Prediction of the Performance of a Spined-Tube Absorber—Part 1: Governing Equations and Dimensional Analysis

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ABSTRACT

The falling liquid film has become a popular means of transferring heat and mass from a vapor to a binary liquid, especially in gas-fired heat pump systems. Ideally, the required amount of heat and mass transfer can be accomplished by using a simple cylindrical tube; however, increasingly stringent size and weight requirements for the machine generally prohibit use of the simple cylindrical surface and other, more complex, surfaces have been sought that are believed to have a higher absorption capacity. Thus, in this paper, absorption of water into a lithium-bromide-water ($LiBr$-H_2O) film on a spine-finned or spiny tube is considered. The governing equations for the fluid flow and heat and mass transfer problems are derived, and it is shown that solution to the full equations for use in a design is not practical. Dimensional analysis is employed to simplify the governing equations considerably when the influence of surface tension is small. It is shown that on the local spine scale, the temperature distribution is conduction-dominated and when the influence of surface tension is small, the mass transfer problem is governed by essentially smooth tube dynamics. A model for the performance of the tube is developed in part 2.

INTRODUCTION

In this paper the nature of the flow and heat and mass transfer on a spine-finned tube is examined. A model to predict the total mass absorbed on the tube and other relevant design quantities is developed in part 2 of this work (Conlisk 1996) based on a dimensional analysis of the problem given in this paper; the nature of the flow and heat and mass transfer on the tube depends on the relative magnitude of a number of dimensionless parameters. There are three major objectives of this work. First, the primary objective has been to develop a simplified design procedure to calculate the mass absorbed on the "spiny" tube as a function of a number of geometric and flow parameters. Second, there has been a concerted effort to minimize the use of free constants to better fit the experimental data. Finally, we have also sought to minimize the need for large-scale computing power. The present model may be executed on a personal computer, with the solution for a single design point taking only seconds to run.

The geometry of the spiny tube is complicated (Figure 1), with the flow theoretically able to vary on a length scale much shorter than the length of the tube; a schematic of the tube defining the local spine coordinate system is shown in Figure 2. At this point it is perhaps appropriate to define the spiny tube geometry and define terms. In the local spine coordinate system, the direction of the z coordinate is termed the *axial* direction, the direction of the x coordinate is termed the *spanwise* direction, and the direction of the y coordinate is termed the *normal* direction. The axial direction is the primary direction of fluid motion. The pitch of the tube is the distance between spine rows, and the chord of the spine is the length of the spine in the primary flow direction. The chord at the base of the spine is termed the *root chord*, and the length scale (L_z) depicted in Figure 2 is the pitch. A spine row is the collection of spines in the tube azimuthal direction. For example, a single spine from three spine rows is depicted in Figure 2b. A spine channel is the region between two spines in the same row. A spine channel has a local spanwise length equal to L_x and an axial length equal to the local spine chord.

As with all flows that can be modeled as a continuum, the fluid flow is governed by the solution of the Navier-Stokes equations along with conservation of mass. To make matters more complex, the energy and mass transfer equations must be solved simultaneously in the same geometry. Because of the extremely complicated geometry, solution of the full Navier-Stokes equations in each spine channel

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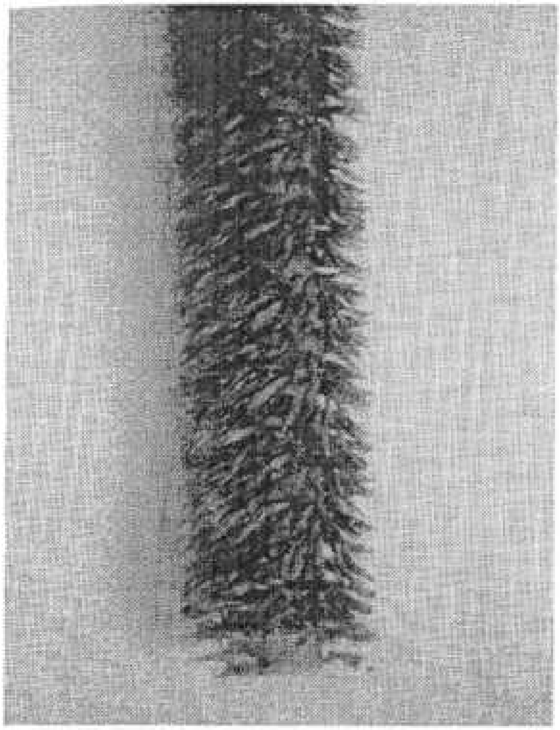

Figure 1 Photograph of a typical section of a spiny tube.

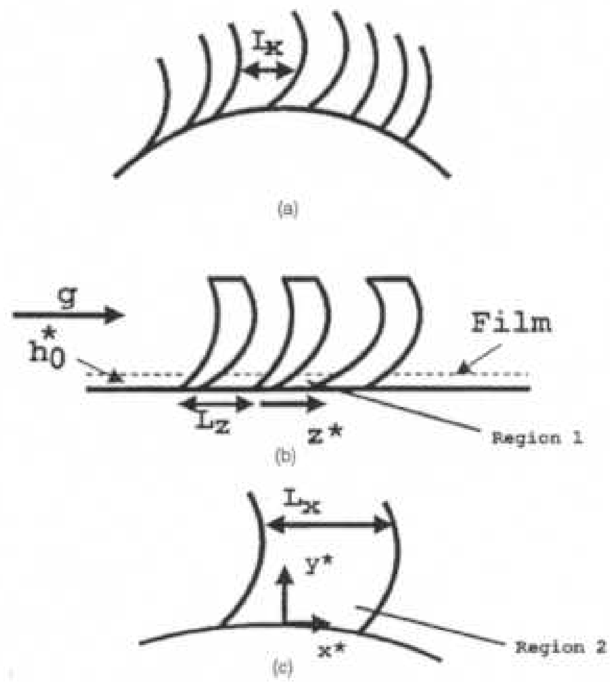

Figure 2 Geometry of the spiny tube: (a) schematic of a typical end viw of a portion of a tube; (b) side view of three spine rows; (c) end view of a single spine channel.

and between spine rows would be a formidable task, the results of which would likely be of no use for design of these tubes. Consequently, a different approach is taken here; the simplifications considered are those that arise when some of the physical effects inherent in the flow and heat and mass transfer problems may be neglected. Whether certain physical effects can be neglected depends on the magnitude of the relevant dimensionless parameters for each of the fluid, heat, and mass transfer problems. Two results of this type of approach are the facts that, on the local spine scale, the temperature may be assumed to be conduction-dominated and that significant mass transfer can be shown to take place only near the liquid-vapor interface (Conlisk 1992, 1994a, 1994b, 1994c). In the present problem, a critical simplification is the fact that for the parameters considered in the experiments (Miller 1993; Miller and Perez-Blanco 1994), direct calculation of the surface-tension-driven flow near the spines does not appear to be necessary to develop a simplified model for the best-performing tubes (Conlisk 1996).

The model for the absorption problem on the spiny tube is based upon analysis of two regions of interest. In the region between the spine rows that is depicted in cross section in Figure 2b (region 1), it is assumed that the influence of the spines themselves is negligible and the heat and mass transfer problems are governed by the global variables of total flow rate and total length of the tube. In the region between individual spines that collectively make up a spine row, termed here a *spine channel*, the heat and mass transfer is assumed to vary on the axial length scale defined by the tube pitch (Figure 2c, region 2). In this region between the spines it is assumed that the wall temperature is constant because the chord of the spines is so short, and in this region a closed form for the mass absorbed is obtained (Conlisk 1995).

Patnaik et al. (1993) and Perez-Blanco (1988) have developed design tools in ammonia-water and LiBr-H_2O mixtures for absorption into falling films on smooth tubes. For the LiBr-H_2O problem, Patnaik et al. (1993) use a one-dimensional approach to calculate the amount of water vapor absorbed into the liquid film. They use standard correlations for heat and mass transfer coefficients to model the heat and mass transfer in the film. Heat and mass transfer coefficients are taken from the literature to complete the problem so, strictly speaking, the work is not truly predictive. Design charts are produced for a wide variety of coolant and film-side parameters (see also Andberg and Vliet [1983]). Fundamental calculations have also been performed by Grossman (1983). Additional references and an extensive discussion of past work in the area appears in Conlisk (1992). It is instructive to note that all this work is for smooth tubes and, to the author's knowledge, no work of a predictive nature for an enhanced tube has appeared in the literature (although Conlisk [1994c] has formulated the problem for the fluted tube). Indeed, the numerical techniques used in much of the work in the absorption area simply cannot be applied to more complicated geometries.

The plan of this paper is as follows. In the next section, the assumptions of the model are presented and the full governing equations are presented next. Then, dimensional analysis is employed to determine the important physical effects that must be incorporated into the model; in this section, the magnitude of each of the dimensionless parameters is calculated and, on this basis, the fluid flow and heat and mass transfer problems may be greatly simplified. The solution for the fluid flow on the tube is given and, following this, the heat and mass transfer problems are formulated and solved; in particular, because the influence of surface tension is weak, at least for the best-performing tubes, the solution for the smooth tube may be employed as the generator of a model for the Heatron tube.

In past work by the author (Conlisk 1992, 1994a, 1994b, 1994c), the wall temperature must be specified. To eliminate this requirement, in part 2, the coolant-side problem has been formulated and coupled with the film side, and a model for the prediction of the absorbed mass flux on the spine-finned tube is developed. In this way, the entire absorber system is modeled; a countercurrent mode of operation is assumed. The model is subject only to the specification of a single coolant-side heat transfer coefficient and, in the absence of surface tension effects, no other free constants need to be specified. Moreover, the model requires only the numerical calculation of two integrals and thus a minimum of numerical work is required.

ASSUMPTIONS

The methods to be used are similar to those used in Conlisk (1992). The analysis is fully two-dimensional and does not require the use of any heat and mass transfer coefficients on the solution side. The following are assumptions are made.

- The fluid is Newtonian having constant properties. The tube is oriented vertically, and the flow and heat and mass transfer problems are assumed to be steady.
- The vapor is a pure fluid, and hence, no mass transfer takes place in the vapor. The velocity in the vapor is low and so the liquid-vapor interface is, to leading order, stress free. The vapor is taken to be water vapor.
- The film is thin in the sense that acceleration of a fluid particle within the film is negligible. This means that the Reynolds number is small, typically being on the order of 20 to 60. The effect of surface waves on mass transfer is neglected, as suggested by Javdani (1974).
- Conduction is the dominant mode of heat transfer in the liquid film.
- The relevant balance for mass transfer is between ordinary diffusion of mass and bulk convection. Thermal diffusion is negligible and, because the pressure changes within the system are small, pressure diffusion is negligible.
- The liquid-vapor interface is in equilibrium. For the LiBr-H_2O mixture of interest, this means that the mass fraction at the interface is a linear function of the temperature there.
- The flow is evenly distributed between each spine channel, so the flow is independent of the azimuthal direction. The flow in each spine channel is independent of the flow in the other spine channels.
- In a given spine channel, the local wall temperature is assumed to be constant.

GOVERNING EQUATIONS ON THE LOCAL SPINE SCALE

The primary effect of the spines is to allow the flow, and hence the heat and mass transfer, to vary on a scale much shorter than the tube length, much as the fluted tube generates a transverse flow driven by surface tension from the crest to the trough of the flute (Johnson and Conlisk 1987; Conlisk 1994c). In that case, this transverse flow prevents rapid growth of the film thickness down the tube and changes the vertical length scale over which the film thickness may change from the length of the tube to a length based on a balance between surface tension and gravity (Johnson and Conlisk 1987).

In this section the form of the governing equations in the local spine coordinate system is examined to determine the nature of the flow and heat and mass transfer problems there; the initial point of the domain is the inlet to the spine channel, and the end point is the inlet of the next spine channel. The spanwise width of the domain is on the order of the distance between the spines at the liquid-vapor interface (L_x; Figure 2). Thus, the new length scale in the primary flow direction is the pitch of the tube. It is then indicated how the governing equations on the global scale, defined by using the total length of the tube instead of the pitch for the axial length scale and by assuming axisymmetry, are obtained.

In what follows, the governing equations are nondimensionalized on the local scale based on the length triad (L_x, h_0^*, L_z), where the directions (x^*, y^*, z^*) are defined in Figure 2 and L_x is the nominal spine separation in the spanwise direction, h_0^* is the film thickness at the inlet to the domain, and L_z is the pitch of the tube. For the tubes of interest, it appears that $L_x \sim L_z$. The velocity scale, U_0, is computed from flow rate considerations, which yields one equation in the two unknowns, U_0 and h_0^*; the two scales are fixed by using the classic one-dimensional Nusselt relation $U_0 = g h_0^{*2} / \nu$, where ν is the kinematic viscosity.

Based on the flow rates of interest in this problem, the film thickness scale, h_0^*, is much smaller than all the other length scales in the problem, and the flow variations in the y-direction are much greater than those in the other two directions. Because of this, the flow is viscous in nature throughout the film. All parameters in this section are defined on the local spine scale unless otherwise specified.

To illustrate the basic procedure, consider the momentum equation in the z-direction, which is given by

$$\rho\left(u^*\frac{\partial w^*}{\partial x^*}+v^*\frac{\partial w^*}{\partial y^*}+w^*\frac{\partial w^*}{\partial z^*}\right)=-\frac{\partial p^*}{\partial z^*}$$

$$+\rho g+\mu\left(\frac{\partial^2 w^*}{\partial x^{*2}}+\frac{\partial^2 w^*}{\partial y^{*2}}+\frac{\partial^2 w^*}{\partial z^{*2}}\right),$$

where μ is the dynamic viscosity, ρ is the density, and all the * variables are dimensional. To define dimensionless distances, we write, for example, in the x-direction, $x = x^*/L_x$, and similarly for the other directions using the length triad as defined above. For the velocities, we write $(u, v, w) = (u^*, v^*, w^*)/U_0$. Then, for example,

$$\frac{\partial}{\partial z^*}=\frac{1}{L_z}\frac{\partial}{\partial z},$$

and, substituting into the momentum equation, we obtain

$$\rho U_0^2\left(\frac{u}{L_x}\frac{\partial w}{\partial x}+\frac{v}{h_0^*}\frac{\partial w}{\partial y}+\frac{w}{L_z}\frac{\partial w}{\partial z}\right)$$

$$=-\frac{1}{L_z}\frac{\partial p^*}{\partial z}+\rho g$$

$$+\mu U_0\left(\frac{1}{L_x^2}\frac{\partial^2 w}{\partial x^2}+\frac{1}{h_0^{*2}}\frac{\partial^2 w}{\partial y^2}+\frac{1}{L_z^2}\frac{\partial^2 w}{\partial z^2}\right).$$

Multiplying through by $h_0^{*2}/\mu U_0$, we obtain

$$\varepsilon_{sp}\mathrm{Re}\left(\lambda u\frac{\partial w}{\partial x}+\frac{1}{\varepsilon_{sp}}v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)$$

$$=-\varepsilon_{sp}\frac{\partial p}{\partial z}+\mathrm{ReFr}$$

$$+\left(\lambda^2\varepsilon_{sp}^2\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\varepsilon_{sp}^2\frac{\partial^2 w}{\partial z^2}\right),$$

where

$$\lambda = L_z/L_x,$$

$$\varepsilon_{sp} = h_0^*/L_z,$$

$$\mathrm{Re} = U_0 h_0^*/\nu,$$

and the dimensionless pressure is $p = p^*/(\mu U_0/h_0^*)$ and Fr is a Froude number, defined by $\mathrm{Fr} = g h_0^*/U_0^2$. This equation corresponds to the third component of Equation 1.

Governing Equations for the Fluid Flow

The governing equations for this problem are the Navier-Stokes equations; as noted above, only the steady flow equations are considered. In the present problem, the flow is of the lubrication type and it is natural to use the film thickness to define the Reynolds number. Nondimensionalizing the equations as described above, the governing equations in nondimensional form are given by

$$\varepsilon_{sp}\mathrm{Re}(\vec{V}\cdot\nabla_1)\vec{V}=\vec{P}+\frac{\partial^2\vec{V}}{\partial y^2}$$
$$+\varepsilon_{sp}^2\frac{\partial^2\vec{V}}{\partial z^2}+\lambda^2\varepsilon_{sp}^2\frac{\partial^2\vec{V}}{\partial x^2}, \tag{1}$$

where Re is the Reynolds number based on the length (h_0^*) and the scale velocity (U_0) as defined above. The operator ∇_1 is defined by

$$\nabla_1=\left(\lambda\frac{\partial}{\partial x},\varepsilon_{sp}^{-1}\frac{\partial}{\partial y},\frac{\partial}{\partial z}\right). \tag{2}$$

The vector $\vec{P}$ is the pressure gradient and body force and is defined by

$$\vec{P}=\left(-\lambda\varepsilon_{sp}\frac{\partial p}{\partial x},-\frac{\partial p}{\partial y},\mathrm{ReFr}-\varepsilon_{sp}\frac{\partial p}{\partial z}\right).$$

Because the film is very thin, the continuity equation requires that the velocity normal to the film surface is $O(\varepsilon_{sp})$, so the velocity vector $\vec{V}=(u, \varepsilon_{sp}v_0, w)$, where $v = \varepsilon_{sp}v_0 + \ldots$ to leading order. Thus, any mass absorbed is first and foremost $O(\varepsilon_{sp})$. The continuity equation is given by

$$\nabla_1\cdot\vec{V}=0. \tag{3}$$

The above equations are similar in form to equations derived in the three-dimensional lubrication theory except that the present flow is induced by gravity rather than by a pressure gradient (Hamrock [1991], p. 144). Equations 1 and 3 are a set of four highly nonlinear partial differential equations, which, in general, must be solved numerically.

Referring to Figure 2, the boundary conditions are no-slip at the spines and on the tube; that is, the fluid velocity must be zero at these locations. If each spine channel is to be analyzed as a separate fluid system, upstream and downstream conditions are required. For example, one can specify the undisturbed velocity upstream of the spine channel; the downstream condition can be formulated in several ways. If the pitch of the tube is long enough, to a good approximation, it can be specified that the solution is locally independent of the axial coordinate.

Finally, there are conditions at the free surface. The free surface conditions express the fact that the stress must be continuous across the interface; these conditions serve to specify the pressure distribution as well as (in the usual case) velocity gradients at the interface (Weyhausen and Laitone 1960) and are given by

$$p\lambda\varepsilon_{sp}\frac{\partial h}{\partial x}-2\lambda^2\varepsilon_{sp}^2\frac{\partial h}{\partial x}\frac{\partial u}{\partial x}+\frac{\partial u}{\partial y}+\lambda\varepsilon_{sp}\frac{\partial v}{\partial x}$$
$$-\varepsilon_{sp}^2\left(\frac{\partial u}{\partial z}+\lambda\frac{\partial w}{\partial x}\right)\frac{\partial h}{\partial z}=Ca\lambda\varepsilon_{sp}\left(\frac{1}{R_1}+\frac{1}{R_2}\right)\frac{\partial h}{\partial x}, \quad (4)$$

$$-p+2\frac{\partial v}{\partial y}-\lambda\varepsilon_{sp}^2\left(\frac{1}{\varepsilon_{sp}}\frac{\partial u}{\partial y}+\lambda\frac{\partial v}{\partial x}\right)\frac{\partial h}{\partial x}$$
$$-\varepsilon_{sp}\left(\frac{\partial v}{\partial z}+\frac{1}{\varepsilon_{sp}}\frac{\partial w}{\partial y}\right)\frac{\partial h}{\partial z}=Ca\left(\frac{1}{R_1}+\frac{1}{R_2}\right), \quad (5)$$

$$p\varepsilon_{sp}\frac{\partial h}{\partial z}-\varepsilon_{sp}^2\lambda^2\frac{\partial h}{\partial x}\frac{\partial w}{\partial x}-\varepsilon_{sp}^2\lambda\frac{\partial u}{\partial z}\frac{\partial h}{\partial x}$$
$$+\frac{\partial w}{\partial y}+\varepsilon_{sp}\frac{\partial v}{\partial z}-2\varepsilon_{sp}^2\frac{\partial w}{\partial z}\frac{\partial h}{\partial z} \quad (6)$$
$$=Ca\varepsilon_{sp}\left(\frac{1}{R_1}+\frac{1}{R_2}\right)\frac{\partial h}{\partial z}, \text{ at } y=h,$$

where R_1 and R_2 are the principal radii of curvature of the interface with respect to medium 1 (say, the vapor) and medium 2 (say, the liquid film), and h is the dimensionless film thickness. Note that the pressure on the viscosity has been scaled ($p = p^*/(\mu U_0/h_0^*)$) and so the capillary number, $Ca = \sigma_0/\mu U_0$, enters the boundary conditions at the interface. Here, μ is the dynamic viscosity and σ_0 is the surface tension coefficient, which is assumed to be constant. If the dimensionless pressure were scaled on the surface tension, then the Weber number would enter the equations. The Marangoni effect, which corresponds to the generation of flow patterns as a result of surface tension gradients, has not been included in the free-surface boundary conditions; more will be said about this in part 2 of this work (Conlisk 1996). The expression for the radii of curvature is given by Weyhausen and Laitone (1960), and for thin, flat films the result is

$$\frac{1}{R_1}+\frac{1}{R_2}=\varepsilon_{sp}^2\left(\frac{\partial^2 h}{\partial z^2}+\lambda^2\frac{\partial^2 h}{\partial x^2}\right).$$

Note that the influence of the vapor has been neglected in Equations 4 through 6. Consider, for example, Equation 6, which, in the absence of surface tension ($\varepsilon_{sp}^3 \mathrm{Ca} << 1$) reduces to

$$\frac{\partial w}{\partial y}=0.$$

The vapor stress would normally appear on the right side of this equation; however, because the film is thin and the vapor flow varies on a much larger scale, it is not high speed (assumption 2), and, because the liquid viscosity is much greater than the vapor viscosity, the leading-order vapor stress term is of a much lower order.

It is clear that Equations 1 and 3, subject to the boundary conditions, are extremely complex, and a closed-form solution that satisfies all the boundary conditions, including those at the spine surfaces, does not exist. Moreover, the geometry of the spiny tube effectively prevents the use of an efficient computational approach to solve the full governing equations in an individual spine channel. Thus an alternative approach must be employed and this approach is based on analyzing the order of magnitude of each of the relevant governing dimensionless parameters; this approach is also employed in the heat and mass transfer problems as well.

The flow rate of fluid across the interface (i.e., the mass absorbed into the film) is obtained by taking the dot product of the velocity vector with the gradient of the film thickness; the result is

$$\dot{m}_a = v_0-\lambda u\frac{\partial h}{\partial x}-w\frac{\partial h}{\partial z} \text{ at } y=h. \quad (7)$$

In the present nondimensionalization on the local spine scale, z ranges from zero to one, y ranges from zero to h, and the domain in x ranges from $x=-f_1(y,z)$ to $x=f_2(y,z)$, where f_1 and f_2 are functions defining the spine profiles (Figure 2).

Heat Transfer

Using the scalings of the previous section, the mechanical energy equation may be written in dimensionless form as

$$\varepsilon_{sp}\mathrm{RePr}(\vec{V}\cdot\nabla_1)T$$
$$=\frac{\partial^2 T}{\partial y^2}+\varepsilon_{sp}^2\frac{\partial^2 T}{\partial z^2}+\lambda^2\varepsilon_{sp}^2\frac{\partial^2 T}{\partial x^2}, \quad (8)$$

where Pr is the Prandtl number. The boundary conditions are

$$T = T_{sp} \text{ at the spines}, \quad (9)$$

$$T = T_W \text{ at the wall}, \quad (10)$$

and

$$q_{\hat{n}} = (\hat{n}\cdot\nabla_1)T$$
$$=\alpha\mathrm{RePr}\dot{m}_a h_{abs}/c_p, \text{ at } y=h, \quad (11)$$

where $\hat{n}$ denotes the normal direction to the interface, $q_{\hat{n}}$ is the heat flux in that direction, and h_{abs} is the heat of absorption. The heat flux defined in Equation 11 is due solely to the latent heat released in the phase-change process. There may be energy convected back into the vapor if the velocity in the vapor is fast enough. However, it is easily shown that the ratio of the heat transfer due to convection to the latent heat released in the absorption process is proportional to a vapor Lewis number ($Le_V = c_{pV}\Delta T_V/h_{abs}$). In the experiments, Le_V is on the order of 0.01. Thus, unless ΔT_V is very large, convection of heat back into the vapor should be small. This fact is supported by experiments (Miller 1995).

Note that once the velocity field is known, the energy equation becomes essentially linear in temperature. This is a simplifying feature; however, it is observed that the equation is still fully three-dimensional in its present form and is dependent upon an unknown film surface flux, $\dot{m}_a$, and the unknown shape of the fully three-dimensional film surface, h.

Mass Transfer

The mass transfer problem is defined by a balance of ordinary Fick diffusion and bulk convection of mass and is given by (Bird et al. [1960], p. 555 fol.):

$$\varepsilon_{sp}\mathrm{ReSc}(\vec{V}\cdot\nabla_1)\omega_A = \frac{\partial^2\omega_A}{\partial y^2} + \varepsilon_{sp}^2\frac{\partial^2\omega_A}{\partial z^2} + \lambda^2\varepsilon_{sp}^2\frac{\partial^2\omega_A}{\partial x^2}, \tag{12}$$

subject to

$$\frac{\partial\omega_A}{\partial y} = 0 \text{ at } y = 0, \tag{13}$$

$$(\hat{n}\cdot\nabla_1)\omega_A = -\alpha\mathrm{ReSc}\dot{m}_a(1-\omega_A) \text{ at } y = h, \tag{14}$$

and

$$(\hat{n}\cdot\nabla_1)\omega_A = 0 \text{ at the spines.} \tag{15}$$

Sc is the Schmidt number. In Equation 14, the factor $1-\omega_A$ comes in because of the form of the species flux relative to fixed axes and due to the fact that only water vapor is being absorbed (Bird et al. [1960], p. 499). The mass fraction gradient at the spine surface vanishes because the velocity normal to the wall is zero there (the solid wall condition). As with the energy equation, the mass transfer problem is fully three-dimensional and dependent on an unknown surface flux and film surface. The initial condition is that ω_A be specified at the inlet to the spine channel.

The Equilibrium Condition

Note from Equations 11 and 14 that the temperature and mass fraction fields are coupled through the presence of the unknown absorbed mass flux, $\dot{m}_a$. Consequently, one additional equation is required for closure of the system. This equation is obtained from the assumption of equilibrium at the liquid-vapor interface, and, for a linear absorbent (Grossman 1983), the equilibrium condition takes the form

$$\omega_A = C_1T + C_2, \tag{16}$$

where C_1 and C_2 are constants that are fit to equilibrium data for the LiBr-H_2O mixture at a given pressure for the temperature range of interest.

Governing Equations on the Global Scale

The governing equations have been derived for the flow field and the heat and mass transfer problems in the local spine coordinate system in which the length scale in the direction of primary motion is the pitch. The corresponding global coordinate system is defined as that system in which the total length of the tube is the length scale in the primary direction of motion, which is the z-direction. The governing equations in the global coordinate system, defined by using the length of the tube for the length L_z and by assuming axisymmetry, are obtained by setting $\partial/\partial x = 0$. In the global coordinates, because the length of the tube is much greater than the pitch of the tube, $\varepsilon << \varepsilon_{sp}$.

Discussion

In the present section, the governing equations have been derived in nondimensional form in the local spine coordinate system. To solve the problem on the spine scale, each of the governing equations for the fluid flow and the heat and mass transfer problems, along with the boundary conditions, must be solved within all of the spine channels. Clearly, solving the entire system is a daunting task; the equations for the fluid flow, in the general case, are highly nonlinear and require the determination of an unknown free surface. All the equations are fully three-dimensional. Even if a solution were possible, no design procedure could be developed using the computer code because of the large amount of computer time required. Consequently, other methods must be used to simplify the complex system of equations. To this end, the magnitude of each of the dimensionless parameters that govern the problem was analyzed to determine which terms in each of the equations govern the fluid flow and heat and mass transfer problems. For this purpose, two regimes were delineated where the heat and mass transfer take place on different length scales: (1) the region between the spine rows where the fluid is assumed to behave based on the global length and velocity scales defined by the entire length of the tube and total flow rate (region 1) and (2) the region within each spine channel where the flow and heat and mass transfer are governed by the pitch of the tube (region 2). These two regions are depicted in Figure 2.

DIMENSIONAL ANALYSIS

In this section, the order of magnitude of each of the dimensionless parameters that arise in the problem is examined. A simplified system of equations containing the essential features of the full system may be developed, leading to a simplified model for the flow and heat and mass transfer on the spiny tube.

Magnitude of the Dimensionless Parameters

Specifically, six data sets exhibiting different inlet flow rates, temperatures, and mass fractions and, hence, different mixture properties have been studied, two for each of the pitches considered. The pitch is defined as the distance from the inlet of one spine channel to the inlet of the next and is denoted by L_z in Figure 2. There are $N_{sp} = 25$ spine channels per circumference in each spine row (Vandersip 1993). The typical average velocity (U_0) based on the total flow rate down the tube is defined by

$$\dot{m}^* = \rho h_0^* C U_0, \tag{17}$$

where C is the circumference of the tube. The nominal film thickness on the spiny tube may be calculated by an equation that balances the overall effect of gravity and viscous stresses; this equation is given by

$$U_0 = \rho g h_0^{*2}/\mu, \tag{18}$$

which is formally equivalent to the equation derived directly from the Nusselt analysis (Conlisk 1992). In this case, ReFr = 1. This results in a global film Reynolds number of about $\mathrm{Re} = (U_0 h_0)/\nu \sim 20-60$. The same procedure may be repeated for the flow between each spine channel, with the length scale Lx replacing the circumference in Equation 17.

Table 1 gives the overall range of selected parameters for the spiny tubes. The specific data sets studied correspond to those provided by Miller (1993) for three sets of pitches; these data set are denoted by ht1-1, ht1-2, ht2-1, ht2-2, ht3-1, and ht3-2. The other data sets provided by Miller (1993) have qualities similar to those chosen specifically for analysis. The pitches considered are 0.25 in. (ht1-1, ht1-2), 0.1875 in. (ht2-1, ht2-2), and 0.125 in. (ht3-1, ht3-2) (see also Miller and Perez-Blanco [1994]). Table 2 gives the dimensionless parameters calculated for the corresponding spiny tubes for both the global and local length scales.

TABLE 1 Physical Dimensions and Other Parameters of the Spiny Tubes Considered in this Paper*

Tube base radius (in.)	.365
Total flow rate kg/min	0.4-1.0
Flow rate per spine channel (kg/min) check	.016-.04
Average inlet film thickness (in.)	~.004
Total tube length (ft)	5
Number of spine rows/ft	48-96
Axial length scale (L_z;in.)	5/32-9/32
Spine root chord (in.)	1/32-1/16
Spine gap (L_x;in.)	1/16-3/16
Bulk temperature in °C	30-55
Pressure (Torr)	10.34

*The average velocity and film thickness are calculated based on a balance between gravity and viscous forces.

Consider now the nature of the fluid dynamics problem. From Table 2 it is noted that eRe << 1; this means that the

TABLE 2 Spiny and Smooth Tube Parameters for Physical Parameters of Data Sets Provided by Miller (1993)*

Data Set	ht1-1	ht1-2	ht2-1	ht2-2	ht3-1	ht3-2
Mass Flux (kg/min)	.4605	.9215	.4032	.9170	.4633	.9010
Mass Fraction, LiBr in.	.6014	.6252	.6286	.6273	.5927	.6260
Absorber Pressure (torr)	10.34	10.34	10.34	10.34	10.34	10.34
Film Temperature (K)	325.44	325.00	329.53	330.45	322.19	325.92
Coolant Temperature (K)	308.28	308.13	323.40	307.79	308.80	318.89
Mass Transfer Parameter	.0277	.0490	.0121	.0524	.0345	.0269
Film Parameter ($\varepsilon \times 10^4$)	1.8	2.3	1.8	2.4	1.8	2.3
Film Parameter (ε_{sp})	.043	.055	.056	.075	.088	.111
Renyolds Number (Re)	30.94	54.54	22.93	50.65	29.11	49.71
Capillary Number (Ca)	69.20	39.7	68.08	38.08	65.44	39.08
$\varepsilon_{sp}^3 Ca$	.0055	.0066	.0113	.0160	.0446	.0533
Prandtl Number (Pr)	18.85	20.67	21.31	22.17	20.43	22.09
Schmidt Number (Sc)	2149	2456	2412	2690	2389	2565
Lewis Number(Le)	.0088	.0084	.0088	.0082	.0086	.0086
Jakob Number (Ja)	.0026	.0045	.0048	.0048	.0026	.0025

*Here the fit constants $C_1 = -.0048$, $C_2 = 1.9526$, and the total length of the tube is $L = 5$ ft (1.524 m).

acceleration terms in the momentum equations are negligible on the global scale. Moreover, even though $\varepsilon_{sp}\mathrm{Re} \sim 1$, the slow growth of the film thickness due to the large resistance to mass transfer (see below and Conlisk [1992]) renders the acceleration terms in the momentum equation negligible on the local spine scale as well. Thus, the fluid mechanics problem is effectively of the Nusselt type (see the next section) and closed-form solutions for the velocity field may be obtained away from the immediate vicinity of the spines. The influence of surface tension is expected to be felt first in the local spine system, and it is noted from Equations 4 through 6 and the expression for the radii of curvature that the influence of surface tension is governed by the product $\varepsilon_{sp}^3 Ca$. The influence of this parameter begins to increase in the data sets ht3-1 and ht3-2, where this parameter is about 0.05. The value of the surface tension is taken to be 0.083 N/m for all the data sets. Thus for all the data sets, as a first approximation, the influence of surface tension on the velocity field is neglected.

From the form of the energy and mass transfer equations, it is clear that the values of the quantities εRePr and εReSc determine the nature of the heat transfer and mass transfer, respectively, near the liquid-vapor interface. Note from Table 2 that Sc >> Pr; moreover, from the parameters depicted in Table 2, for all of the data sets, εRePr ~ 0.25 or less, which means that the influence of convective heat transfer is relatively minor in the region between the spine rows. The fact that conduction dominates means that heat transfer will take place over the entire film. In contrast, because εReSc >> 1 from the form of Equation 12, appreciable mass transfer will take place only near the interface.

In the region between the spines, $\varepsilon_{sp}\mathrm{RePr}$ can be somewhat large and so convection of heat may be important locally. However, the temperature gradient in the flow direction is small and the influence of thermal convection is reduced, as in the case of a smooth tube. Indeed, Conlisk (1995) has shown that the neglect of thermal convection in the smooth tube problem leads to a negligible error in the prediction of the mass absorbed. Moreover, the results using the conduction-dominated temperature profile across the film lead to a good comparison with the experiment, as shown in part 2 (Conlisk 1996); consequently, a conduction-dominated profile is assumed throughout the film.

To complicate matters, the velocities in each of the flow directions must be brought to relative rest at the surface of the spines. Consequently, there must be a thin region near the spines where the velocity is reduced to zero. From Equation 1, the width of this region is on the order of $O(\lambda\varepsilon_{sp})$. Theoretically, because the Nusselt-type flow cannot be valid near the spines, the three equations that make up Equation 1 must be solved in a local region near the spines and match the solution with the Nusselt-type solutions valid away from the spines; the equations are similar to those described by Hamrock (1991) for multidimensional lubrication equations. However, because the spines are solid surfaces, little mass transfer will take place within the boundary layers on the spines and thus, from a purely mass transfer point of view, solution of the flow near the spines is not required.

Discussion

At this point, based on the foregoing dimensional analysis, the fluid flow problem has been reduced to essentially one of the Nusselt type, where acceleration of the fluid particles is negligible. Moreover, the problem is effectively the same as the smooth-tube problem since surface tension is negligible. In addition, the heat transfer problem has been shown to be conduction-dominated. Finally, because εReSc >> 1 everywhere, mass transfer will take place only near the liquid-vapor interface as in all previous work. In summary, the magnitude of the dimensionless parameters suggests that we can begin developing a model for the spiny tube based on effectively smooth-tube fluid dynamics and heat and mass transfer.

FLUID DYNAMICS

The influence of surface tension from Equations 4 through 6 is to induce a stress on the flow in the film. While it has been shown that the influence of surface tension is not significant for the best-performing tubes, it is useful to write down the solution for the velocity field, including surface tension, for discussion later in part 2. A major result from the dimensional analysis is that fluid acceleration through a given spine channel is negligible. In this case, the solution for the flow in the film in global variables is given by

$$u = -\varepsilon\lambda\frac{\partial p}{\partial x}(hy - y^2/2) \tag{19}$$

and

$$w = \left(1 - \varepsilon\frac{\partial p}{\partial z}\right)(hy - y^2/2), \tag{20}$$

where the film thickness, h, is yet to be determined. The velocity normal to the film surface may be obtained from the continuity equation (Equation 3). For negligible surface tension, note that $u = 0$ and

$$w = hy - y^2/2 + O(\varepsilon^3 Ca),$$

which, to leading order, is the same as in Conlisk (1992) for the smooth tube. The solutions (Equations 19 and 20) do not satisfy any boundary conditions on the spines, and the influence of surface tension is proportional to $\lambda\varepsilon Ca^3$. The effect of surface tension in the local region between the spines is discussed in part 2.

In the next section, the heat and mass transfer problems on the global length and velocity scales are formulated for the case where the influence of surface tension is negligible to leading order.

HEAT AND MASS TRANSFER PROBLEMS

In this section the heat and mass transfer problems are formulated for the situation where surface tension effects are negligible. In this case, to leading order, the formulation of the problems in global variables is the same as for the smooth tube; this is essentially a shorter version of the presentation by Conlisk (1992) and, since the smooth-tube problem will form the core of the spiny tube model, it is repeated here for clarity and completeness.

Mass Transfer

The film-side mass transfer problem has been formulated in previous work; in what follows the authors give a brief development of the problem to motivate the inclusion of the coolant-side flow and heat transfer. Let ω_A denote the mass fraction of water in the mixture. Then, with $\eta = y/h(z)$, and defining

$$\Omega = \frac{\omega_A - \omega_{ABULK}}{\omega_{ASin} - \omega_{ABULK}}, \tag{21}$$

and near the interface, $\bar{\eta} = (1-\eta)/\delta^{1/2}$; then, in terms of Ω we have, to the leading order,

$$\frac{\partial^2\Omega}{\partial\bar{\eta}^2} = h^2 w\frac{\partial\Omega}{\partial z} - \frac{B}{\alpha}\frac{\partial\Omega}{\partial\bar{\eta}}\bigg|_{\bar{\eta}=0}\frac{\partial\Omega}{\partial\bar{\eta}}, \tag{22}$$

where B is an overall mass transfer driving parameter and is defined by

$$B = \frac{\omega_{ASin} - \omega_{ABULK}}{1-\omega_{ASin}}, \tag{23}$$

and α is a scaling constant defined by the ratio of the water density to the solution density in the bulk. In this problem, B is small and often much less than δ. At the liquid-vapor interface,

$$\delta^{1/2}B\frac{\partial\Omega}{\partial\bar{\eta}} = \alpha h\dot{m}_a(1+O(B)) \text{ at } \bar{\eta} = 0. \tag{24}$$

Equation 22 balances for $B \to 0$; however, the boundary condition at $\bar{\eta} = 0$ does not. The conclusion that must be reached is that since B and δ are both small, then

$$\dot{m}_a \sim \delta^{1/2}B; \tag{25}$$

that is, the lack of an $O(1)$ mass fraction difference and the fact that δ is small limits the amount of vapor that may be absorbed at the interface. Thus, to leading order, we may take $h = 1$. This result, which follows from simple scaling arguments, has been assumed *a priori* in previous work based on physical considerations (Grossman [1983] and others). It is interesting to note that the physical concept of a rapidly varying mass fraction near the liquid-vapor interface occurring at large values of ReSc merging into a regime where the mass fraction is essentially constant is mentioned in the treatise of Levich (1962, p. 57 fol.), so the present formulation in terms of a singular perturbation problem is well founded.

The solution to this problem subject to the initial condition $\Omega = 0$ at $z = 0$ and $\Omega = 0$ at $\bar{\eta} \to \infty$ has been given previously and the result is

$$\Omega = -\alpha\{2/(\pi)\}^{1/2}\int_0^z \dot{m}_{a0}e^{-\bar{\eta}^2/8(z-t)}dt/(z-t)^{1/2}, \tag{26}$$

where $\dot{m}_{a0}$ is defined by $\dot{m}_a = \delta^{1/2}B\dot{m}_{a0} + \ldots$ and is the leading order nonzero term in the absorbed flux. The leading order film thickness variation is defined by

$$\dot{m}_{a0} = -\frac{dh_1}{dz}, \tag{27}$$

where $h = 1 + \delta^{1/2}Bh_1 + \ldots$ is written in light of the discussion above. The average value of the mass fraction is required and the result is

$$\Omega_{AVE} = \delta^{1/2}\int_0^\infty \Omega d\bar{\eta} = 2\alpha\delta^{1/2}h_1. \tag{28}$$

Heat Transfer

The energy equation in the present problem is given by

$$\frac{\partial^2\theta}{\partial\eta^2} = \varepsilon\text{RePr}w\frac{\partial\theta}{\partial z}, \tag{29}$$

where θ is the dimensionless temperature, defined by

$$\theta = \frac{T - T_{Win}}{T_{Sin} - T_{Win}},$$

where T_{Sin} and T_{Win} are the film surface and wall temperatures, respectively, at the inlet to the tube. The boundary conditions are given by $\theta = \theta_W$ at $\eta = 0$, $\theta = 0$ at $z = 0$. At the interface, we have, to leading order,

$$\text{Ja}\frac{\partial\theta}{\partial\eta} = -\varepsilon\text{RePr}\delta^{1/2}B\dot{m}_{a0} \text{ at } 1. \tag{30}$$

In Equation 30, $\text{Ja} = c_p\Delta T/h_{abs}$ is the Jakob number, $\Delta T = T_{Sin} - T_{Win}$, and h_{abs} is the heat of absorption. Note that the assumption of a constant wall temperature has not been invoked.

As noted, in the region between the spine rows, on the global scale, $\varepsilon\text{RePr} << 1$ and in this case the convective terms in the energy equation are small away from the entrance. Moreover, even when $\varepsilon\text{RePr} \sim 1$, because of the slow growth in the film thickness due to the high liquid-side mass transfer resistance, the temperature distribution becomes essentially conduction-dominated soon after entering the tube (Conlisk [1992], Figures 2 through 5). In this case, write

$$\theta(\eta, z) = (\theta_S - \theta_W)\eta + \theta_W, \tag{31}$$

where θ_S is the film surface temperature and θ_W is the wall temperature on the solution side, both of which are unknown. It is interesting to note that Patnaik et al. (1993) employ this distribution for the temperature, although a numerical solution must be calculated for the average mass fraction across the film. For the conduction-dominated temperature problem, a closed-form solution for the absorbed flux may be found. Before this solution is discussed, we discuss the conditions at the interface.

Interface Conditions

As mentioned, to close the problem for the calculation of the absorbed mass flux, $\dot{m}_{a0}$, equilibrium will be assumed at the interface, and the temperature and mass fraction are assumed to be related linearly. In dimensionless form,

$$\Omega_S = \frac{C_1 \Delta T}{\Delta\omega}\theta_S + \beta, \tag{32}$$

where

$$\beta = \frac{C_1 T_{Win} + C_2 - \omega_{ABULK}}{\Delta\omega}. \tag{33}$$

Here C_1 and C_2 are constants obtained by curve-fitting experimental data in the mass fraction range of interest. Note that the parameter β is a measure of how far the wall temperature is away from equilibrium in the bulk. The determination of the scale factors ω_{ASin} and T_{Sin} is as described in Conlisk (1992) and relies on the fact that, near the inlet, there is a thermal boundary layer developing near the interface and (Conlisk 1992)

$$\theta_S = \frac{B\mathrm{Le}^{1/2}}{\mathrm{Ja}}\Omega_S + \theta_B, \tag{34}$$

where Le = Pr/Sc is the Lewis number and θ_B is the dimensionless bulk temperature. To determine the interface mass fraction at the entrance to the tube, we use the definition of Ω and Equations 32 and 34. Using the definition of B, assuming the bulk properties are constant, the equation for ω_{ASin} is given by

$$\begin{aligned}
&\omega_{Asin}^2 - (1 + \omega_{ABULK} + \beta_0 \\
&- C_1\frac{\mathrm{Le}^{1/2} h_{abs}}{c_p} - C_1(T_{BULKin} - T_{Win}))\omega_{Asin} \\
&+ \omega_{ABULK} + \beta_0 - C_1\frac{\mathrm{Le}^{1/2} h_{abs}}{c_p}\omega_{ABULK} \\
&- C_1(T_{BULKin} - T_{Win}) = 0,
\end{aligned} \tag{35}$$

where $\beta_0 = \beta\Delta\omega$.

SUMMARY AND CONCLUSIONS

The geometry of the spiny tube is complicated; moreover, the solution to the absorption problem requires the simultaneous solution of the velocity field and the temperature and mass fraction over the entire tube. The problem near the spines is, in general, fully three-dimensional. Consequently, rational approximation to the full problem has been sought by evaluation of the various physical effects through analysis of the magnitude of the relevant dimensionless parameters of the flow and heat and mass transfer problems. In particular, it has been shown that in the absence of surface tension effects at flow rates typical of application, the governing equations for the velocity field may be reduced to the Nusselt type in the bulk of the film; that is, only the boundary conditions on the velocity field at the tube wall and at the interface need be satisfied. The magnitude of the dimensionless parameters for heat transfer indicate that a good approximation may be obtained by a simple conduction-dominated profile. Finally, the mass transfer problem may be reduced to the smooth-tube case, where mass transfer takes place only near the liquid-vapor interface.

The results of the dimensional analysis have thus indicated that since the vast majority of the tube is not covered by spines, a spine-finned tube model may be developed using the smooth-tube solutions for the fluid flow, temperature, and mass fraction along with consideration of the mass absorbed between the spines. The physical model for the spiny tube is developed in part 2.

ACKNOWLEDGMENTS

This work has been supported by Martin-Marietta Energy Corporation. Frequent discussions with W.A. Miller have been extremely helpful during the course of this work. Mr. H. Vandersip has provided information concerning the geometry of the spiny tubes.

NOMENCLATURE

B	=	mass transfer driving parameter (Equation 23)
Ca	=	$\sigma_0/\mu U_0$
c_p	=	specific heat
D_{AB}	=	mass diffusion coefficient
Fr	=	Froude number
g	=	acceleration due to gravity
h	=	dimensionless film thickness = h^*/h_0^*
h^*	=	dimensional film thickness
h_0^*	=	dimensional film thickness at the inlet
h_1	=	$h = 1 + \delta^{1/2}Bh_1 + \ldots$
h_{abs}	=	heat of absorption
Ja	=	Jakob number = $c_p\Delta T/h_{abs}$
k	=	thermal conductivity
L	=	tube length
Le	=	Lewis number = Pr/Sc
L_x	=	spine spacing in the azimuthal direction
L_z	=	spine pitch plus spine length (Figure 1)

$\dot{m}_a$ = $\dot{m}_a^*/\rho U_0$, dimensionless absorbed mass flux
$\dot{m}_a$ = dimensional absorbed mass flux
$\dot{m}_{a0}$ = scaled absorbed mass flux
p = dimensionless pressure
Pr = Prandtl number = $\mu c_p/k$
$q_{\hat{n}}$ = defined by Equation 11
R_1 = radius of curvature with respect to the vapor
R_2 = radius of curvature with respect to the liquid
Re = Reynolds number = $U_0 h_0^*/\nu$
Sc = Schmidt number = ν/D_{AB}
T_{BULKin}= bulk temperature at the inlet
T_{Sin} = film surface temperature at the inlet
T_{sp} = temperature at the spine wall
T_{Win} = wall temperature at the inlet
ΔT = $T_{Sin} - T_{Win}$
u = dimensionless fluid velocity in the x direction
U_0 = gh_0^{*2}/ν, velocity scale
v = dimensionless fluid velocity in the y direction
v_0 = $v = \varepsilon_{sp} v_0 + \ldots$
w = dimensionless velocity in the z direction
x = dimensionless coordinate in the spanwise direction
x^* = dimensional coordinate in the spanwise direction
y = dimensionless coordinate normal to wall
y^* = dimensional coordinate normal to wall
z = dimensionless coordinate in the axial direction
z^* = dimensional coordinate in the axial direction

Greek Symbols

α = ρ_w/ρ in the bulk
β = defined in Equation 33
β_0 = $\beta\Delta\omega$
δ = $1/\varepsilon$ReSc
$\Delta\omega$ = $\omega_{ASin} - \omega_{ABULK}$
ε = h_0^*/L
ε_{sp} = h_0^*/L_z
λ = L_z/L_x
η = y/h
μ = mixture dynamic viscosity
ν = mixture kinematic viscosity
ρ = mixture density
ρ_w = density of water
σ_0 = surface tension
θ = $(T - T_{Win})/\Delta T$
θ_B = $(T_B - T_{Win})/\Delta T$
θ_S = dimensionless film surface temperature
θ_W = dimensionless wall temperature
ω_A = mass fraction of species A
ω_{ASin} = film surface mass fraction at the inlet
ω_{ABULK}= mass fraction in the bulk, constant
Ω = scaled mass fraction (Equation 3)

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QUESTIONS AND COMMENTS

G. Grossman, Senior Development Staff Member, Lockheed Martin Energy Research, Oak Ridge, Tenn.: The analysis assumes a developing concentration boundary layer but no thermal boundary layer; instead it shows a linear temperature profile across the film. Should there not be a developing thermal boundary layer also?

A. Terrence Conlisk: There is a developing thermal boundary layer but, because of the fact that the film thickness is almost constant, the length of this developing region is very short, about 1% of the tube. This is confirmed by a complete numerical computation of the solution for the temperature as described in my paper concerning the smooth tube problem (published in *AIChE J.* 1992, vol. 38, no. 11, p. 1716). This is referred to in part 1 of this work.