. This will be accomplished by requiring mass and energy conservation, and applying appropriate mass and heat transfer relationships.

We will adopt a one-dimensional model by assuming that the state of the water and air varies only with vertical position