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FURTHER VERIFICATION OF A THEORY FOR MASS AND HEAT TRANSFER FROM EVAPORATING POOLS

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Summary

In an earlier paper (by the present author), a theory was described for predicting evaporation rates from plane liquid surfaces, or heat transfer rates from a region of uniform temperature rise in a smooth or rough plane surface, into a neutral turbulent boundary layer. In this paper the theory is compared with seven further sets of data. Without any modifications, the theory works for laboratory conditions to a good degree of accuracy. Application to the conditions of accidental chemical spills in the open air introduces more uncertainty for various reasons, but gives results that should be acceptable for hazard assessment purposes, provided the terrain is flat and unobstructed.

1. Introduction

The computer model GASP [1] has been developed to predict the vaporisation rates from spreading pools of volatile hazardous liquids. This is an important element in major hazard risk analysis, which yields the source term needed for subsequent calculations of the dispersion of flammable and toxic vapours. A previous paper [2] described an analytical theory for predicting mass transfer rates from such a pool caused by turbulent diffusion into the wind.

The theoretical model consisted of an approximate solution of the diffusion equation with a linearly varying eddy diffusivity. Boundary conditions corresponding to an evaporating pool or a region of uniform temperature rise were incorporated by using relations for the laminar sublayer obtained from the literature for equilibrium wall flows. The mathematical difficulties of the logarithmic velocity profile were removed by using power-law profiles matched at the typical depth of the vapour layer: the accuracy of this approximation depends on the smallness of the power-law index n. The model does not involve

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any fitting of parameters to the experimental data specific to evaporating pools or heat transfer downstream of a stepwise temperature rise.

Brighton [2] included a review of the experimental results on pool evaporation rates, but it was concluded that much of that data was inadequate for testing the theory, because of lack of measurement of relevant parameters, and poorly controlled operating conditions, resulting in large inconsistencies between different sets of the data. Several experiments did, however, give good agreement with the theory, as did results from a few numerical solutions of the problem.

We have now found a considerable amount of further relevant data, most of which is of good quality. This paper reports the comparison of that data with the theoretical predictions. As explained in the earlier paper by Brighton [2], the mathematical solution developed (in dimensionless form) is equally applicable to an analogous heat transfer problem, with the Prandtl number, Pr, playing the role of the Schmidt number, Sc. Indeed it appears rather easier to carry out well-controlled experiments for heat transfer rather than mass transfer, so experiments of both types have been examined.

Most of the notation is described as it is introduced. Other symbols have the same meaning as in Brighton [2]: in particular j is the local non-dimensionalised heat or mass transfer rate. For heat the scale factor is $\rho c_p \theta u_*$, where ρ is the density, c_p the specific heat at constant pressure, θ the imposed temperature difference and u_* the friction velocity. For mass transfer, the scale factor is $c_s u_*$, where c_s is the saturated vapour concentration at the liquid surface. Two further transfer rates are used: $\bar{j}$, the mean value of j over the region of transfer and j', the mass transfer rate including the effects of the vertical vapour velocity at high vapour pressures (see Section 5 of Ref. 2).

2. Heat transfer experiments

2.1 Ligrani and Moffat's wind tunnel experiments

Our theory applies to evaporating pools or regions of uniform temperature rise which are embedded in an extensive flat plane over which the turbulent flow field is uniform in horizontal directions. To achieve these conditions in wind tunnel experiments the transfer region must be located well downstream of the leading edge of the flat plate over which the boundary layer develops. In 1985 Ligrani and Moffat [3] presented measurements of heat transfer from a region of uniform temperature rise in artificially thickened boundary layers, which change very slowly over the transfer region. These appear to be measurements of high quality ideally suited for checking the numerical results found in this study.

They first used a smooth wall, obtaining a boundary layer whose characteristics are described in Ligrani and Moffat [4]. Heat transfer rates are described in terms of a Reynolds number, Re_x, based on the free-stream speed, U_∞, and

the distance from the (imaginary) leading edge of the plate, x. The friction velocity is stated to agree with the correlations of Schultz–Grunow, namely

$$u_*/U_\infty = 0.430\ (\log Re_x)^{-1.292} \tag{1}$$

(cf. eqn. (21.19a) in [5]). Local heat transfer rates were measured from individual plates of length $L_p = 0.1017$ m, with the uniform temperature rise starting at a point distance $x = \xi$ from the effective leading edge. Three values of Re_ξ were used, 1.73×10^6, 2.15×10^6 and 2.56×10^6, and measurements extended up to $Re_x = 3.4 \times 10^6$ in each case (see Fig. 2 of Ligrani and Moffat [3]). Their results were well correlated by the relation

$$j = 0.0339\ Re_x^{-0.2} [1 - (Re_\xi/Re_x)^{0.9}]^{-0.111} U_\infty/u_* \tag{2}$$

in our notation. Here we have inserted the Prandtl number value 0.71 in their Stanton number formula. We have evaluated eqn. (2) for the data with $Re_\xi = 2.15 \times 10^6$, using values of Re_x and U_∞/u_* for individual measurement points. u_*/U_∞ varies by only 4%, from 0.0397 to 0.0381, over the whole heat transfer region. The dimensionless heated distances are given by

$$d/z_0 = (Re_x - Re_\xi) u_*/0.13 U_\infty \tag{3}$$

The smallest value of d used was 1.5 L_p, i.e. the midpoint of the *second* heated plate. The measurements from the first plate were treated as mean heat transfer rates (see below).

The results from eqn. (2) for $Re_{\xi=}$-2.15×10^6 are shown in Fig. 1 and compared with our theoretical curves. The results for the other values of Re_ξ are indistinguishable, since u_* varies so slowly. The experimental correlation follows the trend of the theory quite closely, but is about 10% lower. In fact, close inspection of Ligrani and Moffat's Fig. 2 [3] suggests that on average their data lies a few percent above the correlation, making the agreement with our theory rather better. Also shown on Fig. 1 is the single experimental heat transfer result of Snijders et al. [6], which was briefly described in our earlier paper [2]. It falls almost exactly on Ligrani and Moffat's curve.

The first heated plate in Ligrani and Moffat's [3] experiments gives a mean heat transfer rate for $d/z_0 = 2 \times 10^4$. The first data point from each of their three unheated starting lengths ξ gives values for $\bar{j}$ of 0.0923, 0.0960 and 0.1422. The first two of the values agree rather well with the predicted $\bar{j} = 0.0933$; Ligrani and Moffat discount the other, for the lowest value of ξ, because of high mixing in the immediate vicinity of the artificial thickening device.

We now turn to results for a rough-wall boundary layer. Ligrani and Moffat's [3] heat transfer results all come from a boundary layer with $U_\infty = 26.8$ m/s, whose characteristics are defined by Ligrani et al. [7]. The roughness elements were spheres of diameter 1.27 mm with an equivalent sandgrain roughness size of $h_s = 0.79$ mm. According to Monin and Yaglom ([8], p. 289) or Schlichting

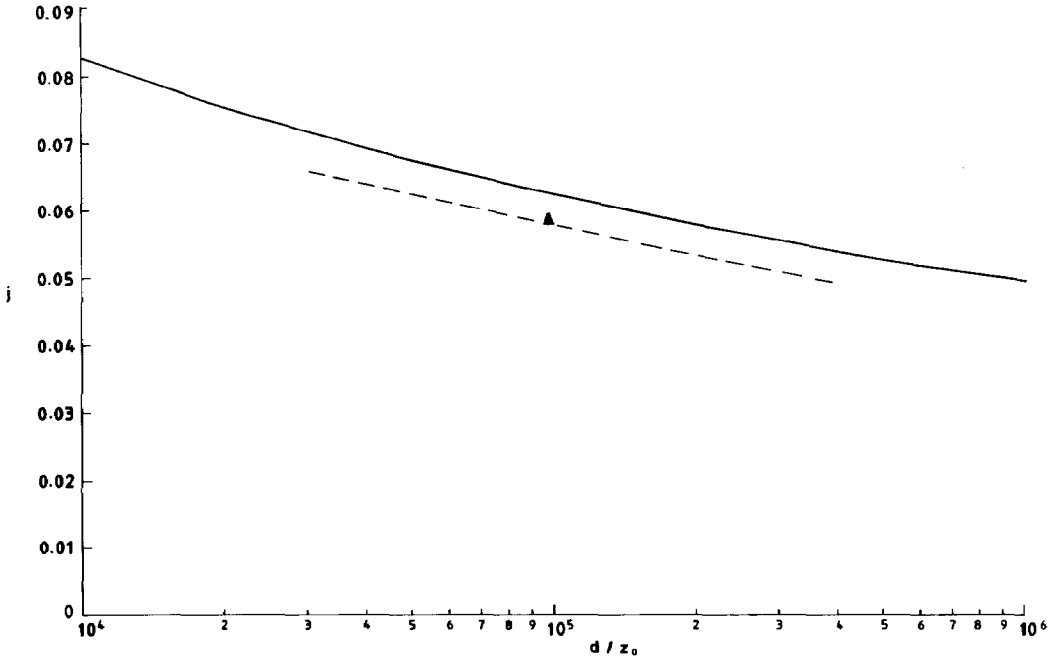

Fig. 1. Local heat/mass transfer coefficient as function of dimensionless pool length for a smooth wall at $Pr=0.71$. —: prediction [2] - -: correlation of data of Ligrani and Moffat [3]: ▲: experimental result of Snijders et al. [6].

([5], Ch. XX), this corresponds to a roughness length $z_0=h_s/30=2.63\times10^{-5}$ m. The skin-friction distribution was given by

$$\frac{u_*^2}{U_\infty^2}=0.00751\left[\frac{x}{r}\right]^{-0.149} \tag{4}$$

from Ligrani et al.'s Fig. 10 and eqn. (11), where r is the roughness element radius. Ligrani and Moffat [3] present heat transfer measurements with unheated starting lengths ξ ranging from 0.61 m to 4.52 m. For this range of ξ, u_* ranges from 1.392 m/s to 1.199 m/s, and the roughness Reynolds number is $Re_0=u_*z_0/\nu=2.3$ to 2.0 (assuming $\nu=1.6\times10^{-5}$ m^2 s^{-1}, corresponding to 30°C). This value of Re_0 is just sufficient to ensure completely rough dynamical conditions ([8] p. 289). The parameter determining the near-wall heat transfer is $Re_0^{1/2}\,Pr$ in our theory — values of this parameter range from 1.07 to 1.00.

Again Ligrani and Moffat give a correlation which matches their data closely:

$$j=0.00729\left[\frac{x}{r}\right]^{-0.149}\left[1-\left(\frac{\xi}{x}\right)^{0.90}\right]^{-0.145}\frac{U_\infty}{u_*} \tag{5}$$

To compare with our prediction we have evaluated this for $\xi=2.93$ m, taking x

from $\xi + 1.5\, L_p$, for reasons explained above, to the end of the heated section at $x = 4.7$ m. The results are shown in Fig. 2 where it is seen that there is very good agreement with our predictions for $Re_0^{1/2} Pr = 1$ at the lower values of d/z_0, though the experimental correlations decrease slightly faster with a discrepancy of about 7% developing at the end of the heated section.

As for the smooth wall, we have treated results from the first heat transfer plate as mean values over the plate. This gives five values $\bar{j}$ for $d/z_0 = 3.9 \times 10^3$; these range from 0.084 to 0.089. The lower end of this range coincides with our prediction.

We conclude this section by noting that Ligrani and Moffat [3] present in their Fig. 14 a plot of distributions of the turbulent Prandtl number σ with distance from the surface, determined from their own and previous experiments. This provides additional evidence that the value $\sigma = 0.85$ used in the model of Brighton [2] is a good estimate, particularly in the inner part of the boundary layer, which is the more important in this problem.

Ligrani and Moffat [3] refer to several earlier studies of the problem dating from the period 1951–1961. They report good agreement between their results and those of Reynolds and co-workers for smooth walls. We have taken three

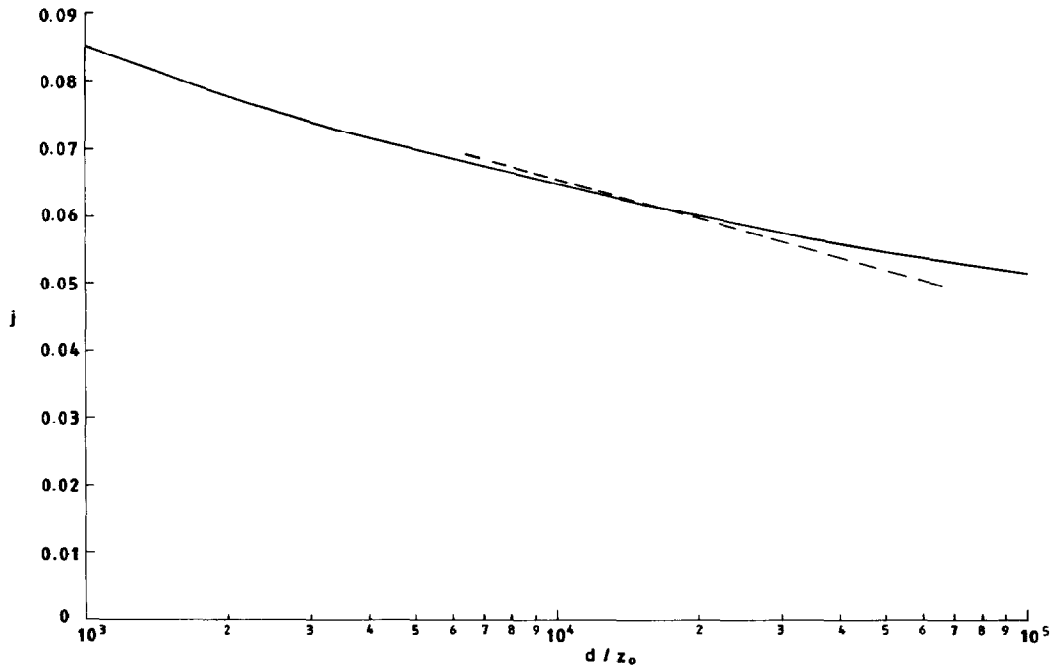

Fig. 2. Local heat/mass transfer coefficient as function of dimensionless pool length for a rough wall. — : prediction Brighton, [2], $Re_0^{1/2}\, Pr = 1$. - - - : correlation of data of Ligrani and Moffat [3], $Re_0^{1/2}\, Pr$ between 1.00 and 1.07.

of the other papers referred to in order to make further comparisons with our theory. These have been chosen mainly for ease of procurement. The experiments of Johnson [9] and Ede and Saunders [10] concern heat transfer, while that of Sogin and Goldstein [11] involved sublimation of naphthalene and so is included in Section 3 below.

2.2 Johnson's wind tunnel experiments

In this 1957 paper, details of velocity and temperature profiles are reported for a single free-stream speed of 7.62 m/s for a uniformly heated (i.e. with a uniform temperature rise) flat plate of length 1.829 m at the end of a wind tunnel of length 6.096 m and flush with the floor. Johnson [9] found considerable scatter (by a factor of two) in values of the skin-friction coefficient evaluated by various means at one location, and presents his best estimates in his Fig. 9 based on a semi-empirical relation. The heat transfer rates were determined from the convection thickness δ_c of the thermal boundary layer, but values of this computed from the measured profiles of velocity and temperature vary rather erratically with distance from the leading edge, and so Johnson's local heat fluxes as presented in his Figs. 11 and 12 appear subject to considerable error, being obtained from $d\delta_c/dx$. Therefore we compare the measurements with values of mean transfer coefficient $\bar{j}$ of our theory, which should be related to δ_c by the relation

$$\bar{j}=\frac{U_\infty}{u_*}\frac{\delta_c}{d} \tag{6}$$

where d is the distance from the leading edge of the heated plate. Values of d/z_0 were obtained by using the friction velocity values averaged over the distance d — over the whole heated plate u_* ranged from 0.298 m/s to 0.280 m/s. An air temperature of 30 °C close to the plate was assumed (the temperature excess was 15 K). The data are plotted in Fig. 3, where it will be seen that there is considerable scatter with an average result about 20% greater than predicted. In view of the uncertainty surrounding determination of u_*, this seems reasonably satisfactory agreement.

2.3 Ede and Saunders' water-channel experiments

The 1958 results of Ede and Saunders are particularly useful because they used water as the working fluid, with Prandtl numbers in the range of 6–7. One of the problems with data on evaporation examined by Brighton [2] was that water mass transfer rates at $Sc=0.6$ did not always show the expected increase over values for organic liquids with Sc around 2. However, not much weight could be placed on this finding because of the various shortcomings of the experiments concerned. The comparison between heat transfer rates for air and water provides a test for the theory over the same range of molecular dif-

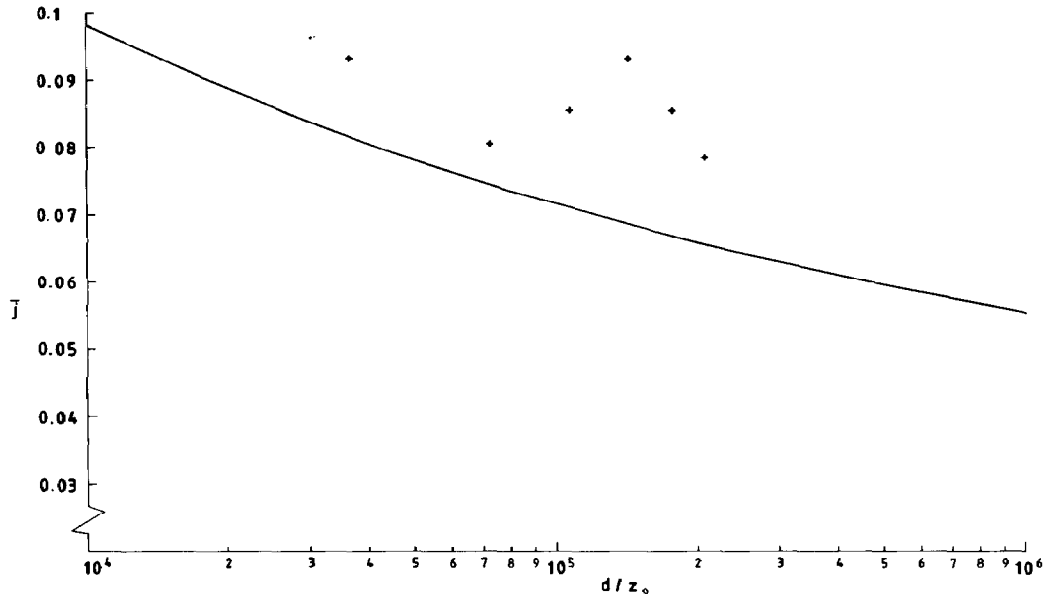

Fig. 3. Mean heat/mass transfer coefficient as function of dimensionless pool length for a smooth wall at $Pr=0.71$. — : prediction, +: data of Johnson [9].

fusivities (relative to viscosity), without the uncertainties associated with surface ripples, contamination etc., inherent in evaporation experiments.

Ede and Saunders used a heated smooth plate of streamwise length 0.106 m and width 0.0762 m set flush at various positions in a larger unheated surface. Water speeds ranged from 0.13 to 1.20 cm/s and for most speeds and positions it was found possible to obtain a range of heat-transfer coefficients: this was attributed to transition between laminar and turbulent flow. Results were categorised according to whether the flow could be classified as laminar by visualisation or whether use of an artificial tripping device caused any change in the measured heat transfer rate. The plate was maintained at a temperature varying by no more than 10% of the mean difference between the plate and water temperatures. The skin friction distribution in turbulent conditions was determined by fitting the logarithmic velocity profile to measured values.

For comparison with our theory [2] we used the data for fully turbulent flow with the heated plate extending between 0.61 and 0.71 m from the leading edge of the complete assembly. Twenty-six data points are reported of which six are with Prandtl numbers between 6.07 and 6.26; and the remainder with Pr between 6.57 and 6.85, as a result of the heated plate being operated at differing temperatures.

The results for mean dimensionless heat transfer rate are plotted against dimensionless plate length in Fig. 4. The scatter in the data is not correlated

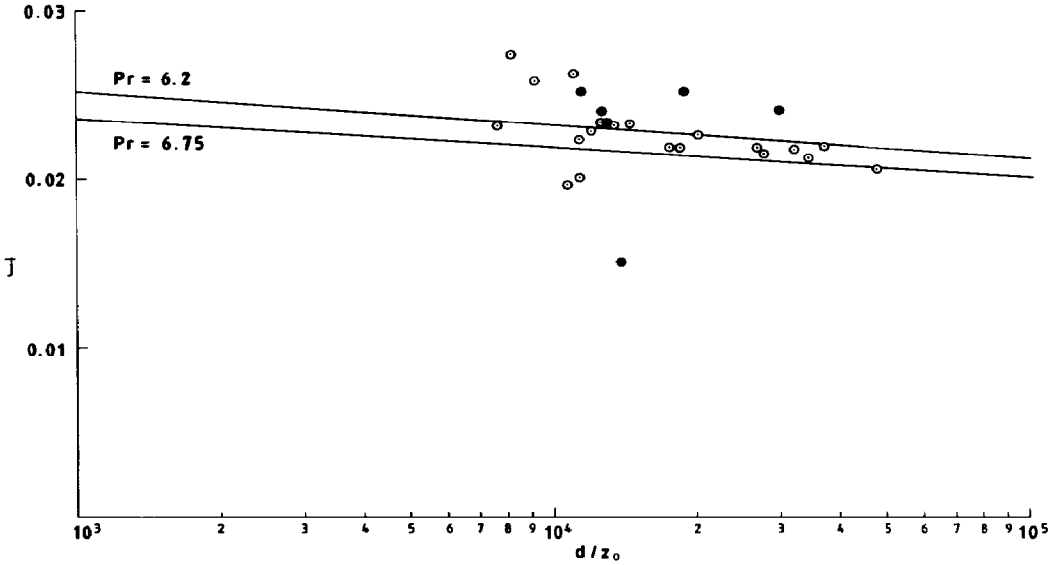

Fig. 4. Mean heat/mass transfer coefficient as function of dimensionless pool length for a smooth wall at Pr between 6.07 and 6.85. — : predictions (Brighton 1985), $Pr=6.2$ and 6.75. •: data of Ede and Saunders [10] for $6.57 \leq Pr \leq 6.85$, •: ditto for $6.07 \leq Pr \leq 6.26$.

significantly with Pr and is considerably larger than the theory would predict from the variations in Pr. The mean trend of both sets of results is about 5% higher than the predictions for the approximate mean Prandtl number values of 6.2 and 6.75, and so the agreement is good.

3. Mass transfer experiments

3.1 Sogin and Goldstein's naphthalene sublimation experiments

Sogin and Goldstein [11] used trays of solid naphthalene set flush into a smooth plate of length 0.381 m. The trays were about 0.12 m wide and either 0.0127 or 0.0381 m long in the streamwise direction, and mounted at various distances from the leading edge, where the boundary layer was tripped. The mass transfer rate was determined by extremely careful weighing of the specimens. The duration of tests had to be limited because of the roughness which developed as sublimation proceeded. For comparison with the theory, the friction velocity had to be estimated from the Schultz–Grunow correlation, eqn. (1), which the authors suggest is applicable in this case. The mid-point of the naphthalene strip was the position for evaluating u_* in each case. The mass transfer rate was evaluated by Sogin and Goldstein so as to apply to the whole length of the flat plate, and so to convert to $\bar{J}$ they had to be divided by the fraction of the length occupied by the naphthalene. The results are plotted in

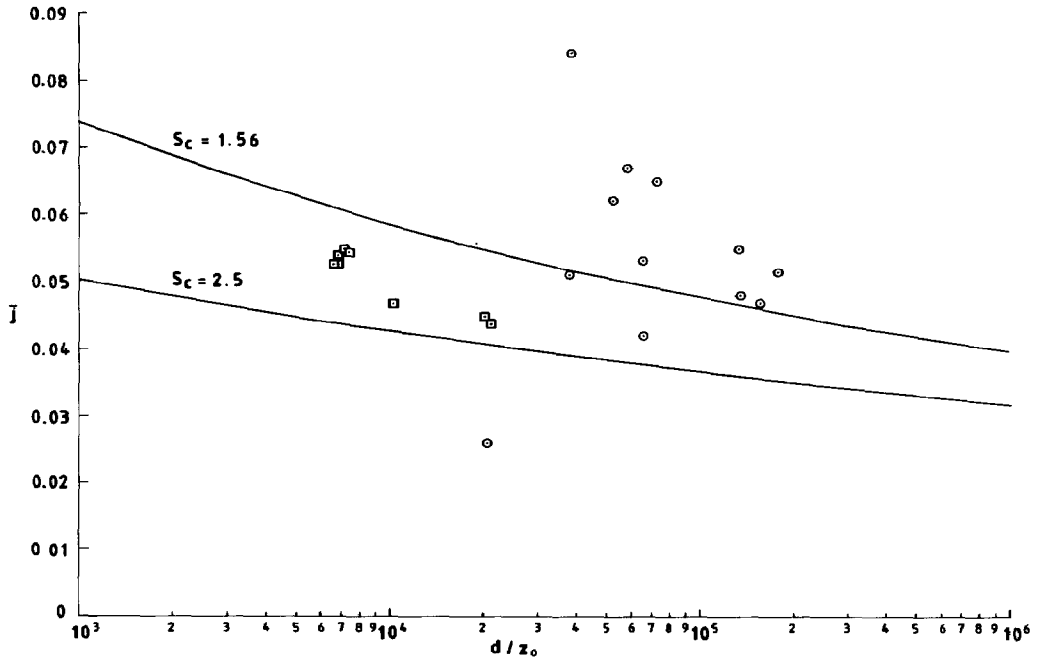

Fig. 5. Mean heat/mass transfer coefficient as a function of dimensionless pool length for a smooth wall. — : predictions (Brighton [2]), $Sc = 1.56$, 2.5. ○: data of Hankinson and Murphy [13] for $Sc = 1.56$. □: data of Sogin and Goldstein [11] for $Sc = 2.5$.

Fig. 5 and compared with the theory for $Sc = 2.5$, the value given by Sogin and Goldstein. Of the nine conditions for which results are given, three give values of $\bar{j}$ slightly less than 10% greater than the prediction while the rest form a consistent group about 20% above the theoretical curve. The length of the naphthalene strip as a fraction of the distance from the effective origin of the boundary layer to the trailing edge of the strip (i.e. a measure of the uniformity of the boundary layer over the strip) ranged from 0.039 to 0.195.

3.2 Dodge et al.'s windtunnel experiments

The US Coast Guard has sponsored a large programme of model development and experiments on the spreading and evaporation of chemicals on water (Dodge et al. [12]). The work concentrated on substances boiling above ambient temperatures and included an extensive series of pan evaporation tests in a wind tunnel at the Southwest Research Institute, Texas. These experiments are important because they were used to validate the Coast Guard's computer system for assessing hazards from volatile chemical spills. They are included here for that reason, even though we find serious inaccuracies and inconsistencies in the results.

The wind tunnel had a rectangular test section of height 0.305 m, width 0.610 m and length 4.84 m. Air was drawn in through a bell mouth by an exit fan — no measures were taken to develop or control an equilibrium turbulent boundary layer structure. The evaporation pan was located about 3 m from the entrance and was 1.219 m long. Eight baffles were installed below the liquid surface to prevent wave formation and also a 35-mm square horse-hair filter was installed at the downstream end to damp surface waves. The liquid level was maintained constant at about 5 mm below the tunnel floor.

The velocity profile was measured by a hot-wire anemometer near the end of the pool at 1.066 m from the leading edge. Mean air speeds were varied from 2 to 5 m/s and friction velocities were calculated.

The mass flux from the surface was determined by two independent means. The first method was based on measurement of concentration profiles at 1.111 m downwind from the upwind edge of the pool. These were analysed to give local mass transfer coefficients (assuming a turbulent Schmidt number value of 0.85). Secondly, the total mass transfer rate was estimated by measuring the concentration at the fan outlet (and assuming complete mixing).

In view of the efforts made in these experiments to ensure a smooth liquid surface and to measure velocity profiles close to the surface, details which have been sadly lacking in most previous evaporation experiments [2], it is very disappointing to find that the mass transfer results are highly erratic and that any consistent trends that do emerge are at total variance with previous experimental and theoretical conclusions. Dodge et al. tabulate in their Table IV.16 local and mean Dalton numbers which are in fact defined identically to our mass transfer coefficients j and $\bar{j}$. The substances used were ethyl acetate ($Sc=1.82$), hexane ($Sc=2.16$), hexanol ($Sc=2.19$), octanol ($Sc=2.36$) and octane ($Sc=2.61$).

When j is plotted against values of d/z_0 deduced from the velocity data, we find that the results vary by a factor of almost 5, much more than the expected variability, with the middle of the range near to the theoretical curve for $Sc=2$. Moreover the experimental results have no consistent variation either with Sc — ethyl acetate having mass transfer rates considerably lower than octane, for instance — or with d/z_0, where for individual substances there is often a rather steep increase with d/z_0.

The mean mass transfer rates are found to be somewhat more self-consistent, and are again centred on the curve for $Sc=2$, but they still have an anomalous increase with d/z_0, except for hexane.

In fact, it seems that both sets of measurements represent random scatter — the correlation coefficient between the profile measurements and the exhaust flow measurements is 0.097, i.e. practically no correlation whatsoever.

3.3 British Gas butane trials

Hankinson and Murphy [13] carried out eight tests in which an insulated square bund of side 1.22 m was filled with liquid n-butane which was allowed

to evaporate under the influence of wind and sun. It was found that reasonably constant pool temperatures were rapidly attained and thus evaporation rates could be derived from data collected over a period of the order of $1–4 \times 10^3$ seconds. Here we are concerned only with the comparison of the mass transfer rates with those predicted by the evaporation model of Brighton [2]. The pool temperature is regarded as a known parameter – the GASP program does calculate the temperature balance of such a pool and so can predict the temperature achieved: this is a separate exercise which has been undertaken elsewhere (cf. Webber and Jones [1]).

The difficulties of interpreting evaporation field trials data have already been discussed by Brighton [2]. Hankinson and Murphy's trials are an improvement on previous investigations in two ways. First, the bund was recessed into the ground, so that its edges do not project as bluff bodies to produce a gross distortion of the wind field: the pool was set into the top of a mound with slopes of 1:15, which should not produce too severe a departure from flat-ground boundary-layer conditions. However, as the recess depth was 75 mm and tests were conducted without replenishment of the liquid, the wind did cross a downstream-facing step to reach the liquid surface. The second felicitous feature of these trials is that the windspeed measurements were made at a height of just 0.3 m, at the side of the pool, in the cross-wind direction. Thus while not actually that over the pool, this windspeed is considerably closer to that at the liquid surface than in previous field trials discussed by Brighton [2].

The maximum height of the mound was less than 0.3 m and it was located at the NE corner of a 100 m $\times$ 100 m flat concrete pad. For wind directions $< 180°$ and $> 266°$ the land upwind of the pool was undulating rough hillside; for directions within the range $186°–266°$ the surface upwind was very flat for about 80 m. The trials were almost all conducted with the wind within or near the edges of this quadrant. The results of the trials are summarised in Table 1. Variables are listed as averages, for one or two periods of time within each Test during which conditions were reasonably steady. The evaporation rate was determined by the change in level of the butane over each time period using a least-squares straight line fit of the data. Two means of measuring depth were used in most tests – the "dip stick" and the "dip tube".

3.3.1 Comparison with rough surface model

Brighton [2] recommended that for predicting evaporation from outdoor pools one should assume a rough, wavy liquid surface and take $z_0 = 2.28 \times 10^{-4}$ m. This value is probably an upper bound as it is representative of lake and sea surfaces in moderate winds, where there is an extended fetch for wave development. Using this value of z_0 to evaluate U/u_* puts Hankinson and Murphy's data into the appropriate dimensionless values given in Table 2.

In this table, the vapour pressure has been evaluated at the central temperature listed in Table 1. It gives temperature ranges about this value, which

TABLE 1

Summary of data of Hankinson and Murphy [13] on evaporation of n-butane from a 1.22 m × 1.22 m pool

Test No.	Period (s)	Ambient pressure (mbar)	Ambient temperature (K)	Relative humidity (%)	Wind speed (m/s)		Wind direction (magnetic degrees)	Pool temperature (K)	Evaporation rate ($g\ m^{-2}s^{-1}$)	
					9.15 m	0.3 m			Dip tube	Dip stick
1	2800–5500	970.0	283	49	3.5	2.0	200	236.6 ± 2.3	–	3.7
2	0–2400	977.0	282	54	3.0	–	155 ± 45	243.5 ± 2.0	–	3.9
	2900–6800	977.0	282	54	3.0	–	155 ± 45	241.1 ± 0.6	–	2.7
3	1000–3200	976.0	280	82	1.5	1.0	305 ± 20	244.2 ± 1.1	–	2.3
	3700–6300	976.0	280	82	1.5	–	288 ± 10	241.0 ± 0.9	–	3.2
4	0–1000	986.5	291	84	3.6 ± 0.5	1.4 ± 0.6	117 ± 5	247.8 ± 6.0	–	4.6
	1000–3000	986.5	291	84	2.4 ± 0.4	1.6 ± 0.6	61 ± 13	241.1 ± 3.0	4.1	3.9
5	0–1200	982.0	289	100	5.8 ± 1.1	4.7 ± 0.9	257 ± 9	237.6 ± 0.4	4.1[a]	6.1
	1200–2200	982.0	289	100	4.8 ± 1.1	3.9 ± 0.9	255 ± 5	236.3 ± 0.4	5.9	5.5
6	0–2000	987.0	275	100	5.5 ± 1.0	4.0 ± 0.8	279 ± 12	238.8 ± 2.9	6.0	5.6
7	0–1400	988.5	287	89	6.9 ± 1.4	5.5 ± 1.1	254 ± 10	236 ± 4	8.1	7.2
8	0–3700	989.0	282	89	1.0 ± 0.6	0.5 ± 0.4	281 ± 18	263	2.0	2.0

[a]During this period of Test 5, the dip tube became blocked by ice and so this result is considered unreliable.

TABLE 2

Analysis of Hankinson and Murphy's [13] n-butane tests assuming rough-surface mass transfer with $z_0 = 2.28 \times 10^{-4}$ m, $d/z_0 = 5351$ and $Sc = 1.56$

Test No.		P_v/P_0	$\bar{J}'$	$\bar{J}$	$Re_0^{1/2}Sc$	Ri
1		0.205	0.054	0.048	2.4	0.034
2	Period I	0.290	0.047	0.040	2.5	0.030
	Period II	0.257	0.037	0.032	2.5	0.029
3	Period I	0.300	0.047	0.040	1.7	0.15
	Period II	0.256	0.076	0.066	1.7	0.14
4	Period I	0.353	0.058	0.047	2.0	0.092
	Period II	0.255	0.058	0.050	2.1	0.061
5	Period I	0.214	0.036	0.032	3.7	0.007
	Period II	0.200	0.042	0.038	3.3	0.009
6		0.226	0.037	0.033	3.4	0.009
7		0.198	0.040	0.036	4.0	0.005
8		0.690	0.038	0.022	1.2	0.99

reflect variations between different thermocouple positions and variations in the course of each test. The thermocouple producing most of these values was at a fixed height of 25 mm above the pool bottom, and thus its depth below the surface varied during the course of a test — thus the difference between this temperature and that at the surface relevant to evaporation may have an ar-

tificial variation due to this effect. In Table 2 the atmospheric pressure values used are those measured in the tests. The dimensionless mass transfer rate is

$$\bar{J}' = J/c_s u_* \tag{7}$$

with c_s determined from the ideal-gas law and vapour pressure, and u_* determined from the mean windspeed taking $z = 0.3$ m (thus ignoring the drop in level of the liquid). The "dip stick" evaporation rates were used, since they were available for all tests and were consistent with the "dip tube". For Test 2, only the windspeed at 9.15 m was obtained, as 3.0 m/s. Judging from other tests at similar windspeeds, a plausible value at 0.3 m is 1.8 m/s, which has been used in compiling Table 2. Also for the second period in Test 3, no windspeed is given at the lower level. Since the windspeed at 9.15 m was the same, it was assumed that at 0.3 m continued at 1.0 m/s.

The mass transfer rates were corrected for the high vapour pressure levels by using the standard "film theory" factor as explained in Brighton [2]. The resultant mass transfer coefficient $\bar{J}$ is then the quantity predicted by the model with results given in Fig. 4 of [2] for rough liquid surfaces.

Since d/z_0 is fixed for this set of data, $\bar{J}$ should be a function of $Re_0^{1/2} Sc$ alone, with $Re_0 = u_* z_0/\nu$ and Sc the Schmidt number. Values are tabulated in Table 2 and compared graphically with the theory in Fig. 6. The measured evaporation rates are rather scattered and on average around 60% of the predictions. Test 8, carried out at night at a low windspeed, gave a particularly low evaporation rate, less than a third the prediction. This may be because of the large variability in the windspeed (0.5 ± 0.4 m/s), because of low atmospheric turbulence in the stable nocturnal conditions, or because of the suppression of turbulence

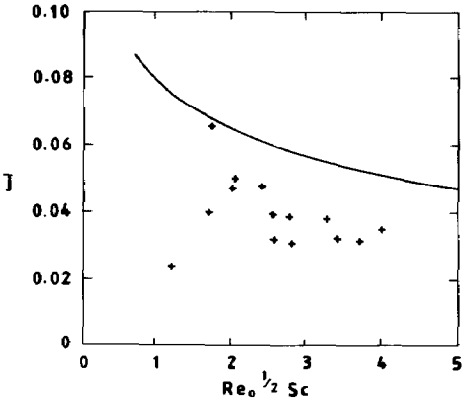

Fig. 6. Mean heat/mass transfer coefficient as a function of $Re_0^{1/2}$ Sc for conditions of Hankinson and Murphy's [13] butane experiments, assuming $z_0 = 2.28 \times 10^{-4}$ m. — : prediction (Brighton [2]), +: data using mean temperatures.

by stratification caused by the presence of vapour. In this trial the Richardson number, $Ri = g\Delta' z_1/U_1^2$, was 0.99, whereas the other trials had $Ri \ll 1$ indicating buoyancy effects unimportant. Here Δ', the relative density difference, was calculated from

$$\Delta' = \rho_s/\rho_0 - 1 = \frac{T_0}{T} - 1 + \frac{T_0}{T}\frac{P_v}{P_0}\left[\frac{M_v}{M_A} - 1\right] \tag{8}$$

where ρ_s is the vapour saturated air density, ρ_0 ambient air density, T_0 ambient absolute temperature, T vapour absolute temperature, P_v vapour pressure, P_0 atmospheric pressure, M_v vapour molecular weight and M_A air mean molecular weight. Here $\rho_0 = M_A P_0/RT_0$, i.e. effects of both molecular weight and temperature are included, whereas Brighton [2] only included the molecular weight effect in Section 6.

To check the effects of the uncertainties in the temperature measurements, the values of $\bar{J}$ were recalculated using the limits of the temperature ranges shown in Table 1: this did not improve overall agreement significantly.

3.3.2 Comparison with smooth surface theory

On the hypothesis that the pool surface could be treated as aerodynamically smooth, the data were evaluated to find u_* and z_0 from the smooth-wall formulae. The results are listed in Table 3 and plotted on Fig. 5 together with the theoretical curve for $Sc = 1.56$. The fit turns out fairly well with the scatter of values centred on the predicted curve with maximum deviations from the theory of about 50%, including Test 8, whose anomalously low value has been

TABLE 3

Smooth-wall data analysis (square bund liquid n-butane) with $U = (u_*/\kappa)\log(u_* z/0.13\nu)$, $z_0 = 0.13\nu/u_*$, $Sc = 1.56$, $z = 0.3$ m and $\nu = 1.1 \times 10^{-5}$ m^{2}s^{-1}

Test No.		u_*	d/z_0 ($\times 10^4$)	$\bar{J}$
1		0.084	7.2	0.065
2	I	0.076	6.5	0.053
	II	0.076		0.042
3	I			
	II	0.045	3.8	0.051
4	I			0.084
	II	0.061	5.2	0.062
5	I	0.0685	5.8	0.067
	II	0.183	15.6	0.047
6		0.154	13.1	0.055
7		0.158	13.4	0.048
8		0.211	17.9	0.053
		0.024	2.05	0.026

explained above. The other main anomaly is Test 3, Period II, in which the measured evaporation rate exceeded that in Period I by 39%, despite a slight fall in temperature and no change in wind speed.

The agreement is encouraging but should be treated with caution because the pool surface was observed to be rippled, and the change in roughness might be expected to produce an acceleration of air flow over the pool. These effects would, however, be expected to produce anomalously high mass transfer rates. On the other hand, the retreat of the pool surface below the top of the bund should have produced a small sheltering effect (see Section 3.4 below).

3.4 The influence of the edges of the container

One of the possible reasons for the inconsistencies in early experimental results on evaporation from liquid pools (cf. Ref. 2) is that in some cases the liquid level was not flush with the surrounding surface. Prata and Sparrow [14] have carried out experiments very relevant to this issue. They measured evaporation rates from a cylindrical container of diameter $D=38$ mm set into the floor of a rectangular duct of height $h=19.38$ mm and width $w=82.7$ mm. The duct was sufficiently long to establish developed turbulent flow and Reynolds numbers Re_D based on mean velocity upstream of the container and diameter D, ranged from 7.3×10^3 to 4.86×10^4. Evaporation data were obtained by sensitive determination of the weight of the container to yield mean mass transfer coefficients.

The depth, H, of the liquid surface below the duct floor ranged from a minimum of 0.1 D down to 3 D. Unfortunately no data could be obtained for any smaller values of H. The results at all Reynolds numbers show an initial decrease in mass transfer rate as H increases from 0.1 D, followed by a rise to a maximum at about 0.5 D. With further increase in H there is first a sharp decrease, then a smoother more gentle decrease in mass transfer rate. For $Re_D=7.3\times10^3$, 1.46×10^4, 2.92×10^4 and 4.85×10^4, the enhancements of mass transfer at the maximum, compared to that for $H/D=0.1$, were, respectively, about 20, 35, 30 and 15%. At the higher Reynolds numbers, vigorous sloshing of the liquid surface was observed. These results were all obtained with water ($Sc=0.6$). Toluene ($Sc=1.83$) was also studied for $Re_D=7300$ and a similar maximum for $H/D\simeq0.4$ was observed.

In view of the developed nature of the flow and the relative shortness of the evaporating container it is worthwhile comparing the results of Prata and Sparrow with our predictions. The skin-friction coefficient in the duct was estimated by applying the Blasius correlation to the Reynolds number based on the mean bulk velocity and the hydraulic diameter, as recommended by Schlichting ([5], Ch. XX). The length of the pool was taken as the side of the square of the same area as the circular container. Using these values to obtain d/z_0, the theoretical dimensionless mass transfer coefficients $\bar{j}$ were converted to the Sherwood numbers, Sh, in which Prata and Sparrow's results are ex-

pressed. For $Re_D = 7.3 \times 10^3$, 1.46×10^4, 2.92×10^4 and 4.86×10^4, our predictions are respectively: $Sh = 40$, 66, 108 and 157. These should be compared with Prata and Sparrow's Fig. 3 and they are seen to be a plausible extrapolation of the results to $H/D = 0$, except possibly for the lowest Reynolds number. Prata and Sparrow considered the flow to be transitional in this case, and this may explain the possible overestimation. For the two intermediate Reynolds numbers, Prata and Sparrow obtained a good collapse of the data by using the parameter $Sh\, Re_D^{0.75}$. The predicted values of this parameter in this study are 4.82×10^{-2} and 4.94×10^{-2} and these agree very well with extrapolation of the data to $H/D = 0$ in Prata and Sparrow's Fig. 5, which is at a considerably larger scale than their Fig. 3. Finally in their Fig. 6, these authors give some results for toluene ($Sc = 1.83$) at $Re = 7300$. The present study's prediction for $H/D = 0$ is $Sh = 52$, somewhat lower than extrapolation of the data would suggest.

The conclusions that the effect of a recessed water surface is to reduce the evaporation rate by around 25% as H/D increases from 0 to about 0.2, but that it then increases again to about its value for a flat surface when $H/D \simeq 0.5$. By $H/D \simeq 0.6$, a second decrease of about 25% has occurred.

In Hankinson and Murphy's experiments [13] discussed in the last section, the recess depth increased as evaporation proceeded to a maximum of 0.06 when the bund became empty. The results of Prata and Sparrow suggest that this would have reduced the evaporation rate by around 5%, an insignificant amount relative to the other sources of variation from ideal conditions evident in the data.

A further study of evaporation in the same duct flow has been made by Chuck and Sparrow [15]. They used rectangular pans spanning almost the full duct width and extending either 6.5 or 14.5 duct heights downstream. Again the depth of the liquid surface below the floor was varied. Since the pans were so long that the vapour diffusion boundary layer would have filled most of the duct, and their data is correlated in a form making very difficult a comparison with our theory, the results of Chuck and Sparrow will be discussed no further.

4. Conclusions

The accuracy of the theory of Brighton [2] is potentially limited by several factors: the validity of the gradient-transport hypothesis for turbulent diffusion; the mathematical approximation made to achieve an analytical result; the validity of the assumption of passive vapour behaviour; and failure of experimental conditions to match the idealised assumptions in various ways (e.g. developing boundary layer, change of surface roughness, etc.).

Ligrani and Moffat [3] carried out heat-transfer experiments in air ($Pr = 0.71$) conforming almost perfectly to the idealised situation considered in the theory. For smooth surfaces, our predictions lie about 10% or slightly

less above the experimental heat transfer rates. For rough surfaces, the agreement is even better, with a maximum discrepancy of about 7%.

A similar experiment is reported by Johnson [9]. The data here show considerable scatter and difficulties in determining the friction velocity are reported. On average the predictions are about 20% below the reported heat-transfer rates.

Ede and Saunders [10] provided a test of the dependence of Prandtl number by carrying out heat transfer measurements in water with Pr between 6 and 7. There is considerable scatter but the trend in the data is about 5% above the predicted values.

Four datasets on mass transfer have been studied. Sogin and Goldstein's [11] data on naphthalene sublimation give mass-transfer results around 10–20% higher than the predictions. An extensive set of wind-tunnel tests on evaporation of ethyl acetate, hexane, hexanol, octanol and octane by Dodge et al. [12] were sponsored by the US Coast Guard for the specific purpose of developing a chemical spills hazard assessment model. Unfortunately internal inconsistencies in their data, as well as failure to conform to the expected scaling relations, show these data to be subject to random errors up to 100%. All that can be concluded is that the overall mean of this data is consistent with the theory.

Brighton [2] found that available data from outdoor field trials were of little value because of poor experimental design and inadequate reporting. Experiments on butane evaporation by Hankinson and Murphy [13] are a major improvement, though scatter in the results when expressed in our non-dimensional form still has a range of $\pm 50\%$. The average of the results is, however, in good agreement with the smooth-wall theory, though we would have expected the surface to be aerodynamically rough with mass transfer rates around 40–50% higher.

Finally a recent paper by Prata and Sparrow [14] provides important information on how mass transfer rates vary if the liquid surface is lower than the surrounding terrain.

We therefore conclude that the theory works for laboratory conditions to a good degree of accuracy. Application to accident conditions in the open-air introduces more uncertainty for various reasons, but gives results that should be acceptable for hazard assessment purposes, if the terrain is unobstructed. Any improvements would have to come from much more complex analysis of the precise geometry of a pool and of the details of the atmospheric turbulence, as well as possible heavy-gas effects. Accidental releases are likely to occur in the wake of rectangular or cylindrical structures. It is difficult to estimate how big an effect this may have. The reduced mean velocities in the wake will tend to decrease mass transfer rates but the increased turbulence will have to opposite effect. We have not found any experimental data which apply to this important practical problem.

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Notation

c_p	Constant-pressure specific heat (J/kg K)
C_s	Saturated vapour concentration at pool surface (kg/m^3)
d	Length of heated zone or liquid pool (m)
D	Diameter of cylindrical container (m)
g	Acceleration due to gravity (m/s^2)
h	Duct height (m)
h_s	Equivalent sand-grain roughness height (m)
H	Depth of liquid surface below top of container (m)
j	Local dimensionless transfer coefficient
$\bar{j}$	Mean dimensionless transfer coefficient
j'	Dimensionless mass transfer coefficient including effect of high vapour pressure
$\bar{j}'$	Mean value of j'
J	Mass transfer rate (kg/m^{2}s)
L_p	Plate length (m)
M_A,M_v	Molecular weights of air and vapour (g/mol)
n	Power-law index for velocity profile
Pr	Prandtl number
P_v	Vapour pressure (Pa)
P_0	Atmospheric pressure (Pa)
r	Roughness element radius (m)
R	Universal gas contant (8.31434 J/mol K)
Re_D, Re_X, Re_ξ	Reynolds numbers based on D, x and ξ
Re_0	Roughness Reynolds number, based on u_* and z_0
Ri	Richardson number for vapour blanket
Sc	Schmidt number
Sh	Sherwood number
T	Absolute temperature of vapour (K)
T_0	Ambient absolute temperature (K)
$U(z)$	Mean velocity at height z (m/s)
U_1	Mean velocity at height z_1 (m/s)
U_∞	Free-stream velocity (m/s)
u_*	Friction velocity (m/s)
w	Duct width (m)
x	Distance from leading edge of flat plate (m)
z	Vertical distance from surface (m)
z_0	Roughness height (m)

z_1	Height scale for vapour blanket (m)

Greek

β_1	Concentration profile shift term
δ_c	Convection thickness of thermal boundary layer (m)
Δ'	Relative density difference
θ	Temperature difference (K)
κ	Von Karman constant (0.41)
ν	Kinematic viscosity (m^2/s)
ξ	Unheated starting length on flat plate (m)
ρ	Density (kg/m^3)
ρ_s	Density of air saturated with vapour (kg/m^3)
ρ_0	Density of ambient air
σ	Turbulent Prandtl or Schmidt number

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