Review

The Naphthalene Sublimation Technique

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■ The naphthalene sublimation technique is reviewed in detail. The fundamentals on which it relies are discussed, and then its basic characteristics and typical procedures are presented. The discussion focuses on the different ways the technique can be used to obtain experimental heat transfer information. Finally, a large number of papers on the subject in the heat transfer literature are briefly discussed to illustrate the potential of the technique.

Keywords: *naphthalene sublimation technique, heat and mass transfer analogy*

INTRODUCTION

The naphthalene sublimation technique is an experimental technique employed to determine heat transfer coefficients in convection flows. The basic characteristic of the technique is that the heat transfer problem to be investigated is replaced by an analogous mass transfer problem. In the laboratory, only mass transfer experiments are performed, and then heat transfer results are obtained by exploring the concept of analogy between heat and mass transfer. Naphthalene ($C_{10}H_8$) is employed in the mass transfer experiments because of some of its properties, such as the fact that it sublimes at room temperature, its low toxicity, and its good casting and machining properties.

This paper is intended to be an introductory text on the naphthalene sublimation technique. It is subdivided into three main sections. In the first section, the analogy between heat and mass transfer is discussed. The second section deals with the peculiarities of the naphthalene sublimation technique, and the third section reviews some heat transfer problems investigated using the technique.

THE ANALOGY BETWEEN HEAT AND MASS TRANSFER

In this section, the fundamentals of the analogy between heat and mass transfer processes are outlined. A thorough understanding of this analogy is important for a proper application of the naphthalene sublimation technique.

Pure Diffusion

Heat diffusion in solids and in quiescent fluids is related to the temperature distribution in the medium by means of Fourier's law [Eq. (1)]. This law states that heat diffusion is driven by a temperature gradient and its intensity depends upon the thermal conductivity of the medium where it takes place.

$$\mathbf{q} = -\kappa \operatorname{grad} T \tag{1}$$

In a similar manner, diffusion of mass in solids and in quiescent fluids is related to the mass-fraction field in the medium by means of Fick's law [Eq. (2)], which says that mass diffusion occurs in a two-component mixture due to a mass-fraction gradient of one of the components in the mixture. Furthermore, mass diffusion also depends on the mass diffusivity of one component in the other.

$$\mathbf{j} = -\rho \mathscr{D}_{AB} \operatorname{grad} \omega_A \tag{2}$$

Heat and mass diffusion are transport processes that originate from molecular activity. In both cases, the aforementioned laws [Eqs. (1) and (2)] are widely accepted for most situations of practical interest. Moreover, the two rate equations display identical forms, the heat flux $\mathbf{q}$ corresponding to the mass flux $\mathbf{j}$, temperature T corresponding to mass fraction w_A, and so forth.

Forced-Convection Flows

For heat transfer to laminar flows, the principle of conservation of energy yields, for constant-property fluids in the absence of heat sources,

$$\frac{dT}{dt} = \alpha \operatorname{div}(\operatorname{grad} T) \tag{3}$$

where $d(*)/dt$ stands for the material derivative of $*$, defined as $d(*)/dt \equiv \partial(*)/\partial t + \mathbf{v} \cdot \operatorname{grad}(*)$. Equation (3) states that the rate of change of temperature experienced by any material particle of the flowing fluid is due to heat diffusion at its instantaneous position.

Similarly, for laminar flow of a fluid mixture of two nonreactant species A and B, the following equation is obtained if the principle of mass conservation is evoked for species A:

$$\frac{d\omega_A}{dt} = \mathscr{D}_{AB} \operatorname{div}(\operatorname{grad} \omega_A) \tag{4}$$

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Experimental Thermal and Fluid Science 1991; 4:510–523
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0894-1777/91/$3.50

The meaning of the above equation is analogous to that of Eq. (3), stating that the rate of change of mass fraction experienced by a given material particle of the flowing mixture is due to mass diffusion at its instantaneous position.

Considering boundary conditions of known temperature T_w for Eq. (3) and of known mass fraction ω_{Aw} for Eq. (4), dimensionless dependent variables may be defined as

$$\theta = \frac{T - T_f}{T_w - T_f} \quad \text{and} \quad \Omega_A = \frac{\omega_A - \omega_{Af}}{\omega_{Aw} - \omega_{Af}} \tag{5}$$

The subscript f in the definitions indicates a characteristic value of the subscripted quantity in the flow. Equations (3) and (4) can be rewritten in a dimensionless form if Eqs. (5) are used together with an appropriate characteristic length.

The dimensionless versions of Eqs. (3) and (4) have identical forms, as well as the corresponding boundary conditions. Choosing an appropriate way of making these equations dimensionless, the Reynolds number Re is found in both the energy and mass-fraction equations. The Prandtl number Pr appears in the energy equation and corresponds to the Schmidt number Sc that appears in the mass-fraction equation.

Therefore, the dimensionless temperature field depends on the Reynolds and Prandtl numbers, whereas the dimensionless mass-fraction field is a function of the Reynolds and Schmidt numbers. Furthermore, for the same geometry and flow conditions, the functional relationship describing the two fields is the same.

Consequently, the boundary values of the temperature and mass-fraction gradients, which are directly related to the boundary fluxes, are also analogous to each other. These fluxes at the boundary are usually represented by the Nusselt number Nu for heat transfer and by the Sherwood number Sh for mass transfer. Evidently, for the same geometry and flow conditions, Nu and Sh are analogous to each other, and if Pr is equal to Sc, then Nu also equals Sh.

$$\text{Nu} = f(\text{Re}, \text{Pr}); \qquad \text{Sh} = f(\text{Re}, \text{Sc}) \tag{6}$$

It is not difficult to show that all of the above conclusions, obtained for known temperature or mass fraction at the boundary, also hold for other types of boundary conditions. Moreover, similar arguments may be used for turbulent flows, and the conclusions drawn also coincide with the ones discussed above, provided it is assumed that the eddy diffusivities of energy and mass are equal. There exists strong experimental evidence that this is a correct assumption.

Buoyancy-Driven Flows

For buoyancy-driven flows, the analogy between heat and mass transfer also exists for situations where the Boussinesq hypothesis is applicable, that is, when density gradients are very small but large enough to drive the flow. The Boussinesq hypothesis allows substitution of the buoyancy term $g(\rho_{\text{ref}} - \rho)$ that appears in one of the components of the momentum equation by either $\rho_{\text{ref}} g\beta(T - T_{\text{ref}})$ or $\rho_{\text{ref}} g\beta'(\omega_A - \omega_{A,\text{ref}})$, depending upon whether a temperature or a mass-fraction gradient is responsible for the density gradient. β and β' are expansion coefficients, respectively defined as $-(1/\rho)(\partial\rho/\partial T)_p$ and $-(1/\rho)(\partial\rho/\partial\omega_A)$. Either of these expressions is readily obtained by expanding into a Taylor series around ρ_{ref} and neglecting higher order terms. Therefore, the governing equations for heat transfer are again analogous to the ones for mass transfer. However, instead of the Reynolds number, which arises in forced-convection flows, the relevant parameters for buoyancy-driven flows are the Rayleigh number Ra for heat transfer problems and the modified Rayleigh number Ra_* for mass transfer problems. Hence, following the same reasoning employed before for the forced-convection flows, it becomes clear that

$$\text{Nu} = f_*(\text{Ra}, \text{Pr}); \qquad \text{Sh} = f_*(\text{Ra}_*, \text{Sc}) \tag{7}$$

For the same geometry, in the above expressions Nu corresponds to Sh, Ra corresponds to Ra_*, Pr corresponds to Sc, and the functional relationship f_* is the same for both transfer processes.

As a last comment about the heat/mass transfer analogy for buoyancy-driven flows, it is worth mentioning briefly the fundamental difference that exists between forced-convection and buoyancy-driven flows, and its consequences as far as the analogy is concerned.

The velocity field is not affected by heat or mass transfer in forced convection flows. Furthermore, heat and mass transfer may occur simultaneously without any effect whatsoever on each of the processes, as long as the boundary conditions of one of the processes is not affected by the other.

In buoyancy-driven flows, however, the velocity field is temperature- and mass-fraction-dependent. Hence, in the case where heat and mass transfer occur simultaneously, the two processes will affect each other, and the analogy between them is lost. In this case, an additional condition for the heat/mass transfer analogy to exist is that the heat transfer process involves no mass transfer and vice versa; that is, no heat transfer occurs in the corresponding mass transfer process.

Variable-Property Fluid Flows

For flows of variable-property fluids, the requirement for the heat/mass transfer analogy to exist is that the fluid properties should display similar dependence on temperature and on mass fraction. This is a rather restrictive requirement, in most cases being fulfilled only in narrow ranges of the governing parameters. Therefore, the analogy has been employed to a much lesser extent when fluid property variation is important.

Finally for this section, it is important to point out that the analogy exists only in the cases of no heat sources in the heat transfer process and no chemical reactions in the mass transfer process, regardless of the physical situation under study. Moreover, heat diffusion due to gradients of quantities other than temperature should be negligible, and the same should be true for mass diffusion due to gradients of quantities other than mass fraction. These restrictions were implicitly imposed in all physical situations discussed in this section.

THE TECHNIQUE

When an experiment is planned, one of the preliminary decisions to be made is the choice of experimental technique. If heat transfer coefficients are the results sought, the naphthalene sublimation technique may be an excellent alternative. In what follows, some typical procedures and characteristics of the naphthalene sublimation technique are presented and discussed.

Outline of the Technique

The basic characteristic of the naphthalene sublimation technique is that it employs mass transfer experiments to study heat transfer situations. The heated boundaries of the actual problem of interest are substituted by solid naphthalene boundaries in the laboratory simulation, and air is employed as the flowing fluid.

As air flows adjacent to the solid naphthalene surface, sublimation occurs and mass is transferred from the wall to the airstream, in the same way that heat would be transferred from the heated surface to the flowing fluid in the heat transfer situation under investigation.

Strictly speaking, the transverse velocity that occurs at the subliming surface due to mass transfer has no counterpart in the heat transfer problem. Therefore, different flow conditions occur in the two processes, and, in principle, the two processes cannot be analogous. However, this transverse velocity is typically many orders of magnitude smaller than the characteristic velocities of the airflow, and its effect on the mass transfer results has been shown [1] to be negligible.

After the mass transfer data are obtained in the laboratory, the sought-for heat transfer results are obtained by evoking the analogy between heat and mass transfer mechanisms.

There are two types of thermal boundary conditions that are easily simulated in the laboratory with the aid of the naphthalene sublimation technique. First, an isothermal surface in a given heat transfer situation is simulated by a surface of solid naphthalene whose mass fraction is also uniform. This absence of mass-fraction gradients at the subliming surface is achieved when the temperature of the subliming wall is uniform. This is so because the mass fraction of naphthalene vapor at the mass transfer boundary is a sole function of temperature, as will be discussed later in this text. Second, an adiabatic boundary in a heat transfer problem is perfectly simulated by any nonsubliming surface, for example, a metallic surface, since it does not exchange mass with the airflow.

Typical sublimation experiments are now briefly described. First, the naphthalene pieces are weighed and then assembled in the experimental apparatus in their respective run positions. Following this, the experiment is started, which is characterized by the exposure to air of the naphthalene surfaces. After an appropriate period of exposure, the run is stopped. Then the naphthalene parts are removed from the test apparatus and weighed again. These measurements are made to determine the mass losses of the naphthalene surfaces, which, together with the time duration of the run, allow calculation of the total mass transfer rate from each of the surfaces.

Experimental Apparatus

A typical apparatus employed in naphthalene sublimation, forced-convection experiments consists basically of an open-loop flow circuit. The circuit includes a test section, where the naphthalene pieces are exposed to the airstream, a flow- or velocity-measurement section, and a blower.

The flow circuit is normally operated in the suction mode, and the test section is located at the upstream end of the circuit. The choice of such an arrangement guarantees that the flowing air will not be heated or contaminated with lubricating oil in the blower before it reaches the test section.

In typical applications of the technique, it is desirable to make certain that the air entering the flow circuit is free of naphthalene vapor. In other words, it is convenient that the air entering the test section has its bulk mass fraction of naphthalene vapor equal to zero. The main reason for such practice is to avoid complicated mass-fraction measurements in determining the air inlet conditions. To achieve this zero-mass-fraction inlet condition, it is crucial that the environment from where fresh air is drawn into the flow circuit has no connections with the environment to where air containing naphthalene vapor is exhausted.

In natural convection experiments, the experimental apparatus acquires different characteristics. The test section with the naphthalene pieces is normally held by a frame that often allows change of orientation of the active surfaces with respect to the driving body force field. In situations of external or open-cavity flows, protection against stray air currents must be provided, such as a protecting enclosure or a surrounding channel. This protection must be designed such that possible changes in freestream concentration of naphthalene vapor are kept to a minimum; otherwise the mass transfer rates may be seriously affected. Due to the typically low mass transfer coefficients, enough sublimation is achieved only with data runs that last several hours. Hence, to prevent significant changes in the mass-fraction boundary condition during the experiments, the air temperature should be controlled very tightly.

My experience with natural convection experiments using the naphthalene sublimation technique is that they are extremely difficult to control; actually, in my attempts I could never obtain a satisfactory repeatability. Perhaps this explains the absolute dearth of reports on natural convection experiments using this technique in the literature.

The Naphthalene Surfaces

The naphthalene surfaces are often fabricated by using casting techniques. This method of fabrication is normally very successful, resulting in naphthalene pieces of excellent quality.

The molds employed in the casting procedure generally consist of several pieces (often metallic, brass or aluminum, or made of silicone rubber) that are assembled together, forming a cavity whose shape coincides with the shape of the piece to be fabricated. Molten naphthalene is poured into the mold cavity and allowed to solidify. The melting point of naphthalene is 80.3°C, but superheating of the molten naphthalene is often needed to allow complete filling of the mold cavity prior to the occurrence of solidification. In cases of mold cavities that display very narrow gaps and passages, preheating of the mold may also be necessary.

It is generally desired that the naphthalene surfaces that will interact with the airflow have a glasslike smoothness. This may be easily accomplished by polishing the corresponding surfaces of the mold cavity. This practice also facilitates disassembly of the mold after the solidification of the naphthalene piece.

In many instances, it is convenient that one of the pieces of the mold also be part of the assembly at the test section, serving as a shell, frame, or substrate for the naphthalene piece. In such cases, the surfaces in contact with the naphthalene should be rough in order to provide good adherence. These shells are especially convenient when they can be devised to cover all the naphthalene surfaces that are not intended to be active as far as mass transfer is concerned, thus avoiding extraneous sublimation during the experiments.

The maximum size of a naphthalene piece is often dictated by its weight, which should be lower than the capacity of the balance available. If the naphthalene part includes a shell or substrate, its size is further limited because its weight per unit volume is increased. On the other hand, the balance employed to perform the mass measurements should present high resolution, since rather small mass variations of the subliming pieces are typically encountered. Because balances with high resolution have small capacities, generally the naphthalene parts cannot be too heavy. Analytical balances with a 200 g capacity and a smallest scale reading of 0.1 mg have been widely used in naphthalene experiments.

It is worth noting, however, that a desired naphthalene surface may be formed by assembling together in the test section several small and independent pieces, each one being light enough to allow weighing with the balance available. Since each of the pieces is weighed individually, additional information about quasi-local mass transfer coefficients is also obtained with this procedure.

Sometimes the naphthalene parts are fabricated by means of a process that involves both casting and machining operations. This procedure is used chiefly when the desired naphthalene piece consists of a substrate coated with a very thin layer of solid naphthalene. In such situations, a naphthalene layer is cast on the substrate and then machined to the desired thickness. The surface quality obtained from the machining operation is normally excellent.

In applications where the geometry of the model is too complex, the casting and/or machining procedure for obtaining the naphthalene surface may not be practical. In these cases, an alternative way of manufacturing the naphthalene surface is by spray coating. Neal [2] developed a spraying technique that uses a solution at 20% concentration of naphthalene in a solvent called Inhibisol (manufactured by The Penetone Co. Ltd., Cramlington, Northumberland, U.K.). A similar spraying technique has been developed by Lee [3], who uses molten naphthalene in a spray system equipped with heaters for the compressed air as well as for the entrainment air around the jet. However, the surface obtained with the spray technique is not as smooth as the ones obtained with the casting/machining technique. Depending upon the application sought, this may be a problem.

Time Duration of Runs and Weight Uncertainty

The time duration of a data run should be as long as possible to allow plenty of sublimation from the active surfaces, in order that the mass transferred can be determined within a reasonable level of uncertainty. On the other hand, too much sublimation will result in changes of shape of the naphthalene surfaces, which may modify the flow pattern, change the mass transfer area, and cause other undesirable effects.

Therefore, a decision should be made about the tolerable level of geometry change prior to deciding about the run duration. How much change is tolerable will depend on the situation at hand, but typically the change in surface elevation should not exceed a few ten-thousandths of the characteristic length of the problem during a data run [4]. Particular care should be taken in observing the sharp edges of steps, where the local sublimation rates are typically large and hence the geometry change may be critical. Moreover, in many cases a slight rounding off of a sharp corner is enough to change the flow pattern significantly, with direct implications for the mass transfer rates.

Once the maximum change of geometry allowed is chosen, it is possible to calculate the corresponding decrement in volume of solid naphthalene due to sublimation. With knowledge of the density of solid naphthalene ($\rho_s = 1.146 \times 10^3$ kg/m^3), the corresponding allowable loss of mass follows directly. Finally, some preliminary runs should be executed to determine the appropriate durations of the data runs such that the mass losses (and hence geometry changes) are kept within the desired limits.

A low uncertainty in the mass-loss measurements of the naphthalene pieces is strongly desired because they are directly employed to determine the mass transfer coefficients. In this connection, the estimate of allowable mass loss obtained with the criterion described above may also be employed to determine the necessary uncertainty on the weight measurements such that the transfer coefficients are determined within the desired uncertainty level.

The Mass Transfer Coefficients

Once the mass transfer data have been obtained in the laboratory, the first step toward the determination of the mass transfer coefficients is to calculate the mass transfer rate of each naphthalene piece, given by

$$\dot{m} = \Delta m / \tau \tag{8}$$

where Δm is the total mass loss of the subliming surface during the data run and τ is the time duration of the run.

The occurrence of mass losses that are extraneous to the data run is often observed. These losses are normally due to natural convection that occurs during the operations of weighing, mounting, and dismounting the naphthalene pieces.

A good estimate for these extraneous losses can be obtained for a given data run by performing, just after the experiment, an auxiliary run whose time duration is null. In other words, the weighing, mounting, dismounting, and reweighing operations are performed exactly as during the actual data run, but the run itself is not executed. The mass loss observed during this so-called after-run is subtracted from the total mass sublimed during the actual experiment. Then the corrected mass loss is used in Eq. (8) to determine the total mass transfer rate at the surface.

Since the above procedure yields an estimate of the undesired mass losses only, it is good practice to ensure that the mass loss during the data run is much larger than the correction for extraneous losses.

The mass transfer coefficient is defined as

$$K = \frac{\dot{m}/A}{\rho(\omega_{nw} - \omega_{nf})} = \frac{\dot{m}/A}{(\rho_{nw} - \rho_{nf})} \tag{9}$$

In this expression, ω_{nw} is the mass fraction of naphthalene vapor at the subliming wall, whereas ω_{nf} stands for a characteristic value of the mass fraction in the flowing fluid. As shown in Eq. (9), the mass transfer coefficient can also be defined in terms of the mass concentrations, which are directly related to the mass fractions ($\rho_n = \rho\omega_n$).

The ideal gas law can be employed to relate the mass concentration of naphthalene vapor with its partial pressure and temperature because the partial pressures of naphthalene vapor are always very low.

$$\rho_n = p_n / RT \tag{10}$$

In (10), R is the ideal gas constant for naphthalene vapor, equal to 64.87 J/(kg · K).

At the air–solid naphthalene interface, the solid naphthalene is assumed to be in equilibrium with its vapor, and hence the partial pressure of the naphthalene vapor at the wall is a function of the local temperature alone. Sogin [5] used a function of the type

$$\log_{10} p_{nw} = B_1 - B_2 / T_w \tag{11}$$

to fit the experimental data of Thomas [6], who measured the vapor pressure of naphthalene as a function of temperature in the range 0–80°C. Using SI units and switching to natural logarithms, Sogin's equation becomes

$$\ln p_{nw} = 31.23252 - 8587.36 / T_w \tag{12}$$

Sherwood and Bryant [7] measured the vapor pressure of naphthalene by using broken pieces of cast material rather than finely divided crystalline solids. The equation that represents their data well in the range of 0–38°C is

$$\ln p_{nw} = 31.48763 - 8669.23 / T_w \tag{13}$$

Equation (13) gives values for p_{nw} that are slightly lower than those given by Eq. (12); at 20°C, the deviation is 2.4%.

The mass concentration of naphthalene vapor at the subliming wall can be determined as a function of the local wall temperature upon combination of Eqs. (10) and (13) [or (12)]. Therefore, as mentioned earlier, a uniform mass fraction at the wall is achieved when the temperature of the naphthalene surface is also uniform. The foregoing discussion applies regardless of the presence of humidity in the flowing air, since Sparrow and Niethammer [1] showed that the effect of humidity on mass transfer can be neglected for naphthalene experiments.

The characteristic value of the mass concentration in the fluid, ρ_{nf}, is often the bulk value for internal flows and the free-stream value for external flows. In both cases, ρ_{nf} can be easily evaluated, provided that there is no naphthalene vapor in the air approaching the upstream end of the naphthalene test section.

For external flows, ρ_{nf} is simply equal to zero when there is no naphthalene vapor outside the boundary layer. For internal flows, the bulk value of the mass concentration at a given streamwise location x is determined by computing all the mass of naphthalene vapor that is delivered to the airflow per unit time, from the duct entrance up to the location x. Hence,

$$\rho_{nf}(x) = \rho \dot{m}(x) / \dot{W} \tag{14}$$

where $\dot{m}(x)$ is the total mass transfer rate due to sublimation from the duct inlet up to the x streamwise location, and $\dot{W}$ is the mass flow rate through the duct. For practical purposes, $\dot{W}$ is constant throughout the duct and equal to the airflow rate, since the total mass of naphthalene vapor added to the airstream per unit time is typically not greater than 0.02% of the airflow rate.

The mass transfer coefficient is often presented in the form of the Sherwood number, defined as

$$\mathrm{Sh} = KL / \mathscr{D} = KL\,\mathrm{Sc} / \nu \tag{15}$$

The mass diffusivity of naphthalene vapor into air was measured by Mack [8], who obtained $\mathscr{D} = 6.12 \times 10^{-6}$ m^2/s at 1 atm and 25°C. At the same temperature and pressure, the air kinematic viscosity is $\nu = 1.55 \times 10^{-5}$ m^2/s. Hence, the corresponding Schmidt number is equal to 2.53. Sherwood and Träss [9] presented an equation that gives the Schmidt number as a function of temperature, namely, $\mathrm{Sc} = 7.00 / T_w^{0.185}$. At 25°C, this equation gives Sc = 2.44. Sogin [5] recommended a value of 2.5 for Sc, which is often used in naphthalene experiments. Other sources of information about the physical properties of naphthalene are also available in Refs. 10–12.

Local Transfer Coefficients

As mentioned earlier, local-average coefficients can be obtained by designing the test section such that the naphthalene boundaries are formed by several smaller surfaces, each one belonging to an independent piece that can be weighed separately. This procedure gives quasilocal transfer information but may not be practical when local coefficients (rather than a localized average) are desired.

The local rate of mass transfer at any surface location can be evaluated from the local change of surface elevation in conjunction with the duration time of the respective data run. The sublimation depth δ is evaluated by differencing the measured surface elevations before and after a data run. To perform these measurements, a precision dial gauge is needed because the sublimation depths must be kept to a minimum to avoid significant changes in geometry. As the measurements must be made in a large number of locations to determine the distribution of the local transfer coefficient, automation of the measurements is highly desirable.

Therefore, the local mass transfer rate per unit area is given by

$$\dot{m}''(s) = \rho_s \delta(s) / \tau \tag{16}$$

and the local mass transfer coefficient is obtained by means of Eq. (9), where $\dot{m}''$ is used instead of $\dot{m} / A$. In order to check the measurements and procedure, the local mass transfer rate can be integrated along the naphthalene surface and then compared with the total mass transfer rate obtained with the weighing procedure discussed earlier. In applications where the gradient of mass transfer rate along the surface is expected to be very large, precise positioning of the dial gauge is needed; otherwise the uncertainty induced by location errors becomes significant.

An alternative procedure to obtain local transfer coefficients is the thin-film naphthalene mass transfer analogue technique [2], which is simple and inexpensive. This technique consists of using transparent (e.g., Plexiglas) substrates, which are coated with a thin layer of naphthalene (about 0.015 mm thick), deposited with the aid of the spray technique. Local transfer coefficients are determined by observing times taken for the naphthalene to clear from each point. Results obtained with the thin-film technique are free from the location error uncertainty mentioned in the previous paragraph. However, as spray coating is used in conjunction with the thin-film technique, the surface roughness may be an impediment in some applications.

The Heat Transfer Coefficients

As discussed in the preceding section, once the relationship between the Sherwood number Sh and the dimensionless parameter characterizing the flow (Re or Ra_*) has been determined in the laboratory (for Sc = 2.5), the Nusselt number Nu for the same geometry and flow conditions follows directly for Pr = 2.5 [Eq. (6) or (7)]. This value of Pr corresponds to water at approximately 71°C. For other values of the Prandtl number, however, the application of results obtained with the naphthalene sublimation technique in heat transfer problems requires further information about the relationship between Nu and Pr (or between Sh and Sc). This information cannot be obtained with the technique, because data are obtained for only one value of Sc, namely, 2.5.

Fortunately, in most cases, the relationship remains essentially the same for each broad class of heat (or mass) transfer problems. Thus, it is well known that the Nusselt number does not depend on the Prandtl number in fully developed laminar flows inside ducts, regardless of the geometry of the cross section. Hence, for this situation, Sh and Nu are interchangeable.

The Nusselt number for most buoyancy-driven flows of fluids with $Pr > 0.7$ displays only a very mild dependence on Pr. In these cases, it is often possible to use directly the results for Sc = 2.5 in heat transfer situations involving air (Pr = 0.7) and water Pr = 2–10) with negligible error.

For external flows, flows across tube bundles, turbulent flows inside ducts, and many other forced-convection flows, Nu is often correlated via

$$\mathrm{Nu} = C\,\mathrm{Pr}^n\,\mathrm{Re}^n \tag{17}$$

where n lies between 0.3 and 0.4. The Colburn analogy implies that $n = 1/3$ for $0.6 < \mathrm{Pr} < 60$. This value of n is quite successful in correlating the experimental data available for a wide variety of forced-convection flows. Sparrow and coworkers [13, 14] recommended $n = 0.4$ for turbulent internal flows.

Hence, for these forced-convection flows, conversion from Sh to Nu is achieved via

$$\mathrm{Nu} = (\mathrm{Pr}/\mathrm{Sc})^n\,\mathrm{Sh} \tag{18}$$

where Sc = 2.5 for naphthalene experiments. It is important to emphasize, however, that the Sh $\mapsto$ Nu conversion by means of Eq. (18) cannot be made indiscriminately for all flow situations without previous validation [2, 15]. For other classes of problems, similar conversions can be obtained, provided that the dependence of Nu on Pr is known.

Temperature Depression at the Wall

It is important to emphasize that Eqs. (10) and (13) imply that the mass concentration of naphthalene vapor at the naphthalene wall is quite sensitive to temperature. This can also be illustrated with the aid of the expression

$$\frac{\Delta\rho_n}{\rho_n} = \frac{T}{T+\Delta T}\exp\left[\frac{8669.23\,\Delta T}{T(T+\Delta T)}\right] - 1 \tag{19}$$

Therefore, an accurate evaluation of the naphthalene wall temperatures is of prime importance. For example, if a wall temperature of 20°C is mistakenly considered to be equal to 21°C, the mass concentration at the wall will be overestimated by 10.1%. Furthermore, from Eq. (9) it can be seen that such a mistake will result in a minimum underestimate of 9.2% in the mass transfer coefficient, depending upon the value of ρ_{nf}.

In addition, the sublimation process at the naphthalene wall involves absorption of latent heat from the surroundings ($\lambda = 5.5726 \times 10^5$ J/kg), and hence a temperature depression at the surface is normally observed. Therefore, direct measurement of the surface temperatures is highly desirable. These measurements can be performed with the aid of thermocouples embedded in the naphthalene body with their junctions positioned at the surface.

Sometimes, however, it may not be possible or practical to measure the temperature of the naphthalene surface directly as described above. In these cases, the wall temperature can be estimated as follows [5]. First, it is assumed that the latent heat of sublimation comes only from the air, that is, that the naphthalene surface is adiabatic. This is a conservative assumption that overestimates the temperature depression at the naphthalene surface. Using this hypothesis, a heat balance yields

$$hA(T_f - T_w) = \dot{m}\lambda \tag{20}$$

The heat transfer coefficient h is related to the mass transfer coefficient via the analogy between heat and mass transfer. For example, if a power-law dependence of Nu on Pr can be assumed, Eq. (18) is applicable. Hence, upon elimination of h, Eq. (20) yields a relationship between the mass transfer coefficient and the wall temperature T_w. This relationship is now combined with Eqs. (9), (10), and (13), yielding the desired estimate for T_w. It is not difficult to conclude that the temperature depression thus obtained is, for external flows, a sole function of the temperature level during the experiments, whereas for internal flows it also depends on the flow conditions, inasmuch as they affect the bulk concentration ρ_{nf}.

The wall temperature depression is clearly an undesirable effect, because it may cause temperature gradients at the naphthalene surfaces. Hence, as discussed before, mass fraction gradients at the subliming surface will also be present. It is obvious that these gradients should be avoided when a boundary condition of uniform wall temperature is to be simulated with the naphthalene sublimation technique.

Moreover, wall temperature depression implies that convection heat transfer is occurring simultaneously with mass transfer. Although this fact in itself is unimportant for forced-convection flows, it destroys altogether the analogy between heat and mass transfer for buoyancy-driven flows, as discussed earlier.

Due to naphthalene's low latent heat of sublimation, the temperature depression effect for forced convection can normally be kept within acceptable limits, in the sense that the temperature gradient at the surface should be negligible. For natural convection flows, however, temperature depression may be a serious problem, because the temperature difference $T_f - T_w$ may provoke a driving force for the flow of the same order of magnitude as the one due to the mass-fraction difference $\omega_{nf} - \omega_{nw}$.

SOME SUCCESSFUL APPLICATIONS

The naphthalene sublimation technique has been employed to obtain transfer coefficients for a large number of flows and configurations. Investigations aiming at different engineering applications were undertaken with the aid of the technique.

The following subsections provide several examples of applications of the sublimation technique. These illustrate some of the various ways it can be used to obtain basic heat and mass transfer information.

External Flows

Ko and Sogin [16] studied the laminar heat and mass transfer from blunt bodies, aiming at basic data for the design of aircraft equipment. The bodies under study were ellipsoids with axis ratio of 4:1, and the Reynolds number based on the profile length was varied in the range 32,500–280,000.

Christian and Kezios [17] obtained local transfer coefficients for sharp-edged cylinders in axisymmetric laminar flow. This seems to be the first report on measurements of local coefficients with the aid of the naphthalene sublimation technique. The authors obtained the local data by measuring the changes in radii due to sublimation, using a modified lathe and a micrometer dial indicator with a resolution of 2.54×10^{-3} mm. All the measurements were made in an atmosphere nearly saturated with naphthalene vapor, to avoid extraneous losses.

Sogin [18] performed experiments in the laminar flow regime to determine the rates of heat transfer by forced convection from two isothermal spanwise strips in tandem on a flat plate. Sogin and Goldstein [19] investigated this same configuration in the turbulent flow regime. In both investigations, the plate surface was adiabatic everywhere but on the isothermal strips. It is worth noting that it would be extremely difficult to impose this type of boundary condition using heat transfer experimental techniques, owing to unavoidable conduction leaks from the strips to the rest of the plate. However, this boundary condition was accurately and easily imposed with the naphthalene sublimation technique.

The heat transfer characteristics of boundary layer flows over square plates, subjected to both yaw and an angle of attack, were studied by Sparrow and Tien [20]. Local coefficients are presented by Tien and Sparrow [21]. On the basis of the results obtained with the sublimation technique, these authors showed that the transfer coefficients are nearly independent of both the angle of yaw and the angle of attack. They also proposed a correlation for the heat transfer coefficients at the surface, which has several advantages over one previously employed in the evaluation of wind-related losses of the cover plates of flat solar collectors. Similar work for rectangular plates is reported in Ref. 22.

Heat and mass transfer coefficients were obtained for an inclined surface elevated above a parallel host surface [23]. The flow separation caused by the steplike blockage associated with the elevation was shown to enhance markedly the heat transfer at the surface. The enhancement was accentuated at small angles of attack, at large step heights, and at high Reynolds numbers. The implications of these findings in solar collector analysis are also quite important.

Sparrow and Prieto [24] determined heat transfer coefficients for two spheres contacting each other and immersed in a flowing fluid. The effect of the Reynolds number and angle of attack were investigated. It was found that the largest effects of the sphere–sphere interaction on heat transfer occur when the two spheres are in line. A compression-forming method was developed and employed in the fabrication of the spheres, because the conventional casting and casting-machining methods did not perform satisfactorily.

The region adjacent to the attached end of a wall-attached cylinder in cross flow was studied by Sparrow et al [25]. Quasi-local transfer coefficients were determined with a small disklike sensing element positioned at various axial stations along the cylinder. It was found that the wall–cylinder interactions were confined to a region extending about one cylinder diameter from the wall. Local measurements in this region were performed by Goldstein and Karni [26], who detected local peaks of heat transfer (up to 700% of the free-stream value) in a narrow span extending from the wall to about 0.066 diameter above it. Heat transfer coefficients at the wall on which the cylinder is attached were also measured by Goldstein et al [27]. A computer-controlled automated data acquisition system was employed to measure, accurately and rapidly, local coefficients at 1025 discrete locations on the wall at the region adjacent to the base of the cylinder. More local coefficients at both the cylinder and the end wall are found in Ref. 28, and results for a square cylinder are reported in Ref. 29. Local transfer coefficients were measured by Goldstein and Spores [30] at 6387 locations on the end wall in the region between adjacent turbine blades, using an automated system similar to the one described in Ref. 27.

The effect of yaw on heat transfer was investigated for a cuplike cavity facing a free-stream flow [31]. The effect of the cavity depth was also studied, and the heat transfer coefficients were found to be always lower than the no-yaw, zero-depth cavity. Local results for the no-yaw cavity are reported by Sparrow and Geiger [32]. For the zero-depth case, a disk facing a uniform oncoming flow, these same authors [33] developed a correlation for the average Nusselt number that resolved a formerly existing disparity in the literature.

The sublimation technique was used by Goldstein and Taylor [34] to measure local mass transfer coefficients in the neighborhood of a row of jets entering a cross flow, simulating the heat transfer that occurs on a film-cooled wall. Air was blown into the main turbulent boundary layer flow from a row of holes inclined to the surface and aligned in a streamwise location. The transfer coefficient in the neighborhood of the holes was shown to be significantly increased by the jets, especially immediately adjacent to the holes and also some distance downstream of the centerlines of the holes. Webster and Yavuzkurt [35] performed similar experiments, measuring transfer coefficients in the recovery region of a film-cooled surface.

Flows in Ducts and Channels

Local transfer coefficients were determined for the entrance region of the laminar flow in a parallel-plate channel whose inlet is partially constricted [36]. The effects of flow separation, reattachment, and redevelopment on the transfer coefficient were carefully studied with the aid of the sublimation technique.

The transfer coefficients for a wall-attached, transverse blockage plate in a flow inside a square duct were measured, as reported in Ref. 37. It was found that the coefficients at the plate are quite insensitive to the extent of the blockage. Furthermore, the difference between the coefficients at the upstream and downstream faces diminishes with the Reynolds number, the upstream-face coefficients being generally higher. Similar experiments were performed by Sparrow et al [38], with the transverse blockage centered in the duct.

Sparrow and Cur [39] determined local heat transfer coefficients for turbulent flow at the entrance region of a parallel-plate channel having a sharp-edged inlet. They examined the case of both plates at uniform temperature and the case of one adiabatic plate and the other at uniform temperature. The naphthalene sublimation technique was very effective in showing that the flow separation at the duct inlet plays a decisive role in shaping the axial distribution of the heat transfer coefficient at the entrance region. A high heat transfer peak was observed at the point of flow reattachment. The results obtained for the two different boundary conditions investigated were found to be nearly the same, except for the fact that the entrance length for asymmetric heating is greater than that for symmetric heating.

The laminar developing flow in a parallel-plate channel was studied by Fernandes and Saboya [40]. The simultaneous development condition was examined, and results for Reynolds numbers ranging from 10 to 150 were obtained. The experimental results were compared with analytical predictions, and a very good agreement was observed.

The effects of flow maldistribution caused by partial blockage of the inlet of a parallel-plate channel were studied by Sparrow and Cur [41]. Local heat transfer coefficients on the plates were determined for two different blockages. It was found that large spanwise nonuniformities on the transfer coefficient are induced by the flow maldistribution, which persist to locations far downstream.

Axial and circumferential distributions of the heat transfer coefficient were measured in a tube in which the entering airflow is skewed due to competition between the test section tube and a parallel tube, both drawing air from the same plenum chamber [42]. The effect on the transfer coefficient of the intertube spacing and of the flow imbalance between the tubes was investigated. It was found that the effects of the flow imbalance are very mild and are significant only for small spacings.

Sparrow and Molki [43] determined local heat transfer coefficients for turbulent flow in a tube for two different sharp-edged inlet conditions, namely, with and without a baffle plate. The distributions of the local coefficients for the two conditions were shown to differ significantly.

Heat transfer in bends of circular cross section were investigated by Sparrow and Chrysler [44]. Results for turning angles of 30°, 60°, and 9° were obtained, and the Reynolds number was varied between 500 and 1×10^5. The bends used in the mass transfer experiments consisted of a succession of mass transfer modules, each a miniature bend of 5°. Each module was made up of two parts, one serving as the outer wall of the bend and the other as the inner wall. In this manner, quasi-local transfer coefficients were determined, and it was found that the coefficients at the outside of the bend exceed those at the inside of the bend, but the deviations decrease as the Reynolds number increases. The inlet conditions were shown to play a major role on the streamwise distribution of the local transfer coefficient. Transport coefficients for the straight tube situated downstream of the bend have also been determined [45].

An empirical correlation was developed for the prediction of average Nusselt numbers in circular tubes with simultaneous development of velocity and temperature profiles and with sharp-edged inlets [46]. The proposed correlation is applicable to tubes as short as two diameters.

Heat transfer in corrugated channels has been extensively studied by Goldstein and Sparrow [47], who obtained local and average coefficients for laminar, transition, and turbulent flow regimes. Molki and Yuen [48] investigated the effect of interwall spacing on heat transfer in a corrugated channel.

Local and average heat transfer coefficients were obtained for the downstream face of an abrupt enlargement in a pipe [49]. Sparrow and Misterek [50] obtained mass transfer coefficients at the base of a cylindrical cavity in the floor of a flat duct. The effect of the cavity depth on the transfer coefficient was studied, and two local maxima were observed, one at a depth of 6% of the cavity diameter, and the other at a depth approximately equal to the cavity radius. The first maximum is due to the reattachment of the shear layer on the cavity base, whereas the second remains to be explained.

The turbulent flow in a flat rectangular duct with streamwise nonuniform heating at one of its walls was investigated by Sparrow et al [51]. The nonuniformity of the thermal boundary condition at the wall consisted of adiabatic zones periodically inserted between isothermal heated zones. It was found that the presence of adiabatic zones can give rise to substantial enhancement of the heat transfer coefficients at the heated zones.

The mass transfer characteristics of an assembly of partially segmented (louvered-finned) plates in a rectangular duct have been investigated experimentally with the naphthalene technique [3]. Owing to the complexity of the geometry, the naphthalene surfaces were obtained by means of the spray-coating technique. It was found that there are optimum Reynolds number values as far as the mass transfer rate is concerned that depend on the geometrical parameters.

The axial distribution of the heat transfer coefficient downstream of an abrupt contraction in a flat rectangular duct was determined with the aid of the sublimation technique [52]. The contraction was due to a forward-facing step in one of the walls of the duct. The duct wall that constituted the continuation of the step was heated, whereas all the other duct walls were adiabatic. The experiments showed that the contraction caused the coefficient to increase at first in the streamwise location, attain a maximum, and then decrease monotonically to a fully developed value.

Han et al [53] obtained distributions of mass transfer coefficients around sharp turns in two-pass smooth and rib-roughened channels. This configuration is found in cooling passages of modern gas turbine blades. The results showed that the ribs enhanced the transfer on the surfaces around the turn.

Natural Convection

Sparrow and Bahrami [54] reported natural convection experiments in a vertical parallel-plate channel with three different conditions along the lateral edges: (1) both edges open to ambient, (2) blockage of one of the lateral gaps, and (3) blockage of both lateral gaps. The experiments were conducted in a test chamber with insulated walls situated in a windowless, temperature-controlled laboratory. To further avoid the influence of extraneous air currents on the measurements, a system of baffles and screens was employed to shield the apparatus in the chamber. The results showed that the conditions along the lateral edges were important only for small plate spacings. They report similar work in Ref. 55, where natural convection experiments on the fins of a finned horizontal tube are described.

Fins and Heat Exchanger Configurations

Saboya and Sparrow determined local and average transfer coefficients on the fins of one-row [56, 57], two-row [58], and three-row [59] plate fin-and-tube heat exchangers. In the last two reports, the tubes were arranged in staggered rows. The contour of the naphthalene surfaces was measured by a precision dial gauge whose resolution was about 0.001 mm. The gauge was mounted on a fixed strut that overhung a movable coordinate table, which enabled the surface to be independently traversed in two perpendicular directions in the horizontal plane. Contour measurements at as many as 25 discrete locations were made. As opposed to the procedure reported in Ref. 17, the measurements were not carried out in an atmosphere nearly saturated with naphthalene vapor. Rather, the contour changes due to extraneous losses were estimated in after-runs and corrections were applied to the measured depths. It was shown that the distribution of the transfer coefficients at the fin surface is far from being uniform because of hydrodynamic effects such as developing boundary layers, vortices, and wake regions. A similar study was performed by Lau et al [60], where the effects of pin configuration, pin-to-diameter ratio, and entrance length were investigated. It is worth noting that the results reported in the above papers pertain to fin efficiencies equal to unity, since the naphthalene sublimation technique simulates isothermal surfaces. Rosman et al [61] used the results of Saboya and Sparrow [58], together with additional heat transfer data, to determine temperature distributions on the fins as well as their efficiencies for different fin materials. They also showed that Eq. (18) is applicable to this situation, with $n = 0.4$.

Results for plate heat exchangers were obtained by Cur and Sparrow [4, 62], who performed experiments to determine coefficients for each plate of a collinear array of two or more plates. The plates were aligned parallel to the flow direction and were situated in an airflow in a flat rectangular duct. The effects of the plate thickness and interplate spacing were studied. Among other conclusions, they showed that the spacing-to-length ratio of unity is not necessarily optimal as far as heat transfer is concerned. For an array of staggered plates aligned parallel to the flow, Sparrow and Hajiloo [63] observed entrance region effects only on the first row. Lee [64] studied arrays of plates aligned at various angles to the flow and determined plate angles of maximum heat transfer as a function of the Reynolds number.

Transfer coefficients for arrays of pin fins deployed along one of the principal walls of a flat rectangular duct were obtained by Sparrow and Ramsey [13] for staggered arrangements and by Sparrow et al [14] for inline arrangements. For both arrangements, the row-by-row transfer coefficients were found to attain a constant fully developed value after the fourth row. Under conditions of equal pumping power and equal heat transfer area, the inline array transfers more heat than the staggered array, whereas at fixed heat load and fixed mass flow rate, the staggered array requires less heat transfer surface than the inline array. In a similar study, Chyu [65] obtained heat transfer coefficients for both inline and staggered arrays of pin fins with endwall fillets, which are common in turbine cooling applications. The results showed that the pin-endwall fillet is undesirable as far as heat transfer augmentation is concerned, and its effect is more pronounced in the staggered arrangement.

Experiments were performed to investigate the response of heat transfer in a tube bank to (1) maldistribution of the flow at the inlet cross section [66], (2) an abrupt upstream enlargement of the flow cross section [67], and (3) an upstream right-angle band [68]. Per-tube heat transfer coefficients were measured at different rows along the bank. Effects of the maldistribution are observed as far downstream as the twelfth row. The enlargements can give rise to appreciable increase in the tube-bank transfer coefficients compared with those for the no-enlargement case. The effect of the right-angle bend on the transfer coefficients of the tubes was found to be very mild. Merker and Hanke [69] determined per-tube transfer coefficients for tube banks of oval tubes. The naphthalene tube was fabricated by pressing naphthalene powder in dividable press molds. The flow circuit was operated in the pumping mode and was equipped with a heating section to control the temperature of the airstream. It was found that exchangers with oval tubes perform better when the pumping power is an important constraint.

The shellside heat transfer of a shell-and-tube heat exchanger was investigated by Sparrow and Perez [70]. A model heat exchanger with an array of 92 tubes was fabricated to be totally free of leakage paths and to have negligible bypass effects. It was found that the per-tube heat transfer coefficients vary on the order of a factor of 2 within a given compartment of the exchanger, giving rise to a nonuniform thermal loading of the tubes. It was also demonstrated that the per-tube coefficients are streamwise periodic.

Heat Transfer Enhancement

The effect on the local transfer coefficients of transverse rod-type disturbance elements situated adjacent to one principal wall of a flat rectangular duct was investigated by Sparrow and Tao [71]. Two types of boundary conditions were analyzed: (1) heat transfer only at the rodded wall and (2) heat transfer in both principal walls. The axial distributions of the local transfer coefficient revealed the rapid establishment of a periodic fully developed regime, characterized by a cyclic axial distribution of the transfer coefficients. The cycle-average, fully developed transfer coefficients were found to be much larger than those for a smooth-walled duct. The case of ducts with both principal walls equipped with disturbance rods was also studied by Sparrow and Tao [72], who found transfer coefficients that exceed by about 40% those for ducts with disturbance rods in only one wall. They also showed that the symmetrically disturbed duct displays better heat transfer performance than the asymmetrically disturbed duct under a wide range of conditions.

The heat transfer and enhancement characteristics of periodically converging–diverging tubes have been studied for turbulent flow conditions [73]. The tubes consisted of a succession of alternately converging and diverging conical sections (ie, modules) placed end to end. The taper angle and module aspect ratio were varied parametrically, as well as the Reynolds number. A performance analysis comparing periodic tubes and conventional straight tubes revealed, for equal pumping power and transfer surface area, enhancements in the 30–60% range, the tubes with the half taper angle equal to 10° displaying the best performance. No enhancement is obtained for laminar flow, as reported in Ref. 74. A similar study was performed for tubes equipped with internal transverse ribs [75], and enhancements of up to 100% were observed when compared to smooth tubes under the constraints of equal pumping power, internal diameter, and axial length.

Impinging Jets

Local transfer coefficients were measured on a surface on which a circular jet impinges at an oblique angle [76]. Angles ranging from 9° (normal impingement) to 3° were investigated. The effect of the distance between the jet orifice and the surface was also studied, as was the effect of the jet Reynolds number. A displacement of the point of maximum mass transfer from the geometrical impingement point was observed when the jet inclination was increased. The largest inclination-induced decreases in the maximum transfer coefficient and in the surface-averaged transfer coefficient were in the 1.5–2% range, showing that neither of the quantities is highly sensitive to the inclination of the jet.

Ward and Mahmood [77] used the thin-film technique [2] to obtain local mass transfer coefficients on a flat surface on which a swirling jet impinged orthogonally. Both the swirl intensity and the nozzle-to-target distance were varied parametrically, and a correlation for the average Nusselt number was presented. It was found that the transfer rate decreases as the swirl intensity increases, and that the position of maximum heat transfer is displaced from the central axis of the jet for high swirl intensity.

Sparrow et al [78] determined quasi-local transfer coefficients on the surface of a cylinder on which a jet impinged in cross flow. The jet Reynolds number, the distance between the jet orifice and cylinder, and the jet diameter were varied parametrically in the course of the experiments. It was found that the axial distribution of the transfer coefficient peaked at the impingement point and that this peak value increased with decreasing jet diameter and with decreasing distance between the jet orifice and the cylinder. An investigation on slot jets impinging in cross flow on a cylinder is also reported [79]. In this study, experiments with jets offset from the cylinder were performed, in addition to experiments where the jets were aligned with the cylinder. Local Nusselt number correlations are presented in Refs. 80 and 81 as a function of the angular position along the cylinder perimeter. In these naphthalene experiments, a test rig similar to the one described in Ref. 25 was employed. Another technique that has been used to obtain the local results consists in running different tests where different segments of the naphthalene surface are covered with cellophane tape to prevent sublimation.

Transfer coefficients were measured by Sparrow et al [82] for a confined disk on which a circular jet impinges. The spent air is collected in an annulus that surrounds the jet delivery tube. The motivation for studying this configuration was that it provides precise control of the surface area affected by the jet and also ensures complete collection of the spent air. It was found that for small jet diameters there is an optimum separation distance between the jet origin and the impingement disk.

Rotating Transfer Surfaces

Transfer coefficients from a rotating disk were measured by Kreith et al [83]. The effect on the transfer rate of a stationary surface placed at various distances parallel to the disk was investigated. It was found that the presence of the stationary surface decreased the transfer coefficients at the disk, but this reduction is less than 20% when the distance between the disk and the surface is larger than 5% of the disk diameter.

Koyama et al [84] performed experiments to measure transfer coefficients on inclined flat plates at rotation. Among other observations, they found that when the angle of attack is such that the flow separates from the entire surface, the transfer rate is tremendously increased by rotation.

A cuplike rotating cavity was investigated by Sparrow and Chaboki [85]. Average transfer coefficients were measured for both the cylindrical and base surfaces. The boundary condition of adiabatic base surface was also studied. Three cavity depths were investigated, whereas the rotational Reynolds number was varied from 60 to 9000. Supplementary zero-depth cavity (rotating disk) experiments yielded data that were in excellent agreement with the prediction, with deviations in the 2–3% range.

The heat transfer characteristics of two coaxial cylinders, one rotating, were determined by Lee and Minkowycz [86] using the spray technique described in [3]. Either both surfaces of the cylinders constituting the annulus are smooth, or one is smooth and the other is grooved axially. It was found that the presence of the grooves on the rotating inner cylinder significantly increases heat transfer and that this geometry has potential merits for high-density heat transfer applications.

Experiments were performed to measure transfer coefficients in a passage in a shaft rotating about its own axis [87]. The passage had a circular cross section, and its axis was orthogonal to the shaft axis. Both radial and offset passages were investigated. It was found that offsetting the passage provides significant heat transfer enhancement, and even a slight offset doubles the transfer coefficient.

Transfer coefficients for an array of rotating annular fins were obtained by Sparrow and Preston [88]. Various rotational speeds and interfin spacings were investigated. They observed that closely spaced fins can be used at high rotational speeds without a significant spacing-related decrease in the transfer coefficient. A correlation was developed to facilitate the use of the results for design.

Axial distributions of the transfer coefficients were measured at the end-adjacent portion of the cylindrical surface of a rotating cylinder [89]. It was found that the axial variations of the transfer coefficients are moderate, not exceeding 15%. Moreover, the axial variations attributable to the presence of the free end do not persist beyond the length of one cylinder diameter in the axial direction.

Electronics Cooling

Forced-convection cooling of an array of rectangular heat-generating modules deployed along one wall of a flat rectangular duct was investigated extensively by Sparrow and coworkers. Transfer coefficients at the modules are reported in Refs. 90 and 91 for fully populated arrays, arrays in which there are missing modules, and arrays where barriers are implanted to obtain heat transfer enhancement. Quasi-local transfer coefficients at the smooth principal wall of the duct are reported in Ref. 92. The heat transfer response to the occasional presence of modules whose height differs from that of the others in the array is investigated in Ref. 93, where results are reported for the part of the array in which fully developed flow would prevail if the odd-size modules were not present. Souza Mendes and Santos [94] investigated the presence of a taller module in the array and its consequences on heat transfer coefficients of other modules for various relative positions of the tall module, including both the entrance and the fully developed portions of the flow.

Other Configurations

Transfer coefficients were measured at a plane surface pierced by one aperture [95], by an array of holes [96], and by an array of slots [97] into which fluid flows from a large upstream space.

An array of cylindrical pin fins, with air entering the array at the plane of the fin tips, turning as it passes through the array, and exiting the array in cross flow, was studied by Sparrow et al [98]. Per-fin transfer coefficients were determined experimentally. It was shown that partial shrouding of the inlet can give rise to nearly uniform per-fin coefficients throughout the array. A similar study is reported [99] for an array of rectangular fins.

Wind tunnel experiments were performed to study the effect of the angle of attack on the transfer coefficient for annular fins [100]. The zones on the fin surface forward, at the side, and behind the tube were investigated. Small angles of attack were studied, namely, from $-2°$ to $+2°$. Heat transfer enhancement due to the angle of attack was about 50% in the forward zone, owing to flow separation at the leading edge. In the side and rear zones, the enhancement was about 10%. Longitudinal fins were also studied for cross-flow banks of finned tubes [101], and it was found that a high degree of heat transfer enhancement can be obtained by finning.

An interesting application of the naphthalene sublimation technique is the determination of the convective component of the heat transfer coefficient for the flow of a gas in a fluidized bed [102]. In heat transfer experiments it is not possible to separate the convective from the conductive component of the heat transfer coefficient. The sublimation technique permits direct measurement of the convective component because there is no counterpart for conduction in the analogous mass transfer experiments.

USEFULNESS AND LIMITATIONS

The main feature of the naphthalene sublimation technique is that it combines simplicity with accuracy. The control of boundary conditions is normally straightforward, and no corrections for radiation are needed. Accurate determination of local transfer coefficients becomes fairly easy with the technique. Moreover, the typical apparatus involved is of simple construction and operation, resulting in low-cost experiments.

The technique also has a number of shortcomings. For example, no temperature or mass-fraction fields in the flowing fluid can be determined; although this information is of no interest for most engineering applications, it is desirable when theoretical models are to be validated. Another limitation stems from the fact that only boundary conditions of zero heat flux or uniform temperature can be easily simulated. The technique is not suitable either when the effects of variation of thermophysical properties are to be investigated. Finally, it should be mentioned that if the dependence of Nu on Pr is not known a priori, the results obtained are limited to Pr = 2.5.

I wish to express my gratitude to Prof. F. E. M. Saboya and to Dr. R. K. Shah for several interesting suggestions and encouragement. Special thanks are due to Prof. L. F. A. Azevedo, who carefully reviewed the text and kindly offered comments and suggestions for improvement. I am indebted to Ms. M. F. Naccache for her enormous help in the literature review. Financial support from CNPq and SCT/PR is also acknowledged.

NOMENCLATURE

A	mass transfer area, m^2
B_1	a constant, dimensionless
B_2	a constant, K
C	A constant, dimensionless
$\mathscr{D}$	air-naphthalene mass diffusivity, m^2/s
$\mathscr{D}_{AB}$	mass diffusivity for the binary mixture between components A and B, m^2/s
g	acceleration due to gravity, m^2/s
h	heat transfer coefficient, $W/(m^2 \cdot K)$
$\mathbf{j}$	mass flux, $kg/(m \cdot s)$
K	mass transfer coefficient, m/s
L	a characteristic length, m
m	a constant, dimensionless
$\dot{m}$	mass transfer rate, kg/s
$\dot{m}''$	mass flux, $kg/(m^2 \cdot s)$
n	a constant, dimensionless
Nu	Nusselt number ($\equiv hL/k$), dimensionless
p_n	partial pressure of naphthalene vapor, Pa
P_{nw}	partial pressure of naphthalene vapor at the subliming wall, Pa
Pr	Prandtl number ($\equiv \nu/\alpha$), dimensionless
$\mathbf{q}$	heat flux, W/m^2
R	ideal gas constant of naphthalene vapor, J(kg · K)
Ra	Rayleigh number ($\equiv g\beta\Delta T_{ref} L^3/\nu\alpha$), dimensionless
Ra_*	modified (or mass transfer) Rayleigh number ($= g\beta'\Delta\omega_{ref}L^3/\nu\mathscr{D}$), dimensionless
Re	Reynolds number ($= VL/\nu$), dimensionless
Sc	Schmidt number ($= \nu/\mathscr{D}$), dimensionless
Sh	Sherwood number ($= KL/\mathscr{D}$), dimensionless
$\mathbf{s}$	position vector on naphthalene surface, m
t	time, s
T	temperature, K
T_f	fluid temperature, K
T_{ref}	reference temperature, K
T_w	temperature at the naphthalene surface, K
$\mathbf{v}$	velocity vector field, m/s
V	a characteristic fluid velocity, m/s
$\dot{W}$	air mass flow rate, kg/s
x	streamwise coordinate, m

Greek Symbols

α	thermal diffusivity, m^2/s
β	thermal expansion coefficient, K^{-1}
β'	mass expansion coefficient, dimensionless
δ	sublimation depth, m
Δm	mass loss during the data run, kg
ΔT	temperature difference, K
ΔT_{ref}	characteristic temperature difference, K
$\Delta\Omega_{ref}$	characteristic mass-fraction difference, dimensionless
κ	thermal conductivity, $W/(m \cdot K)$
ν	air kinematic viscosity, m^2/s
θ	temperature, dimensionless
λ	latent heat of sublimation, J/kg
ω_A	mass fraction of component A, dimensionless
ω_{Af}	mass fraction of component A in the fluid, dimensionless
$\omega_{A,ref}$	reference value of ω_A, dimensionless
ω_{Aw}	mass fraction of component A at the wall, dimensionless
ω_n	mass fraction of naphthalene vapor, dimensionless

ω_{nw} mass fraction of naphthalene vapor at the subliming wall, dimensionless
ω_{nf} mass fraction of naphthalene vapor in the fluid, dimensionless
Ω_A normalized mass fraction, dimensionless
ρ density of the binary mixture, kg/m^3
ρ_n mass concentration of naphthalene vapor, kg/m^3
ρ_{nf} mass concentration of naphthalene vapor in the fluid, kg/m^3
ρ_{nw} mass concentration of naphthalene vapor at the wall, kg/m^3
ρ_{ref} reference value of ρ, kg/m^3
ρ_s density of solid naphthalene, kg/m^3
τ time duration of data run, s

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Received June 19, 1989; revised March 28, 1991