Int. J. Heat Mass Transfer. Vol. 25, No. 6, pp. 889-897, 1982
Printed in Great Britain

0017-9310/82/060889-09 $03.00/0

HEAT AND MASS TRANSPORT IN TURBULENT LIQUID JETS

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(*Received 7 December* 1981)

Abstract—Available experimental data and theory for heat and mass transport in turbulent liquid jets are critically reviewed. It is shown that mass transfer in turbulent jets is an entrance region problem and that experimental data must be analyzed accordingly. Finite difference numerical solution of the entrance region problem shows that turbulent transport has a minor effect on mass transfer for the parameter range investigated. New experimental data for evaporation from turbulent jets are presented for a 4 mm dia. jet. Comparisons are made with data from various previous condensation studies, and an attempt made to explain the trends and apparent discrepancies.

NOMENCLATURE

c, concentration of solute gas;
$\mathscr{D}$, molecular diffusion coefficient;
d, jet diameter;
K_L, mass transfer coefficient [m s^{-1}];
L, jet length;
l, turbulent macroscale;
N_{tu}, number of transfer units;
P, pressure;
Q, volume flow rate;
r, radial coordinate;
R, jet radius;
Re, Reynolds number, Vd/ν or Vt/ν;
Sc, Schmidt number;
St, Stanton number;
T, temperature;
t, thickness of a planar jet;
V, bulk velocity;
v, turbulence intensity.

Greek symbols
ε, eddy diffusivity;
ε_M, ε_H, mass and heat transfer effectiveness, respectively;
ν, kinematic viscosity;
μ, dynamic viscosity;
ρ, density;
σ, surface tension.

Subscripts
in, out, bulk values in inlet and outlet liquid, respectively;
sat, saturation value.

Overscore
$\bar{}$, length average.

INTRODUCTION

TRANSPORT phenomena in turbulent round or planar liquid jets, issuing into a vapor or gas phase, have not received much attention in the technical literature. From a fluid mechanics point of view attention has been focused on such features as axial decay of turbulence and surface instabilities: in the latter context the use of polymer additives to dampen surface waves has been explored with a view to improving fire-fighting equipment. Momentum transport has been of little interest, since in usual situations the effects of gas phase drag on the liquid are negligible.

Mass transport in turbulent liquid jets has received some attention, particularly by Davies and coworkers in England [1-3]. However, since turbulent jets have not been a preferred contacting mode in the liquid-gas transfer operations of chemical engineering practice, there has been no systematic development of a data base spanning the relevant parameters. Recent development of the Claude open-cycle ocean thermal energy conversion concept has indicated the possible use of turbulent jet condensers [4]. Desorption of air from such jets adds to the noncondensable gas problem in the condenser: knowledge of the liquid side mass transfer coefficient is required for calculating noncondensable gas effects.

Heat transport in turbulent liquid jets is of considerable technological interest. Turbulent jet direct contact condensers have been used since the earliest days of steam engines. Very recently the need to use dry cooling towers for power plants in some locations has led to the development of large planar turbulent jet direct contact condensers [5]. Also design of the turbulent jet condensers for the Claude open-cycle ocean thermal energy conversion concept mentioned above, requires adequate knowledge of the liquid side heat transfer coefficient.

In this paper we present a critical review of available experimental data and theory for heat and mass transport in turbulent liquid jets, and present some new experimental results obtained in our laboratory. In particular we will present the following:

(i) A demonstration that the experimental results for mass transfer of Davies and Ting [1] were incorrectly interpreted, both then and subsequently by Brumfield and Theofanous [6]. By numerically solving the governing conservation equations, it will be shown that the problem is one of entrance region transport, rather than fully developed transport as was assumed. The implications for experimental study are discussed, and our unsuccessful experimental program described.

(ii) New experimental data for evaporation from turbulent jets are presented, and compared to data for condensation on turbulent jets from various sources. An attempt is made to explain the apparent discrepancies in the data.

MASS TRANSPORT

Davies and Ting [1] report average mass transfer coefficients for the absorption of CO_2 and H_2 in laminar and turbulent water jets at 25°C. Jet diameters ranged from 0.102 to 0.162 cm, jet lengths from 4 to 8 cm, and Reynolds numbers from 1500 to 20 000. The mass transfer coefficients were calculated from measured outlet bulk concentrations using the theory of a single stream mass exchanger, viz.

$$\varepsilon_M = 1 - e^{-N_{tu}} \tag{1}$$

where

$$\varepsilon_M = \frac{c_{out} - c_{in}}{c_{sat} - c_{in}} \quad \text{and} \quad N_{tu} = \frac{\bar{K}_L \pi d L}{Q}.$$

Although a two-fold variation of jet length was tested, it was concluded that, within the precision of the data, no effect of jet length on $\bar{K}_L$ could be discerned. The results were interpreted using the Levich theory for damping of turbulence at a liquid-gas interface [7], which for fully developed mass transfer gives

$$\bar{K}_L \propto \left(\frac{\mathscr{D} v_0^3 \rho}{\sigma}\right)^{1/2}. \tag{2}$$

If the characteristic turbulence velocity is taken as being constant along the jet, and equal to the friction velocity in the nozzle, $v_0 = 0.2 V Re^{-1/8}$ from the Blasius formula, then

$$K_L = C Re^{1.31} \left(\frac{\mu^3 \mathscr{D}}{\rho d^3 \sigma}\right)^{1/2}. \tag{3}$$

Since the data showed $K_L \propto Re^{1.34} \mathscr{D}^{0.55}$, it was claimed that the Levich model was valid. However, the constant C proved to be dependent on jet diameter d, and in later work, Davies and Hameed [2] found upon testing kerosene that C was dependent on liquid properties as well.

Subsequently Brumfield and Theofanous [6] pointed out that it is unreasonable to assume that the turbulence velocity v_0 will remain constant along the entire length of the jet and that its decay due to viscous effects must be taken into account. Using data obtained for decay of turbulence downstream of a grid, and applying equation (2) on a local basis, they were able to obtain better agreement between the constants C for water and kerosene. However, the allowance for turbulence decay yields a marked variation of K_L along the jet which is in direct contradiction to the experimental data.

The above described interpretations of the experimental data failed to note that the values of $\bar{K}_L$ were surprisingly low. In addition no attempt was made to ascertain whether the assumption of fully developed mass transfer was valid. To illustrate these points the data for $\bar{K}_L$ have been replotted in Fig. 1, in a form suggested by classical penetration theory for laminar flow, for which

$$\bar{K}_L = 2\left(\frac{V\mathscr{D}}{\pi L}\right)^{1/2} \tag{4}$$

or

$$\overline{St}\, Sc^{1/2} (L/d)^{1/2} = (2/\pi^{1/2}) Re^{-1/2}. \tag{5}$$

Penetration theory is not exact since it does not allow for the relaxation of the initial velocity profile. For laminar flow the true values of $\bar{K}_L$ are estimated to deviate by less than 10% [8], while for turbulent flow the discrepancy will be even smaller. Figure 1 shows little evidence for turbulent transport. In fact the $\bar{K}_L$ values at $Re \sim 10^4$ are considerably lower than the laminar theory, which suggest substantial systematic error in the experiments. We further note that these low values of $\bar{K}_L$ at low turbulent Reynolds numbers give rise to high apparent slopes of $\bar{K}_L$ vs Re in the turbulent regime. Figure 1 shows slopes considerably larger than 1/2, and, as mentioned above, Davies and Ting correlated the data as $\bar{K}_L \propto Re^{1.34}$.

Further insight into the problem we obtained by solving the entrance region problem for a turbulent jet. Plug flow is assumed in a jet of constant diameter, and an eddy diffusivity model used to characterize turbulent transport. The appropriate form of the species conservation equation is

$$V\frac{\partial c}{\partial x} = \frac{1}{r}\frac{\partial}{\partial r}\left[r(\mathscr{D} + \varepsilon)\frac{\partial c}{\partial r}\right] \tag{6}$$

which must be solved subject to the initial and boundary conditions

$$x = 0: \quad c = c_{in},$$

$$r = 0: \quad \partial c/\partial r = 0,$$

$$r = R: \quad c = c_{sat}.$$

Two attempts to specify an eddy diffusivity model have been reported in the literature, by Davies and Ting [1], and by Brumfield and Theofanous [6]. Davies and Ting assumed that viscous damping of turbulence along the jet could be ignored, but allowed for damping of turbulence near the liquid–gas interface, following a hypothesis of Levich [7]. Brumfield and Theofanous subsequently modified this approach by allowing for viscous damping of turbulence along the jet using data for the decay of homogeneous turbulence behind a grid. Since numerical calculations show that viscous damping cannot be ignored for the

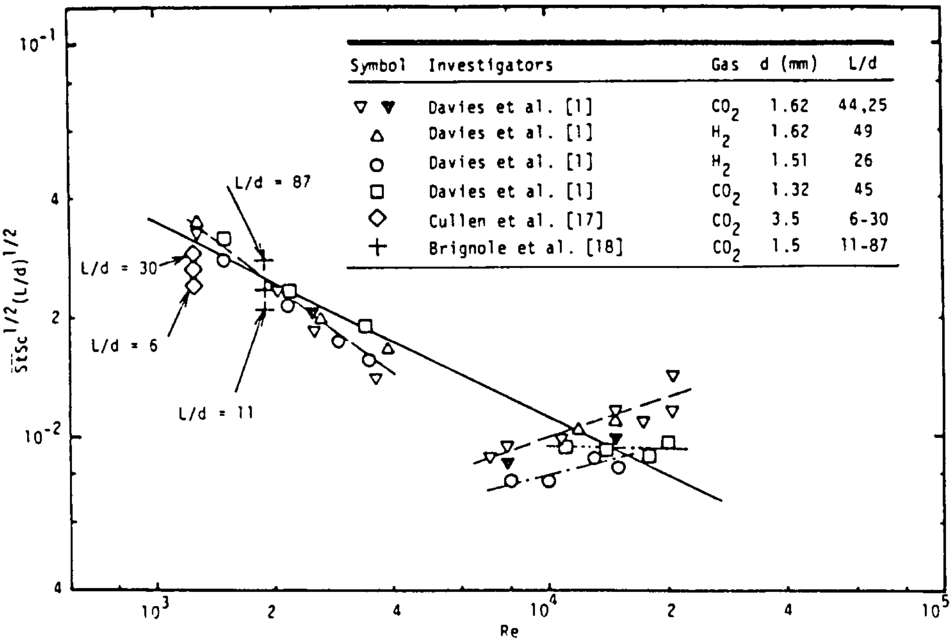

FIG. 1. Experimental investigations of CO_2 and H_2 absorption into round turbulent water jets. Data from refs. [1, 17, 18].

parameter range in question, only the latter model will be considered here. However, Brumfield and Theofanous failed to realize that an entrance region problem required solution, and assumed a fully developed (in some sense) situation. Thus in what follows the essential hypotheses about turbulent transport of Brumfield and Theofanous are incoporated into an analysis of entrance region mass transfer for the jet. The eddy diffusivity model is

$$0 < r < R_c \qquad \varepsilon = v l_c \tag{7}$$

$$0 < R_c \le r \le R \qquad \varepsilon = C(\rho v^3/\sigma)(R - r)^2 \tag{8}$$

$$R_c < 0 \qquad \varepsilon = C(v/R)(R - r)^2 \tag{9}$$

where

$$R_c = R - (\sigma l_c/C\rho v^2)^{1/2} \tag{10}$$

v is the turbulence intensity, l_c is the mixing length in the core, R_c denotes the radius at which damping of turbulence by surface tension commences, and C is (hopefully) a constant.

The viscous damping of v is given as follows: let l be the turbulence macroscale, and $t = x/V$. Then define $T^* = l_0/v_0$ where l_0 and v_0 are values at the nozzle exit, and let $t^* \sim 10T^*$, the duration of the initial decay period. For the initial decay period $0 < t < t^*$

$$v = v_0[1 + (v_0/l_0)t]^{-1/2} \tag{11}$$

$$l = l_0[1 + (v_0/l_0)t]^{1/2} \tag{12}$$

while for the final decay period, $t \ge t^*$

$$v = v^* l^{*5/2}[l^{*2} + 2\pi\nu(t - t^*)]^{-5/4} \tag{13}$$

$$l = [l^{*2} + 2\pi\nu(t - t^*)]^{1/2} \tag{14}$$

where v^* and l^* are values of v and l at t^*. v_0 and l_0 are obtained from pipe flow results as $v_0 = 0.2VRe^{-1/8}$ and $l_0 = 0.03d$. For the core mixing length $l_c = 2l$, but the mass transfer is quite insensitive to this choice.

Levich [7] originally proposed equation (8) for the variation of ε near the interface, with the constant C $\approx$ 1. The approach taken here is to adjust C to fit experimental data for the mass transfer coefficient. Since only water jets at 25°C will be considered, the essential elements of the Levich hypothesis are the distance squared and turbulence intensity cubed dependencies in equation (8).

Equation (6) was solved numerically using finite difference methods. Owing to the very thin concentration boundary layer, care was taken to ensure sufficent node points near the interface. Presented here are the results for a sample case corresponding to parameter values taken from Davies and Ting [1]. For this purpose their correlation using exponents based on the Levich model will be used, viz.

$$K_L = 0.028Re^{1.31}\left(\frac{\mu^3\mathscr{D}}{\rho d^3\sigma}\right)^{1/2} \tag{14}$$

for their "type T" nozzle, which was a long straight smooth walled tube. The parameter values are

$d = 0.151 \times 10^{-2}$ m, $\nu = 0.87 \times 10^{-6}\,\mathrm{m^2\,s^{-1}}$,
$L = 0.08$ m, $\mathscr{D}_{CO_2} = 2.07 \times 10^{-9}\,\mathrm{m^2\,s^{-1}}$,
$Re = 15\,000$, $\rho = 996\,\mathrm{kg\,m^{-3}}$,
$V = 8.64\,\mathrm{m\,s^{-1}}$, $\sigma = 7.2 \times 10^{-2}\,\mathrm{N\,m^{-1}}$.

Davies and Ting report a value of $\bar{K}_L = 5.7 \times$

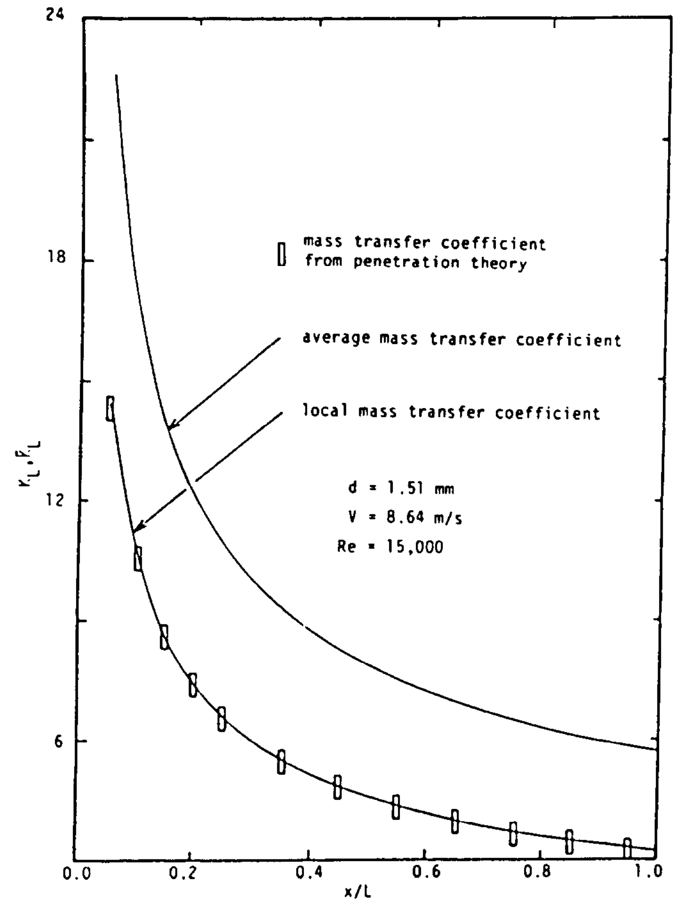

FIG. 2. Calculated variations of K_L and $\bar{K}_L$ as a function of position along a turbulent water jet.

10^{-4} m s^{-1} from which N_{tu} = 0.01398 and ε_M = 0.01388.

Variation of the constant C in equation (8) showed a value of 0.5 matched the experimental results. However, since the value of $\bar{K}_L$ is barely greater than the laminar value, this should not be viewed as a reliable determination of C. Figure 2 shows the variation of K_L along the jet, and Fig. 3 shows concentration and total diffusivity profiles across the jet at three axial locations.

Figure 2 shows that $\bar{K}_L$ varies markedly along the jet and does not attain an asymptotic fully developed constant value, i.e. the mass transfer problem is an entrance region problem. Figure 3 shows the associated concentration profiles and it is seen that, as is characteristic of an entrance region problem, the concentration in the core remains essentially unchanged along the jet. Thus it is impossible to infer c_{out} from a sample of core liquid, as was attempted by Leininger [9] in experiments to be discussed below. Figure 2 also shows K_L from penetration theory, and as mentioned earlier, the laminar values are almost as high as the turbulent values, even though the Reynolds number for this jet is 15 000. One possibility is that the experimental data are in error. On the other hand Fig. 3 shows that the apparent small effect of turbulence on mass transport can be plausibly explained. It can be seen that the concentration boundary layer hardly penetrates into the region of the jet where turbulent transport is significant. As the concentration boundary layer grows in thickness, the turbulence simultaneously decays, and consequently $(\varepsilon + \mathscr{D})/\mathscr{D}$ remains close to unity within the concentration boundary layer. A practical consequence of this observation is that up to relatively high Reynolds numbers, the mass transfer coefficient for the turbulent jet can be taken to be the same as for a laminar jet, for which the simple penetration result can be used. However, it would be useful to have reliable experimental data to confirm this assertion.

In an attempt to reproduce the experimental data of Davies and Ting [1], Leininger [9] built an experimental rig to absorb oxygen into turbulent water jets. Details are given in ref. [9], and only a brief report will be given here. A long straight tube nozzle (similar to the "T-type" nozzle of Davies and Ting) was used. Saturated oxygen at 1 atm. pressure was absorbed in room temperature distilled water: inlet and outlet liquid phase O_2 concentrations were measured using the standard Winkler titration procedure. Following

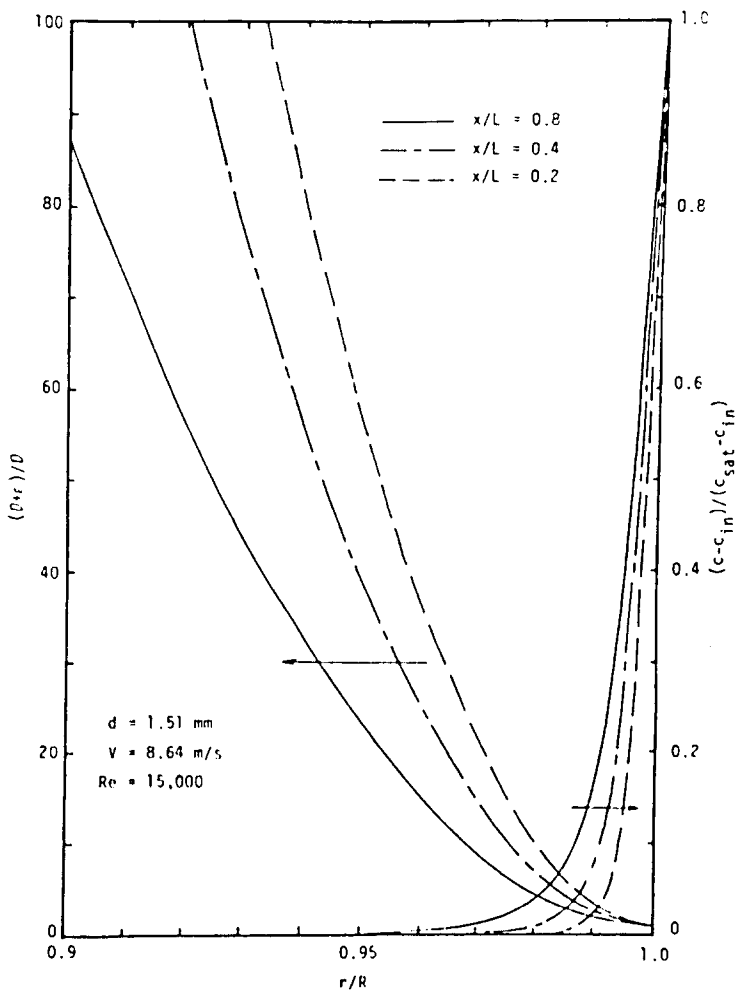

FIG. 3. Calculated eddy diffusivity and concentration profiles in a turbulent water jet.

Davies and Ting, attempts were made to collect the jet with a collector of diameter equal to that of the nozzle. However, it was found that entrainment of oxygen into the outlet stream could not be avoided with such collectors, and gave rise to indeterminate systematic error in the outlet liquid O_2 concentration measurement.

If the mass transfer process is indeed fully developed as assumed by previous workers, then a measurement of the jet core O_2 concentration would suffice. Thus experiments were conducted with a 2.16 mm dia. nozzle and a 0.6 mm collector to eliminate entrainment. The jet length was varied from 3.3 to 6.4 cm and the Reynolds number from 7640 to 12 580. It was found that there was no significant difference in outlet and inlet bulk concentrations, confirming that the fully developed assumption is invalid. Of course Fig. 3 suggests that it is imperative to collect the complete jet in order to make a reliable determination of the outlet bulk concentration. One way to isolate the effect of entrainment would be to use a mixture of two gases of widely different solubilities, e.g. O_2 and CO_2: determination of the apparent mass transfer effectiveness for each gas will allow the entrainment rate to be estimated and corrected for.

HEAT TRANSPORT

In contrast to the mass transfer problem, there have been a number of experimental investigations of heat transport in turbulent liquid jets. Most of these studies have involved condensation of steam on water jets, owing to the application to direct contact condensers. Data for the liquid side heat transfer coefficient, h_L, obtained with evaporating jets can be expected to be similar to that for condensing jets only if a stable jet is maintained, and if effects of factors such as vapor shear are of minor importance. In practice, it is difficult to prevent an evaporating jet from shattering due to cavitation. Not only must the water be sufficiently deaerated, but also the nozzle and manifolding must be designed to prevent flow separations. Nevertheless it is attractive to determine the liquid side heat transfer coefficient in an evaporation experiment, since the

Table 1. Summary of experimental heat transfer investigations

Investigators	Nozzle type	Dimensions of jets d; t (mm)	L (mm)	Re ($\times 10^3$)	L/d	Parameter range T_{in} (°C)	P_{sat} ($\times 10^{-5}$ Pa)
Zinger [10] (1953)	Round short	10, 15	800	200–850	10–80	~10	1.2–1.4*
Isachenko et al. [11] (1971)	Long tube	2.18	20–400	40–80	10–180	~89	1.6–1.7*
Sklover and Rodintin [12] (1970)	Round short	3, 20	200–1200	50–950	8–400	23–65	0.15–0.98*
Zakharov and Chemaya [14]	Punched plate	3.0, 5.1	450	3–15	89–150		1.0*
Barathan et al. [16] (1981)	Slot on a pipe	3.2	750	8–24	235	3–11	0.009–0.04*
Present work	Long tube	4.0	100–170	10–23	24–33	~18	0.01†

* Condensation mode.
† Evaporation mode.

noncondensable gas problem encountered in condensation experiments can be avoided.

The evaluation of experimental data for h_L is made difficult by the wide range of conditions in the various studies. Various types of nozzles have been used, e.g. a long tube, a slotted pipe, or hole punched in a metal plate. Also a wide range of jet velocity, diameter or thickness (for round and planar jets respectively) and water temperature have been tested. Table 1 summarizes the pertinent details of each investigation.

In our work evaporation from round turbulent jets is being investigated. The experimental rig consists of a test chamber and liquid loop, a heat addition system and a heat extraction system. The nozzle is a precision drawn glass tube of 4 mm I.D., and the collector diameter is 5 mm. The inlet and outlet bulk liquid temperatures are measured by pairs of 30 gage chromel–alumel thermocouples, as is the wet bulb temperature in the test chamber. The pressure in the test chamber was measured with a mercury manometer: it is found that the wet bulb temperature accurately approximates the saturation temperature, as expected. The liquid flow rate is measured using Dwyer ball flow meters. The average liquid side heat transfer coefficient is calculated from the single stream exchanger relations, viz.

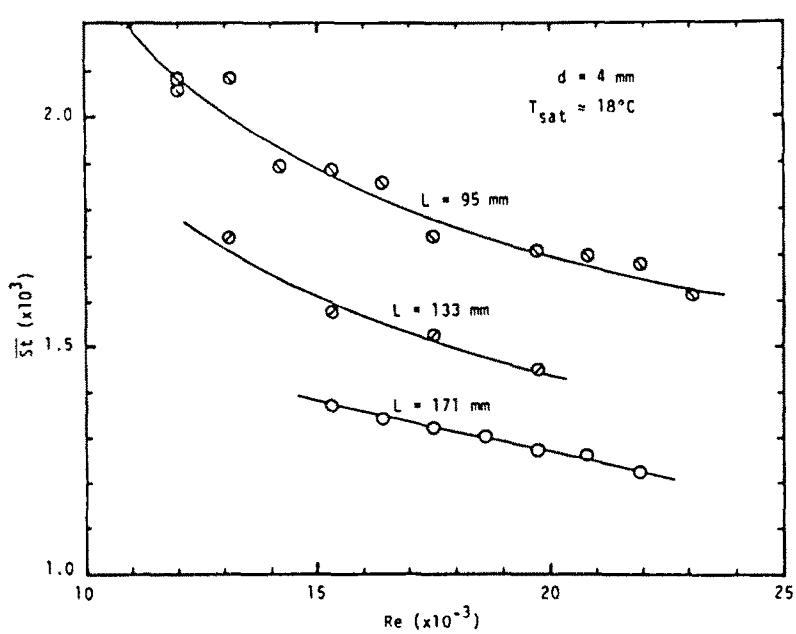

FIG. 4. Average Stanton number vs Reynolds number for evaporation from a turbulent water jet.

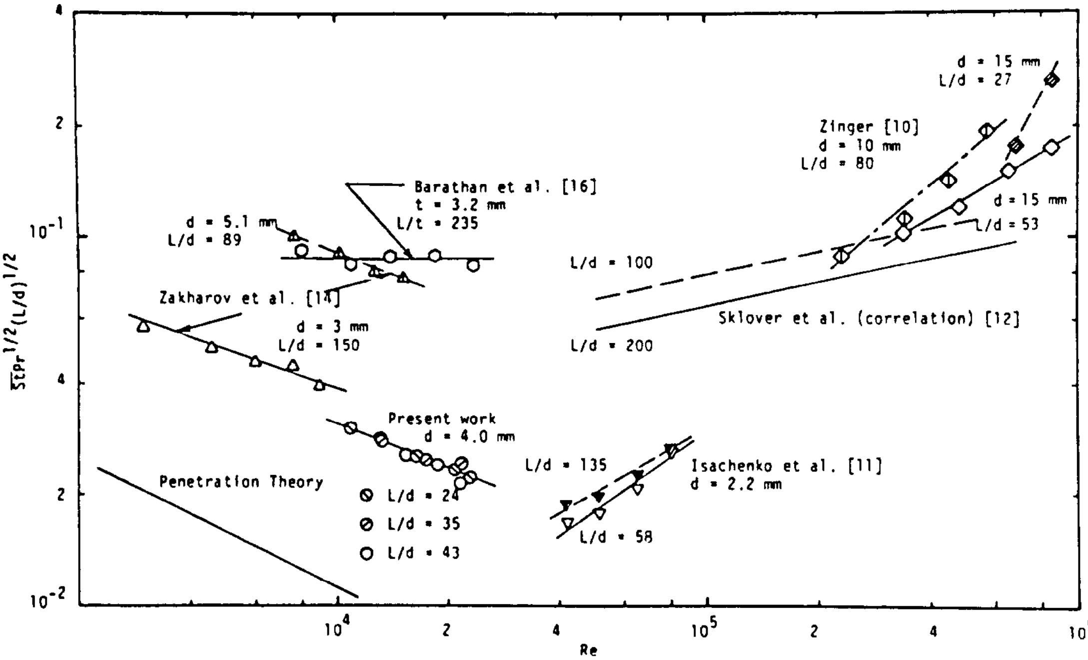

FIG. 5. Comparison of the results of various experimental heat transfer investigations for both condensation and evaporation.

$$\varepsilon = \left(\frac{T_{in} - T_{out}}{T_{in} - T_{sat}}\right) = 1 - e^{-N_{tu}} \quad (15)$$

$$T_{sat} = T_{sat}(P);\ N_{tu} = 4\overline{St}\,L/d.$$

The results are plotted in Fig. 4 as average Stanton number vs Reynolds number for three jet lengths. The maximum Reynolds numbers shown were limited by the onset of jet shattering due to cavitation.

The choice of appropriate variables with which to plot and compare the data of the various investigators is not straightforward, since the dominant transport mechanism might depend on parameter values and nozzle type. In Fig. 5 the data are plotted in a form suggested by penetration theory for laminar flow: the main advantage of this plot is that $\bar{h}_L \propto L^{-1/2}$ dependence of penetration theory does approximately account for the length dependence in some of the data. Although there are large apparent discrepancies in the data, there are some trends which can be discerned. It is also possible to give some plausible reasons for the apparent discrepancies. Our observations are as follows:

(i) At lower values of Re (3000–25 000) the Re exponent is in the range -0.3 to -0.1, which is characteristic of turbulent flows, e.g. ducts or external boundary layers. The absolute values of $\overline{St}$ are 3–11 times larger than predicted by laminar flow penetration theory which also suggests a major role played by turbulent transport. The Reynolds number exponent does not agree with the $+0.31$ of the Levich theory.

(ii) At higher values of Re (40 000–850 000) the Re exponent is positive, suggesting that the physical phenomena controlling heat transfer are different to those at low Re. Jet break up is probably playing a major role with the increased surface area leading to increased heat transfer rates (when based on a nominal surface area determined by the nozzle diameter or jet width). Zinger [10] used a sight glass to make visual observations and reports that the jet was white and opaque, with apparent diameter increasing with distance from the nozzle.

(iii) $\overline{St}$ decreases with increasing jet length. Our data shows $\overline{St} \propto L^{-0.5}$, while Isachenko *et al.* [11] correlated their data as $\overline{St} \propto L^{-0.54}$ for $L/d < 100$. The data of Sklover and Rodintin shown in Fig. 5 was correlated as $\overline{St} \propto L^{-0.75}$ [12], however, in their later work with multiple jets and crossflow of steam [13], the data was correlated with an exponent of -0.5.

(iv) There is some evidence for an increase in $\overline{St}$ with jet diameter or thickness. The data of Zakharov and Chernaya [14] in Fig. 5 shows a very marked effect with $\overline{St} \propto d^{3.2}$, which is in direct contrast to the $-3/2$ exponent given by the Levich theory. Sklover and Rodintin give exponents of 0.25 and 0.5 in their 1970 and 1976 work, respectively [11, 12]. Thus the very high values of St obtained by Zinger may be partly attributable to the comparatively large jet diameters used.

(v) There is evidence that St values for round jets are lower than those for planar jets. Dement'yeva and Makarov [15] was the only study in which the effect of geometry was explored and such a trend was observed, and of course would be expected from the behavior of other turbulent flows. (Insufficient information was given in [15] to enable the data to be plotted in Fig. 5.) Thus the low values of St obtained in the present study might be partly attributable to geometry: the low values of St of Isachenko *et al.* might also be due to the round jet used, but also due to the small diameter, as noted in (iv) above.

(vi) The slotted pipe distributor used by Bharathan *et al.* [16] produced planar jets of very non-uniform thickness and with an appreciable cross-flow velocity: also the jet surface was observed to become very agitated a short distance from the nozzle. Thus the comparatively high values of $\overline{St}$ obtained could be due to abnormal turbulence, and an increased interface area.

(vii) In some of the experiments, e.g. [16], the jet was allowed to fall into a pool, and there was the possibility of further condensation on the highly agitated pool surface before the liquid is collected and its bulk temperature measured. When the ratio of pool surface area to jet surface area is large there is the possibility of an appreciable increase in the apparent value of $\overline{St}$. In the present study the jet was collected in a collector of diameter 20% greater than the nozzle. On one hand no entrainment of steam was observed, but on the other hand it is possible that some low temperature surface liquid was not collected. Thus the measured values of $\overline{St}$ might be low for this reason.

A number of attempts have been made to analyze heat transfer in turbulent liquid jets [11, 14]. However, the eddy diffusivity models used have been rather simple, and have not allowed for viscous damping of turbulence, as described in the previous section (Mass Transport). Thus it is of little value to review these analyses in detail.

CONCLUSIONS

(i) Mass transport in turbulent liquid jets is an entrance region problem and must be analyzed accordingly.

(ii) Available experimental data for gas absorption into turbulent jets does not support the Levich theory: indeed it appears that turbulence transport plays a minor role at Reynolds numbers up to about 15 000.

(iii) There are large apparent discrepancies in data for heat transport in turbulent jets. Some of these discrepancies can rationally be attributed to such factors as nozzle configuration, Reynolds number range and geometry.

(iv) At lower values of Reynolds number (3000–25 000) for which turbulent transport is probably dominating the heat transfer process, the dependence of Stanton number on Reynolds number and jet diameter or thickness does not support the Levich theory.

Acknowledgements—This work was supported by the Department of Energy through the Solar Energy Research Institute University Grants Program, Grant DE-FGO2-SOCS89501. Computer time was supplied by the Office of Academic Computing of the University of California, Los Angeles.

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TRANSPORT DE CHALEUR ET DE MASSE DANS DES JETS LIQUIDES TURBULENTS

Résumé—Des données expérimentales et la théorie du transport de chaleur et de masse dans des jets liquides turbulents sont critiquées. On montre que le transfert massique dans les jets turbulents est un problème de région d'entrée et que les données expérimentales doivent être analysées sous ce point de vue. Une solution numérique, par différence finie, du problème d'entrée montre que le transport par turbulence a un effet minime sur le transfert massique dans le domaine étudié. De nouvelles données expérimentales pour l'évaporation à partir de jets turbulents sont présentées pour un jet de 4 mm de diamètre. Des comparaisons sont faites avec des données antérieures sur la condensation et on essaie d'expliquer les points communs et les différences apparentes.

WÄRME- UND STOFFTRANSPORT IN TURBULENTEN FLÜSSIGKEITSSTRAHLEN

Zusammenfassung—Die vorhandenen experimentellen Daten und die Theorie des Wärme- und Stofftransports in turbulenten Flüssigkeitsstrahlen wurden kritisch überprüft. Es wurde gezeigt, daß der Stofftransport in turbulenten Freistrahlen ein Eintrittsbereichs-Problem ist und daß die experimentellen Daten demgemäß analysiert werden müssen. Die numerische Lösung mit der Methode der finiten Elemente für das Eintrittsbereichs-Problem zeigt, daß der turbulente Transport nur einen Nebeneffekt auf den Stofftransport im untersuchten Parameterbereich ausübt. Neue Versuchsdaten werden für die Verdampfung an turbulenten Freistrahlen für einen Strahl von 4 mm Durchmesser mitgeteilt. Vergleiche werden mit Werten aus verschiedenen vorausgegangenen Kondensationsstudien angestellt, und es wird versucht, die Tendenzen und offenkundigen Widersprüche zu erklären.

ТЕПЛО- И МАССОПЕРЕНОС В ТУРБУЛЕНТНЫХ СТРУЯХ ЖИДКОСТИ

Аннотация — Дан критический обзор экспериментальных и теоретических результатов по тепло- и массопереносу в турбулентных струях жидкости. Показано, что массоперенос в турбулентных струях представляет собой задачу о начальном участке. Поэтому обработка экспериментальных данных должна производиться в соответствии с параметризацией, принятой в задаче о начальном участке. Численное решение методом конечных разностей рассматриваемой задачи показывает, что турбулентный перенос оказывает небольшое влияние на перенос массы в исследованном диапазоне параметров. Представлены новые экспериментальные данные по испарению турбулентных струй диаметром 4 мм. Проведено сравнение с результатами ранее выполненных исследований конденсации и сделана попытка объяснить наблюдаемые закономерности и расхождения в экспериментальных данных.