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Heat transfer from a negatively buoyant wall jet

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Abstract—An experimental investigation has been carried out on the heat transfer characteristics of a turbulent, negatively buoyant, two-dimensional wall jet. The heat transfer to the surface from the jet, the discharge temperature of which is taken as greater than the surface temperature, is measured as a function of distance along the isothermal surface for several values of wall, jet and ambient temperatures. It is found that the total heat transfer rate to the isothermal vertical surface decreases with an increase in the mixed convection parameter. The effect of the surface temperature excess on the downward penetration of the jet and on the heat transfer rate is also investigated. Relevant correlating equations are derived.

INTRODUCTION

BUOYANT jets are frequently encountered in nature as well as in industry. In nature, they are important in meteorological, oceanographic and environmental studies. The rejection of thermal energy and of chemical waste products to the atmosphere and to water bodies involves turbulent buoyant jets, such as the flows emerging from cooling towers or chimneys. Heat extraction and energy storage systems, such as those employed in solar energy utilization, are also often concerned with buoyant jet flows [1, 2].

Laminar buoyant jets have been analyzed in several investigations. These studies have clarified the basic mechanisms underlying such flows. For instance, Mollendorf and Gebhart [3] carried out a perturbation analysis for a vertical, laminar, axisymmetric jet, with a small amount of thermal buoyancy. The laminar buoyant jet flow, with small buoyancy effects, was also studied by Schneider and Potsch [4]. They obtained the first approximation to the effect of buoyancy on the flow by the method of matched asymptotic expansions, for $0.5 < Pr < 1.5$. Several other studies have obtained numerical and experimental results on laminar buoyant jets, as reviewed in ref. [5]. However, turbulent jets are of much greater practical interest and have, therefore, been studied much more extensively [1].

Very little work has been done on jets in which the buoyancy effects oppose the flow, such as heated jets discharged downward. Such flows arise in room fires and in mixed convection transport in enclosures. Energy extraction and heat rejection processes often lead to regions where the jet flow is opposed by the buoyancy force, see, for instance, refs. [6, 7].

The negatively buoyant, axisymmetric, jet has been studied experimentally and analytically, using integral models, as reported by Turner [8] and Seban *et al.* [9]. They found that the flow penetrates to a finite height, where the vertical velocity drops to zero and flow reversal occurs, as expected. They also reported that the flow spreads outward horizontally as it approaches the location of flow reversal. The analysis did not consider the effect of the reverse flow on the jet, which is itself driven by an external momentum source. The idealized circumstances of a point-source jet with momentum and buoyancy input, but zero mass flux at the inlet, were considered in the analysis. Penetration distances were found to be close to the experimentally determined values. But the flow rates and the velocity and temperature distributions were not accurately predicted by the analysis.

In this paper, the rate of heat transfer to an isothermal vertical surface in the mixed convection circumstance of a negatively buoyant, turbulent, wall jet flowing adjacent to the surface is studied in detail. As mentioned above, this problem is of particular interest in room fires and in energy extraction and storage systems. The basic problem considered and the relevant flow configuration are discussed in detail later.

There are many practical situations where turbulent, negatively buoyant, wall flows arise [10]. For example, when the fire plume in an enclosure hits the ceiling, it spreads out and finally turns downward at the corners [11, 12]. At this stage, the flow of the wall jet is downward, whereas the buoyancy force is upward, since the flow is at a temperature higher than that of the surroundings. Similarly, heated flows rising adjacent to two walls may flow along the ceiling, meet at the top and be pushed downward, against the buoyancy force. This results in a negatively buoyant, two-dimensional, free jet. At a later stage in the growth of an enclosure fire, a hot upper layer is established above a relatively cooler lower layer. Then, a buoyancy-induced flow arises in the lower layer, adjacent to the wall which is usually hotter than the neighboring fluid. This flow becomes negatively buoyant as it penetrates into the upper layer, across the interface

NOMENCLATURE

A_0	cross sectional area of the slot through which the jet is discharged
C_p	specific heat of air at constant pressure
D	width of the slot for jet discharge
g	magnitude of gravitational acceleration
Gr	Grashof number, $g\beta(T_0-T_\infty)D^3/\nu^2$
h	local surface heat transfer coefficient, $q''/\Delta T$
H	height of the enclosure
k	thermal conductivity of air
L	height of the isothermal vertical surface
m_{NC}	mass flow rate in the upward natural convection flow for the case when $T_s > T_\infty$
m_0	mass flow rate discharged by jet per unit length of the slot
M_0	momentum flow rate per unit slot length at the jet inlet
Nu_D	Nusselt number based on D, hD/k
Nu_δ	Nusselt number based on δ_p, $h\delta_p/k$
Pr	Prandtl number
q''	local heat transfer flux to the surface
Q	total net heat transfer to the isothermal surface from the heated jet flow, $W\int_0^{\delta_p} q''\,dx$
Q_{IN}	total thermal energy input by the jet, $\rho_0 C_p U_0 A_0(T_0-T_\infty)$
Re	Reynolds number, U_0D/ν
Ri	Richardson number, $g\beta(T_0-T_\infty)D/U_0^2 = Gr/Re^2$
T	local fluid temperature
ΔT	surface temperature excess over the ambient, T_s-T_∞
T_0	discharge temperature of the jet
T_s	temperature of the isothermal surface
T_∞	temperature of the surroundings
U_0	discharge velocity of the jet
V_{max}	maximum velocity in the upward natural convection flow adjacent to the plate for the case of $T_s > T_\infty$
W	width of the vertical plate
x	vertical coordinate distance, measured downward from the slot
X	dimensionless vertical distance, x/D
y	horizontal coordinate distance measured outward from the isothermal surface
Y	dimensionless horizontal distance, y/D.

Greek symbols

β	coefficient of thermal expansion of the fluid
ν	kinematic viscosity of the fluid
δ_p	penetration distance of the jet, measured downward from the slot
δ_H	vertical distance up to which heat is transferred from the jet to the isothermal plate
ρ_0	density of air at the jet discharge
θ	dimensionless temperature, $(T-T_\infty)/(T_0-T_\infty)$
θ_s	dimensionless surface temperature, $(T_s-T_\infty)/(T_0-T_\infty)$.

separating the two regions [13, 14]. Negatively buoyant flows may also result due to the discharge of fluid at a temperature different from the ambient air and water media in which the transport occurs. Such flows arise, for instance, in energy extraction from a solar pond and heat rejection with jets inclined downward [7, 15].

A detailed experimental investigation to study the basic flow characteristics of negatively buoyant wall and free jets was carried out in ref. [16]. An insulated surface was employed in the study on wall jets. The penetration distance δ_p was defined as the vertical distance up to which thermal effects penetrate, as indicated by a sharp increase in the temperature level as an array of thermocouples is moved upward starting from the ambient conditions far downstream. The penetration distance was measured and was found to be largely governed by the mixed convection, or buoyancy, parameter Gr/Re^2, which is also often termed the Richardson number, Ri. All these dimensionless parameters are based on the inlet conditions of the jet, as defined later. The net entrainment into the flow was also determined. It was found that as Gr/Re^2 increases, resulting in larger buoyancy effects, the flow penetration decreases and a stronger reverse flow, in the direction of the buoyancy force, arises. The entrainment into the flow was found to increase with Gr/Re^2, over the range considered, as a result of this increased reverse flow at larger Gr/Re^2.

The effect of the thermal conditions at the wall, i.e. whether it is adiabatic, heated or cooled, on the behavior of negatively buoyant flows has not been investigated in detail. Also, since the wall was adiabatic in the experimental work of ref. [16], no data on the resulting heat transfer was obtained. The dependence of the wall flow on the thermal conditions at the surface and the heat transfer to the wall are of considerable importance in the modeling of the transient heating of the walls in enclosure fires and in the other practical cases mentioned earlier.

It is clear from the above review that the heat transfer characteristics of a turbulent, negatively buoyant, two-dimensional, wall jet are not available in the literature. The objective of the present study was to continue the work initiated in ref. [16] in order to determine the heat transfer from the wall jet to a

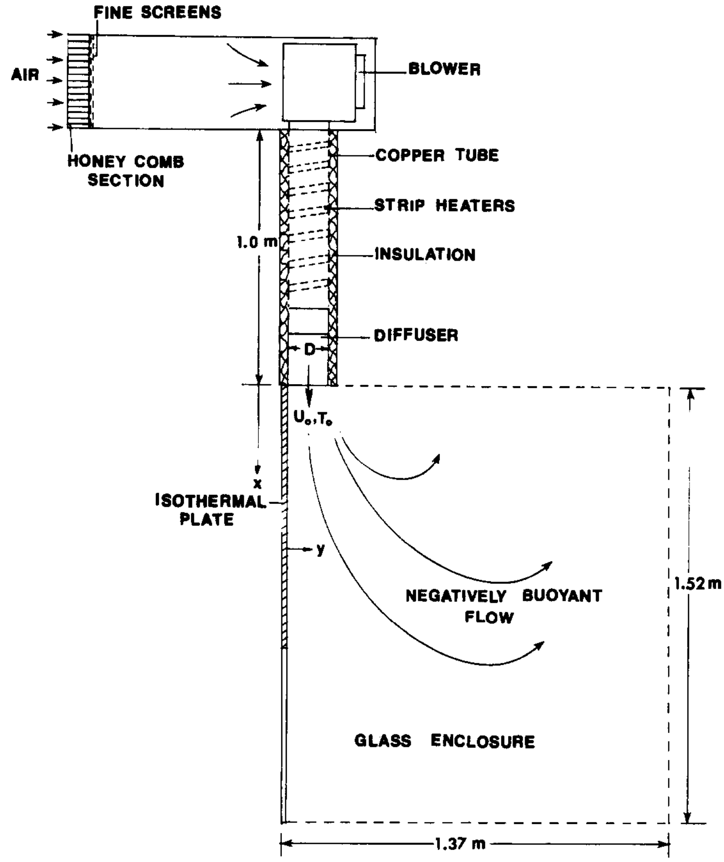

FIG. 1. Experimental arrangement for the study of a heated, two-dimensional downward discharged wall jet in an isothermal ambient medium.

cooled, isothermal surface, with the surface temperature T_s less than the jet discharge temperature T_0. Heat flux data have been obtained for several wall temperatures, varying Gr/Re^2 from 0.05 to 1.05. The local heat flux distribution has been integrated over the surface up to δ_p, the jet penetration distance, to obtain the total heat transfer from the jet to the surface. The effect of the wall temperature excess $T_s - T_\infty$ on the penetration of the jet has also been studied. It is found that, for a given ambient temperature, the penetration distance δ_p decreases as the surface temperature T_s increases, for $Gr/Re^2 > 0.1$. For smaller values of Gr/Re^2, δ_p was found to be essentially independent of the wall temperature excess. The effect of the upward natural convection boundary layer flow, which arises when the surface temperature T_s is greater than the ambient temperature T_∞, is also studied and related to the basic transport mechanisms in the flow. Several very interesting and important trends are observed and considered in terms of the flow characteristics of a negatively buoyant wall jet.

EXPERIMENT

Experimental arrangement

Figure 1 shows a sketch of the experimental arrangement. A blower sends ambient air, over a fairly wide range of flow rates, through a copper tube which is 5 cm in diameter and 1 m in length. The copper tube is heated by means of three fiberglass insulated strip heaters wrapped around it. The energy input to each of the heaters is varied by means of power controllers. A diffuser at the end of the copper tube helps in discharging the heated air as a two-dimensional jet. The width of the slot for discharge may be varied up to about 0.1 m. Several diffuser designs were considered in order to ensure a uniform two-dimensional flow at the exit. Temperature and velocity distributions at the diffuser exit were measured. These indicated that fairly uniform conditions, with variations of less than 10%, were obtained at the discharge location. The heat losses from the copper tube and from the diffuser to the environment were minimized by employing insulation made of foam and fiberglass.

The discharge velocity of the jet U_0 could be varied from about 0.3 m s^{-1} at a slot width of 0.1 m to 3.0 m s^{-1} at a slot width of 0.01 m. The discharge temperature T_0 could be raised to approximately 150°C. Thus, this arrangement could be used for studying fairly wide ranges of the governing parameters Re and Gr, based on the inlet conditions, as defined later. Values as high as around 3500 and 10^6 for Re and Gr, respectively, could be obtained. This study uses a slightly modified version of the system

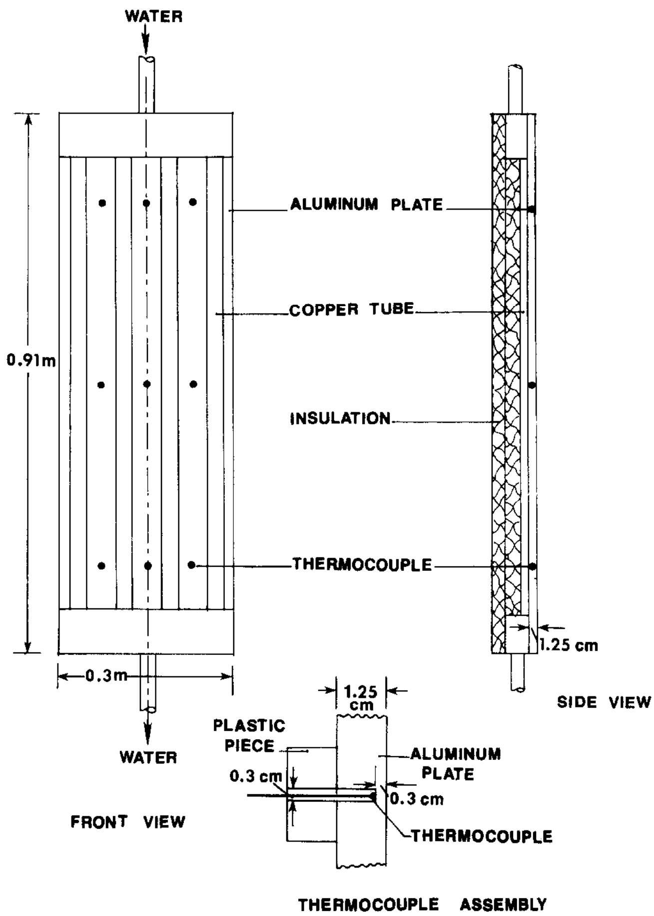

FIG. 2. Sketch of the arrangement used for generating an isothermal vertical surface.

described in ref. [16], which may be consulted for further details.

The heated, two-dimensional, air jet is discharged into a glass enclosure, 1.5 m high and 1.37×0.3 m in cross section. The bottom of the enclosure is kept open in order to allow the wall jet flow to entrain ambient air from below or to flow out of the enclosure at a relatively small velocity, resulting in a negligible effect of the outflow on the negatively buoyant flows under investigation. The top and the far side of the enclosure are also kept open to allow a free exchange with the ambient medium. This arrangement, thus, simulates a wall jet in an extensive ambient medium and, as seen later, in a steady flow circumstance.

A water cooled aluminum plate was located vertically adjacent to the side wall of the enclosure over which the heated jet was discharged. The plate was 91.4 cm in height and 30 cm in width. The remaining portion of the side wall was insulated. The details of the plate design are shown in Fig. 2. Four rectangular copper tubes, each 2.5×1.2 cm in cross section, run along the length of the plate. The tubes were kept flush with the plate surface and were bonded to it by soldering. It was ascertained that the contact between the two surfaces was good by means of temperature measurements. As can be seen from this figure, water circulates over approximately 44% of the plate area. Water from an external tank enters at the top of the plate, with that emerging from this arrangement being allowed to drain into a sink. The temperature of the water entering the plate assembly could be maintained at a desired value by mixing hot and cold water streams from two separate tanks. Nine thermocouples were embedded in the plate to monitor the plate temperature. Details on the thermocouple assembly for plate temperature measurement are also shown in Fig. 2.

Several 0.3 cm diameter blind holes were drilled into the plate. Thermocouples, supported by 0.3 cm diameter glass tubes, were inserted into these holes. It was ensured that the thermocouple beads always touched the plate surface and that a good contact was maintained, so that the plate temperature was accurately obtained. Several similar arrangements have been used by the authors in the past and this background was used in the design of the arrangement. A 2 cm plastic cube was attached to the plate to support each glass tube. Finally, the plate was insulated at the back. It was found that the plate could be maintained at a constant temperature indefinitely. Also, it was confirmed by measurement that a fairly uniform temperature distribution was obtained at the plate surface. The maximum temperature difference measured between any two surface thermocouples was around 0.5°C.

The temperature of the heated jet at the inlet was measured using a rake of five thermocouples. The rake was placed just below the slot through which the jet was discharged. The average of these five temperatures was taken as the jet temperature T_0 in the experiments. The temperature was found to be very uniform, within ± 2°C, across the slot cross section. A DISA constant temperature hot-wire anemometer was employed for the velocity measurements. The hot wire was calibrated by measuring the frequency of the vortices behind a long cylinder [17]. The frequency in the wake allows the determination of air velocity across the cylinder. The velocity distribution at the discharge was again found to be very uniform, within a variation of less than 5%. The average of this distribution was taken to represent the jet discharge velocity U_0. Thus, fairly uniform velocity and temperature distributions were obtained at the discharge, and the measurements yielded the corresponding values of the jet discharge velocity U_0 and temperature T_0. The intensity of the turbulence was found to be less than 5%, as was the intensity of the thermal fluctuations.

The heat transfer to the plate was measured by means of microfoil heat flow sensors (RdF Type 20472-3). Each of the heat flow sensors was 15×6 mm in cross section and 1 mm in thickness. They could easily be attached to the plate surface. The electric output (in mV) from each heat flow sensor was converted into the corresponding heat flux q'' (in W m^{-2}) with the help of individual calibration curves supplied by the manufacturer. The calibration was also checked for accuracy, using electrically heated foils. These heat flow sensors were found to be highly sensitive and accurate. They covered the entire range of heat flux obtained in the experiments. Eighteen such heat flow sensors were attached, at a regular interval of 5 cm, along the centerline of the plate. The sensors were also moved around to confirm the two-dimensionality of the transport process.

The voltage signals from the hot-wire anemometer, thermocouples and the heat flow sensors were fed into a data acquisition system. The signal was also recorded on a strip chart recorder. The fluctuating component was generally found to be small, being less than 10% of the mean temperature and heat flux values. Thus, the mean values, which are of main interest in this study, were obtained easily. The experimental arrangement simulates a negatively buoyant wall jet in an extensive isothermal environment. Considerable care had to be taken to ensure that a high level of repeatability, within 5–10%, was maintained and that accurate measurements of the temperature field and of the heat transfer rate were obtained.

RESULTS AND DISCUSSION

The width D of the long slot discharging the jet is generally taken as the characteristic dimension for a jet in an extensive environment [9, 10]. Then, the dimensionless parameters that govern the flow are the Reynolds number Re, the Grashof number Gr and the Richardson number Ri, which is also often known as the mixed convection parameter, Gr/Re^2. The heat transfer at the wall may be presented in terms of the Nusselt number Nu_D [10]. These dimensionless quantities are defined for a negatively buoyant wall jet in an extensive isothermal medium at temperature T_∞ as

$$Re = \frac{U_0 D}{\nu}, \quad Gr = \frac{g\beta(T_0 - T_\infty)D^3}{\nu^2} \tag{1}$$

$$Ri = \frac{g\beta(T_0 - T_\infty)D}{U_0^2}, \quad Nu_D = \frac{hD}{k} \tag{2}$$

where U_0 and T_0 are the velocity and temperature at the discharge of the jet, g the magnitude of the gravitational acceleration, β the coefficient of thermal expansion of the fluid, h the local convective heat transfer coefficient, given by $h = q''/\Delta T$ where q'' is the local heat flux input at the surface and ΔT the surface temperature excess over the ambient, and ν the kinematic viscosity of air at the discharge temperature T_0.

It must be mentioned that the above dimensionless parameters are obtained when the governing equations and the boundary conditions are non-dimensionalized with U_0, T_0 and D as the characteristic quantities [10]. Another Nusselt number, Nu_δ, based on the penetration distance δ_p, is also defined as

$$Nu_\delta = \frac{h\delta_p}{k} \tag{3}$$

where δ_p is the vertical penetration depth of the thermal effects in the wall jet flow, as defined quantitatively in terms of the thermal field later. If the average heat transfer coefficient $\bar{h}$ is employed instead of h in equations (2) and (3), the average Nusselt number $\overline{Nu}_D$ or $\overline{Nu}_\delta$ is obtained. The dimensionless local temperature θ and the dimensionless surface temperature θ_s are defined as

$$\theta = \frac{T - T_\infty}{T_0 - T_\infty}, \quad \theta_s = \frac{T_s - T_\infty}{T_0 - T_\infty} \tag{4}$$

where T is the local temperature in the flow and T_s the surface temperature.

The data consist mainly of the temperature distributions in the wall jet flow and of the local heat flux measurements along the isothermal surface for different values of the physical variables such as temperatures and flow rates. Results on the flow field were given in ref. [16]. Detailed measurements were taken here for several values of the wall temperature θ_s. The mixed convection parameter $Ri = Gr/Re^2$ was varied from about 0.05 to 1.05 in the experiments, by varying the discharge velocity U_0, slot width D and temperature T_0. However, for convenience, the slot width

Table 1. Physical variables employed in the experiments

Fig. no.	T_0 (°C)	U_0 (m s^{-1})	T_s (°C)	$Gr \times 10^{-6}$	$Re \times 10^{-3}$	Gr/Re^2
3	86.2	0.940	38.0	9.64	28.0	0.123
	101.1	0.702	38.0	9.94	19.4	0.263
4	Varied (75.0–153.0)	Varied (0.43–1.07)	Varied (22.0–48.0)	9.14–9.50	9.6–33.8	0.08–1.04
5	101.4	0.701	22.0	9.95	19.4	0.265
6	122.4	0.578	38.0	9.83	14.5	0.470
7	122.4	0.578	Varied (22.0–48.0)	9.83	14.5	0.470
8–10	Varied (75.0–153.0)	Varied (0.43–1.07)	Varied (22.0–48.0)	9.14–9.49	9.6–33.8	0.08–1.04

$D = 0.065$ m and $T_\infty = 25.0$°C.

D is held constant at 6.5 cm for the results presented here. The effects of varying U_0, T_0 and D were all found to be well correlated in terms of Ri, as found in ref. [16]. Also, even though D is used as a characteristic dimension here, results may be presented in terms of the discharge flow rate, per unit slot length, m_0, momentum inflow rate and energy input rate [5, 8]. For instance, $m_0^2/\rho_0 M_0$, where M_0 is the momentum input per unit slot length at the jet discharge, may be used as a characteristic dimension instead of D. Table 1 gives the physical variables employed for the results presented here.

Figure 3 shows typical vertical temperature profiles taken at $y/D = 0.154$ for the negatively buoyant jet flow at $Ri = Gr/Re^2 = 0.123$ and 0.263. This value of y corresponds closely to the location of maximum temperature in the wall jet. Thus, Fig. 3 indicates the decay of the maximum temperature in the downward wall jet flow with vertical distance away from the jet inlet, $x = 0$. It is seen that the maximum temperature level decreases sharply vertically downward up to a certain distance. This is followed by a more gradual decrease, with the temperature finally becoming equal to that of the surroundings, as expected. The penetration distance δ_p is defined to represent the vertical penetration of thermal effects in the flow. This pen-

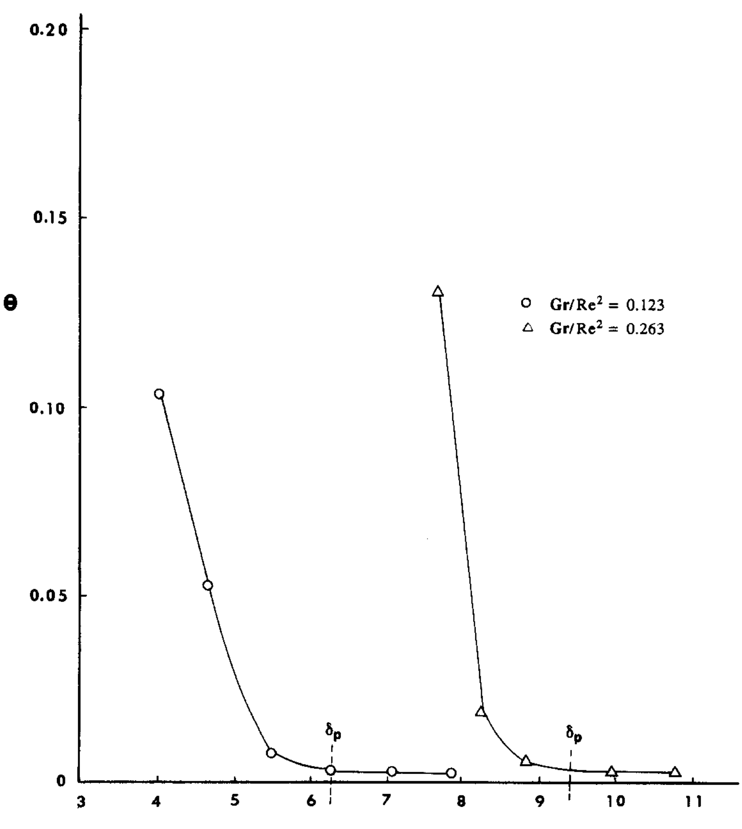

FIG. 3. Typical vertical temperature profiles in the negatively buoyant wall jet flow at $Gr/Re^2 = 0.123$ and 0.263, $y/D = 0.154$ and $\Delta T = 13$°C.

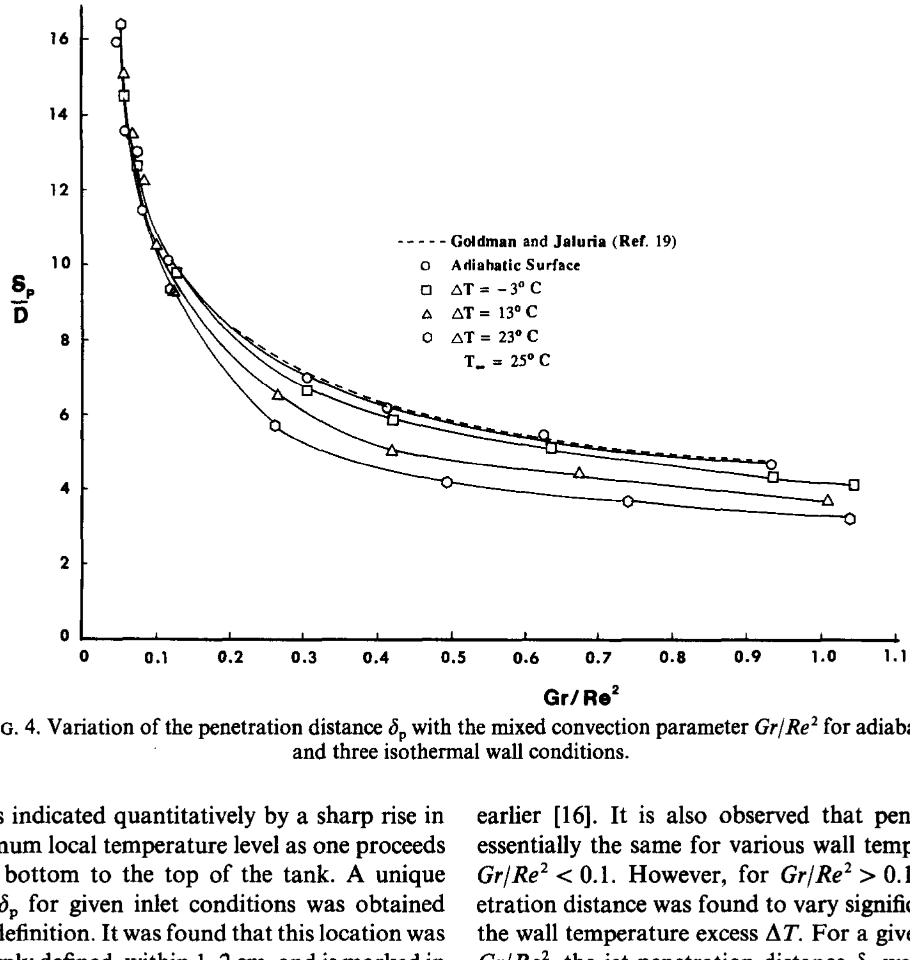

FIG. 4. Variation of the penetration distance δ_p with the mixed convection parameter Gr/Re^2 for adiabatic and three isothermal wall conditions.

etration is indicated quantitatively by a sharp rise in the maximum local temperature level as one proceeds from the bottom to the top of the tank. A unique value of δ_p for given inlet conditions was obtained with this definition. It was found that this location was quite sharply defined, within 1–2 cm, and is marked in the figure. A similar behavior has been observed earlier [16] and repeatability to within 5–10% of the values of δ_p was obtained in separate experiments. The vertical temperature distribution shown in Fig. 3 is important in the understanding of the basic heat transfer and fluid flow characteristics of a negatively buoyant wall jet and will be referred to later.

In fact, δ_p may also be defined as the vertical distance by which the maximum temperature excess $T-T_\infty$ in the downward flow has dropped to, say, 1% of the inlet temperature excess T_0-T_∞. However, such a definition is arbitrary and was found to yield poor repeatability because of the relatively gradual temperature variation far downstream of the discharge. A definition of δ_p based on the temperature profiles shown in Fig. 3 and on a sharp deviation in the distribution due to thermal effects, as x is reduced from large values was found to be appropriate, well defined and repeatable.

The variation of the penetration distance δ_p with the mixed convection parameter Gr/Re^2 for adiabatic and isothermal wall conditions, considering three different typical values of ΔT, the wall temperature excess over the ambient temperature T_∞, are shown in Fig. 4, with δ_p normalized by D. As can be seen from this figure, the penetration distance δ_p for an adiabatic wall condition is very close to that reported earlier [16]. It is also observed that penetration is essentially the same for various wall temperatures if $Gr/Re^2 < 0.1$. However, for $Gr/Re^2 > 0.1$, the penetration distance was found to vary significantly with the wall temperature excess ΔT. For a given value of Gr/Re^2, the jet penetration distance δ_p was found to be smaller for a larger value of the wall temperature excess ΔT. This is an expected trend, since a larger wall temperature excess ΔT results in a smaller heat transfer to the wall. This gives rise to a larger buoyancy effect, which retards the flow more rapidly, resulting in smaller δ_p. Similar trends were observed for other wall temperatures considered in this study.

The effect of the wall temperature on the penetration of the jet also depends on whether T_s is larger or smaller than T_∞. When the surface temperature T_s is higher than that of the surroundings, the surface induces a buoyancy driven flow which originates at the bottom edge of the surface and proceeds vertically upward. This convective mass flow opposes the jet flow which is discharged at the top. The calculations for the resulting laminar natural convection flow in air [10] for a surface temperature of 48°C with $T_\infty = 25$°C, at the locations where the jet was found to reverse direction, show that the maximum natural convection velocity V_{max} is 0.087 and 0.345 m s^{-1}, for $Gr/Re^2 = 0.1$ and 1.05, respectively. Thus, for air, V_{max}/U_0 varies from 0.19 at $Gr/Re^2 = 0.1$ to 0.8 at $Gr/Re^2 = 1.05$. However, the ratio of the mass flow m_{NC} in the natural convection boundary layer to the discharged jet mass flow m_0, m_{NC}/m_0, varies from 0.004 at $Gr/Re^2 = 0.1$ to 0.27 at $Gr/Re^2 = 1.05$. These calculations indicate that the natural convective flow may have a considerable effect on the reversal of the jet flow for $Gr/Re^2 > 0.1$, if the plate temperature T_s is much larger than T_∞. However, in many problems of practical interest, such as the enclosure fire at the early stages of the fire, T_s is quite close to T_∞ and the upward natural convection flow is negligible. This study has considered the circumstances of both negligible and significant natural convection boundary layer flow along the plate.

The variation of the non-dimensional penetration distance δ_p/D as a function of Gr/Re^2 over the range $0 < Gr/Re^2 < 1.0$ may be expressed in terms of appropriate correlations. These were derived from the data obtained for $-5°C < \Delta T < 30°C$ $(0.08 < \theta_s < 0.6)$ and are given as:

for the adiabatic wall condition

$$\frac{\delta_p}{D} = 4.5[Gr/Re^2]^{-0.402}; \qquad (5)$$

for the isothermal wall condition

$$\frac{\delta_p}{D} = (4.1-5.9\theta_s)(Gr/Re^2)^{-(0.4+0.9\theta_s)}; \qquad (6)$$

where $\Delta T = T_s - T_\infty$ and $\theta_s = (T_s - T_\infty)/(T_0 - T_\infty)$. These correlations were found to represent the experimental data closely. The correlation coefficients were about 0.99, indicating this close agreement. In these correlations, the effect of the inlet conditions of the jet is largely represented by Gr/Re^2 and that of the wall temperature by θ_s. The penetration is also seen to be larger for the adiabatic condition, as compared to the isothermal cases, over the ranges considered. This is an expected result, since the buoyancy effect is the largest in this case because of negligible energy loss to the insulated surface.

Figure 5 shows the variation of the dimensionless local heat transfer rate, in terms of the local Nusselt number Nu_D, along the surface for $Gr/Re^2 = 0.265$ and $\Delta T = -3°C$ $(\theta_s = -0.039)$. It is seen that the local heat transfer rate decreases sharply along the plate up to $x/D \approx 5.5$ and then remains essentially constant, at a fairly small value over the remaining portion of the surface. Thus, the local heat transfer to the surface does not become zero even after the jet reverses its direction. This is obviously due to the fact that the ambient temperature T_∞ is slightly higher than the wall temperature T_s, i.e. θ_s is negative. This temperature difference results in heat transfer by natural convection to the surface. The penetration distance δ_p, obtained from the vertical temperature distributions for these experimental conditions, has also been marked in the figure. It is interesting to note that the jet loses most of its thermal energy to the surface within a fairly short distance from the slot at these conditions. This distance represents the region over which the heated jet loses thermal energy to the plate. It is denoted by δ_H and is also shown in Fig. 5. Typically, δ_H was found to be 70–80% of the corresponding penetration distance δ_p in the present experiments. Thus, even though the thermal effects, in terms of temperature disturbances and maximum temperature excess over the ambient, penetrate to a distance δ_p, the heat flux becomes negligibly small upstream of the location of flow reversal. Such trends have been observed in other convective flows as well, such as deviation from the laminar behavior in transition to turbulence [10].

Figure 6 shows the variation of the local Nusselt number Nu_D with the vertical distance x/D for wall temperature excess $\Delta T = 13°C$ $(\theta_s = 0.133)$. The basic trends are similar to those observed in Fig. 5, except that the local heat transfer rate becomes negative after a certain distance, implying heat transfer from the surface to the flow. This distance is δ_H, as defined earlier, and indicates the region over which the surface gains energy from the flow. Some of the corresponding values of δ_H may also be marked on the respective vertical temperature profiles in Fig. 3. It was found that δ_H indicates the vertical location where the local jet temperature near the surface becomes essentially equal to the surface temperature T_s. Thus, from Fig. 6, it is seen that the negatively buoyant jet loses energy to the surface up to a distance δ_H where the local temperature level near the surface in the jet flow becomes essentially equal to the surface temperature. The temperature in the jet flow from δ_H to δ_p is actually lower than the surface temperature for positive θ_s. Consequently, heat is transferred from the isothermal surface to the jet flow in this region. This is shown as Q_- in Figs. 6 and 7 and has been subtracted from Q_+, also shown in the figure, to obtain the total net energy transfer to the plate Q.

The distribution of the local Nusselt number Nu_D along the isothermal surface, at a constant value of Gr/Re^2 and for the three plate temperatures, is shown in Fig. 7. The corresponding penetration distances δ_p have also been marked in the figure. Obviously, the trends are the same as those observed in Figs. 5 and 6. This figure, therefore, indicates the effect of the plate temperature θ_s on the heat transfer rate at a given value of Gr/Re^2. Other wall temperatures and different values of Gr/Re^2 were also considered and similar trends observed.

The variation of the non-dimensional distance for heat loss, δ_H/D, with Gr/Re^2, for different wall temperatures, is shown in Fig. 8. It is seen that δ_H decreases as Gr/Re^2 increases, which is expected since a larger value of Gr/Re^2 implies larger buoyancy effects and thus a shorter penetration distance [16]. These results may be compared with the corresponding results for the penetration distance δ_p shown in Fig. 4. It is seen from Fig. 8 that δ_H is different for different wall temperature excess values, even for Gr/R^2 less than 0.1. This was not found to be the case for the penetration distance δ_p (Fig. 4). From these results, it may be concluded that for small values of Gr/Re^2, even though the penetration distance δ_p is essentially unaffected by the wall temperature excess ΔT, δ_H does vary significantly

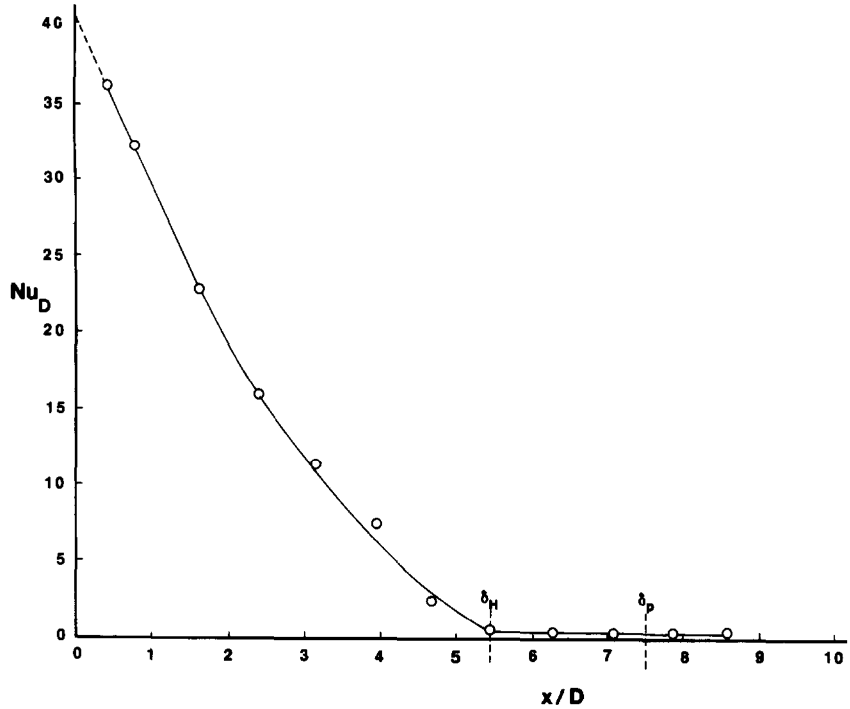

FIG. 5. Local Nusselt number Nu_D distribution along the surface for $Gr/Re^2 = 0.265$ at a surface temperature T_s which is 3°C less than the ambient temperature T_∞ ($\theta_s = -0.039$).

with ΔT. For large values of Gr/Re^2, both the penetration distance δ_p and the heat loss distance δ_H were found to be significantly dependent on the plate temperature excess ΔT.

For various values of Gr/Re^2 and different wall conditions, the local heat flux q'' was obtained as a function of distance x along the plate. Then, the total net heat transfer Q to the plate by the jet was obtained by integrating q'' up to the corresponding penetration distance δ_p and multiplying the result by the plate width W. Thus, for wall temperatures higher than T_∞, the thermal energy transferred by the isothermal plate

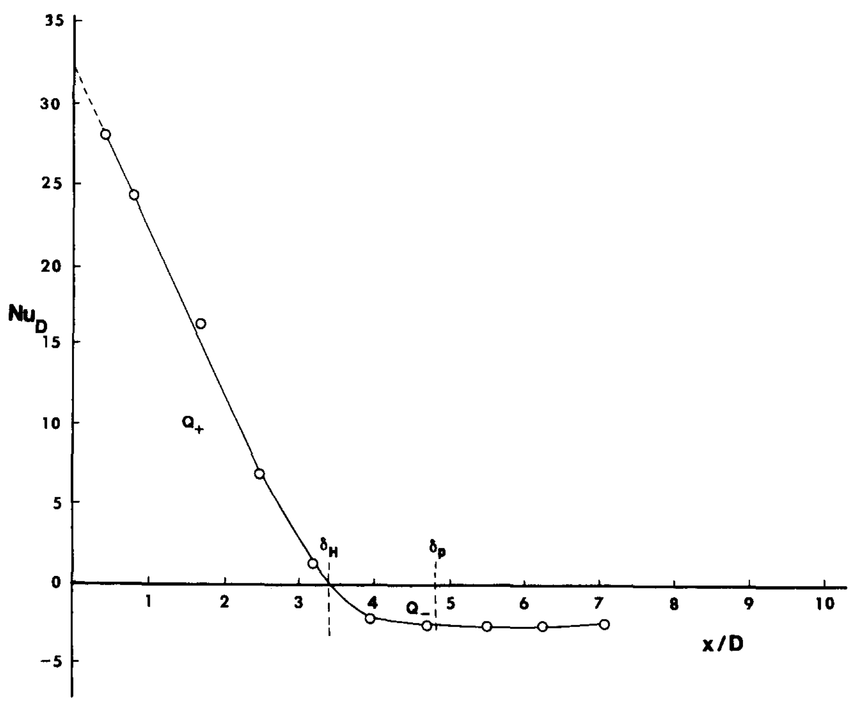

FIG. 6. Local Nusselt number distribution along the surface for $Gr/Re^2 = 0.470$ and a surface temperature excess over the ambient $\Delta T = 13$°C ($\theta_s = 0.133$).

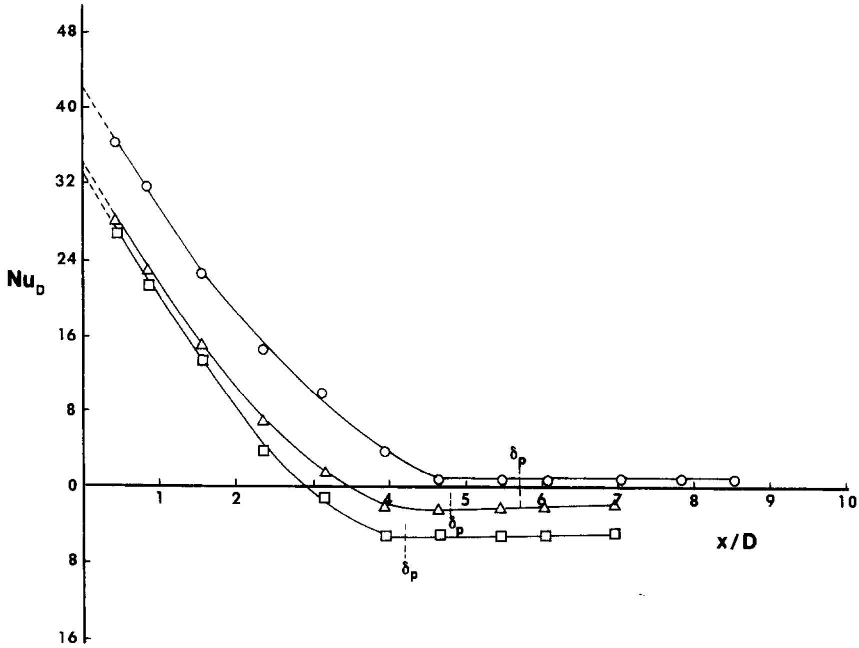

FIG. 7. Variation of the local Nusselt number Nu_D along the isothermal surface at $Gr/Re^2 = 0.47$ for θ_s values of: (a) ○, -0.039; (b) △, 0.133; (c) □, 0.236.

to the jet flow, Q_-, was subtracted from the thermal energy transferred to the plate by the jet Q_+ to obtain Q ($Q = Q_+ - Q_-$). This gives the net energy transferred by the jet to the wall. The natural convection transport beyond δ_p is not considered in this calculation.

The total net heat transfer Q to the isothermal plate by the jet for the three plate temperature excess values considered earlier is shown as a function of Gr/Re^2 in Fig. 9. It is seen that for a ΔT of -3°C, Q decreases sharply up to $Gr/Re^2 \approx 0.25$. This is followed by a gradual decrease up to $Gr/Re^2 = 1.05$. For plate temperature excesses of 13 and 23°C, the variation in Q with Gr/Re^2 is found to be much smaller. In this figure, Q is nondimensionalized by the total energy input by the jet Q_{IN} which is calculated as follows:

$$Q_{IN} = \rho_0 C_p U_0 A_0 (T_0 - T_\infty). \tag{7}$$

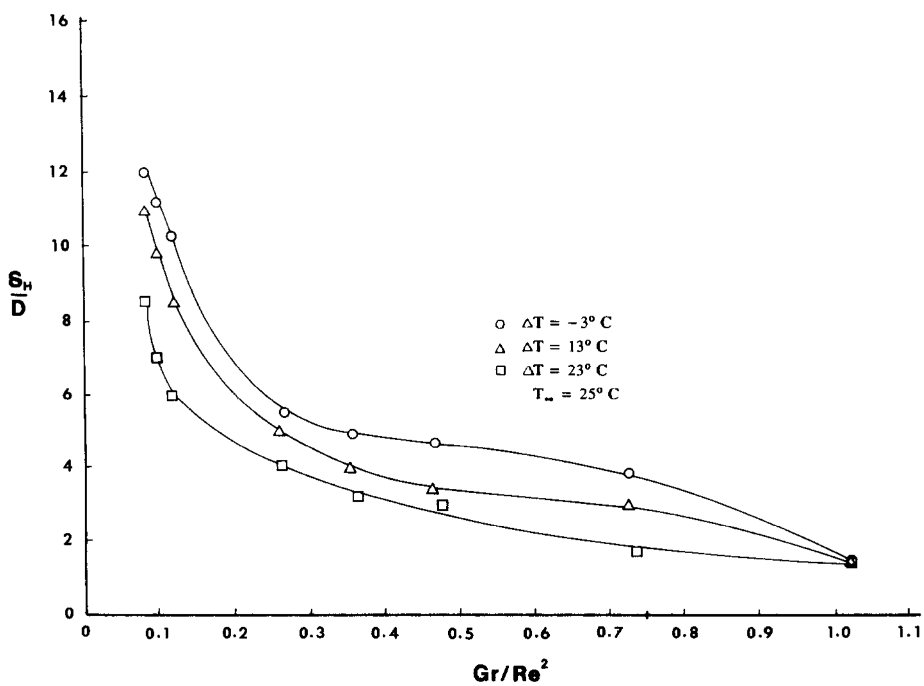

FIG. 8. Variation of the non-dimensional distance for heat loss δ_H/D with Gr/Re^2.

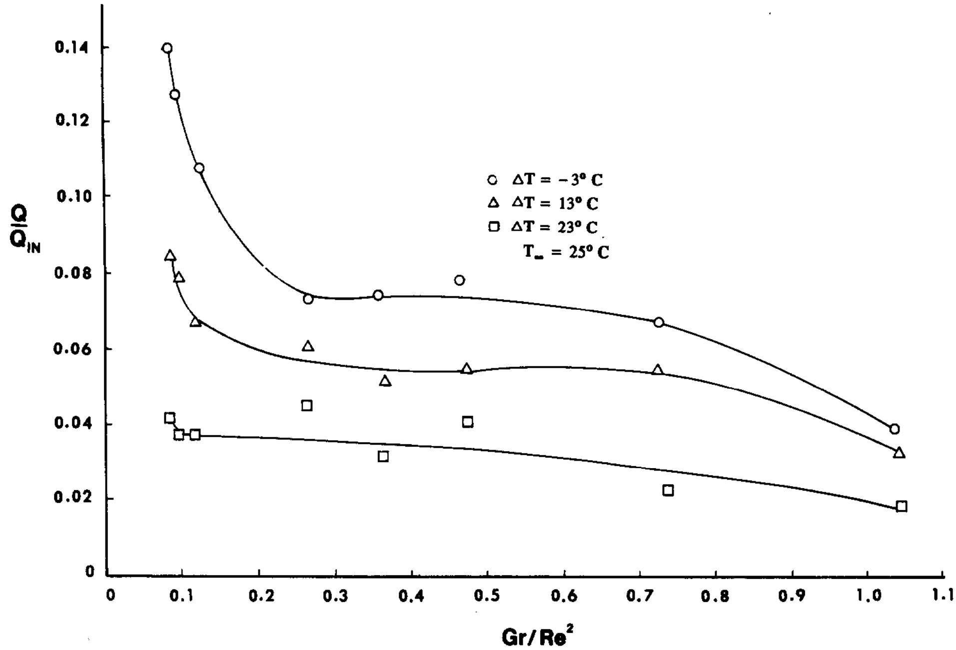

FIG. 9. Variation of the net energy lost to the plate by the jet Q, nondimensionalized by the total energy input Q_{IN}, with the mixed convection parameter Gr/Re^2.

Typically, 3–15% of the jet input energy is transferred to the plate. At large Gr/Re^2, Q/Q_{IN} is found to be weakly dependent on Gr/Re^2, indicating that the penetration distance approaches an essentially constant value as Gr/Re^2 increases to large values. This effect was also seen in Fig. 4.

The corresponding results in terms of the average Nusselt number $\overline{Nu}_D = \bar{h}D/k$, where $\bar{h} = Q/W\delta_p\Delta T$ and $W\delta_p$ is taken as the heat transfer area, are shown in Fig. 10. The trends are similar to those observed in Fig. 9. It is interesting to note that the average Nusselt number $\overline{Nu}_D$ tends to increase to large values as Gr/Re^2 becomes small and to approach zero at large Gr/Re^2. The trends are clearly linked with the dependence of the penetration distance on Gr/R^2. At small Gr/Re^2, δ_p is large and the jet loses much of its energy to the surface, whereas at large Gr/Re^2, δ_p is small and only a small portion of the jet energy is lost to the surface. Also, at a given Re, the inlet temperature T_0 is lower at a smaller Gr/Re^2, resulting in a smaller

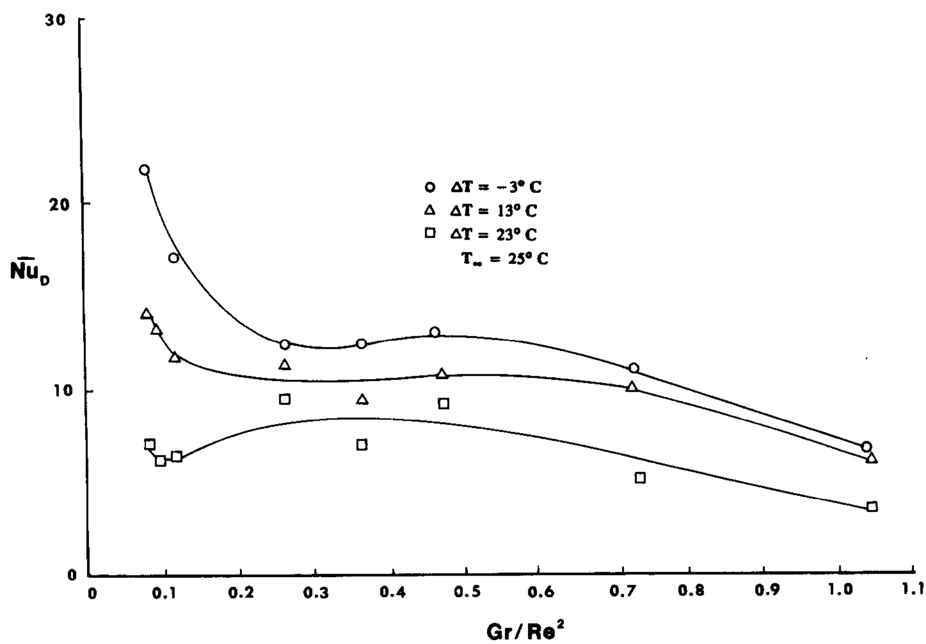

FIG. 10. Variation of the average Nusselt number $\overline{Nu}_D$ with Gr/Re^2.

heat transfer rate to the surface. Thus, there are two competing effects. As Gr/Re^2 becomes smaller, the area for heat transfer increases but the temperature difference $(T_0 - T_s)$ goes down. The combination of these opposing effects also gives rise to a maximum in $\overline{Nu_D}$, as seen in Fig. 10.

It may be mentioned that the net heat transfer Q up to $x = \delta_p$ is considered here. However, only the heat lost by the jet Q_+ may also be considered instead to obtain the heat transfer correlations. The transport process is complicated because of the presence of natural convection if $T_s \neq T_\infty$ and the present approach attempts to focus on the transport process between the jet and the surface, avoiding a consideration of the natural convection transport. Presumably, the jet initially loses energy to the surface as the temperature in the flow approaches T_∞. However, if $T_s > T_\infty$, the jet flow gains energy from the surface for some distance before flow reversal occurs. Beyond $x = \delta_p$, natural convection heat loss occurs. Similarly, if $T_s < T_\infty$, the surface gains energy from the environment by natural convection, which is absent only for $T_s = T_\infty$. This is clearly an interesting and complicated problem which needs a further detailed investigation.

An average Nusselt number $\overline{Nu_\delta}$ based on the penetration distance δ_p may also be defined as

$$\overline{Nu_\delta} = \frac{\bar{h}\delta_p}{k} = \frac{Q}{kW\Delta T} \quad (8)$$

since $\bar{h} = Q/W\delta_p\Delta T$. The variation of this average Nusselt number $\overline{Nu_\delta}$ with Gr/Re^2 can also be plotted for different wall conditions. It was again found that the Nusselt number decreases sharply up to around $Gr/Re^2 < 0.2$. This is then followed by a gradual decrease as Gr/Re^2 increases. Since the effect of δ_p on the Nusselt number is eliminated in this formulation, the variation of $\overline{Nu_\delta}$ is similar to that of Q, as shown in Fig. 9.

The average Nusselt number $\overline{Nu_D}$ varies with the wall temperature excess ΔT and the mixed convection parameter Gr/Re^2, as shown in Fig. 10. Thus, a correlating equation may be derived from the data obtained to yield $\overline{Nu_D}$ as a function of the dimensionless surface temperature excess θ_s and Gr/Re^2. The result obtained is given as

$$\overline{Nu_D} = (10.3 - 13.4\theta_s)(Gr/Re^2)^{-(0.19+0.32\theta_s)}. \quad (9)$$

This equation was found to provide a good representation of the data over the experimental ranges considered. A correlation coefficient of about 0.97 was computed, indicating the accuracy of this correlation.

CONCLUSIONS

An experimental study has been carried out to investigate the heat transfer characteristics of a negatively buoyant, two-dimensional, wall jet. Negatively buoyant flows often arise in many problems of practical interest, such as enclosure fires, thermal discharge into the environment and thermal energy storage systems. A heated, two-dimensional jet is discharged vertically downward adjacent to an isothermal surface in a large enclosure. A water cooled aluminum plate was employed and maintained at the desired wall temperature. The measurements consisted of the local heat flux distribution over the isothermal surface for various values of the mixed convection parameter Gr/Re^2, varying from 0.05 to 1.05. The thermal field in the flow was also measured in detail. The data were obtained for various wall temperatures.

The total heat transfer to the isothermal surface was determined by integrating the local heat flux q up to the jet penetration distance δ_p along the surface. It was found that, typically, the total heat transfer to the surface Q decreases with an increase in Gr/Re^2. This is related to the decrease in jet penetration as Gr/Re^2 is increased. It was also found that, with an increase in the wall temperature excess over the ambient, $T_s - T_\infty$, the total heat transfer from the jet to the surface, Q, decreases, as expected. The effect of wall temperature on jet penetration has also been investigated in detail. It is found that the jet penetrates to a shorter distance when the wall is kept at a temperature higher than that of the surroundings, $\theta_s > 0$, at a constant value of Gr/Re^2. This is attributed to the fact that, when the wall is warmer than the ambient, it induces a vertical upward, natural convection boundary layer flow adjacent to the wall. This flow opposes the jet which is discharged downward, resulting in a reduced penetration distance of the jet. Also the heat transfer reduced, resulting in larger opposing buoyancy effects.

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TRANSFERT THERMIQUE POUR UN JET PARIETAL NEGATIVEMENT FLOTTANT

Résumé—Une étude expérimentale est faite sur les caractéristiques du transfert thermique d'un jet pariétal, turbulent, bidimensionnel et flottant négativement. Le transfert thermique du jet vers la surface, avec une température de sortie du jet plus élevée que celle de la paroi, est mesuré en fonction de la distance le long de la surface isotherme, pour différentes valeurs des températures du jet et de l'ambiance. On trouve que le flux thermique transféré à la surface verticale isotherme décroît quand le paramètre de la convection mixte augmente. L'effet de l'écart de température de la surface sur la pénétration du jet vers le bas et sur le flux thermique transféré est étudié. On obtient des formules intéressantes.

DER WÄRMEÜBERGANG AN EINEM DEM AUFTRIEB ENTGEGENGERICHTETEN WANDSTRAHL

Zusammenfassung—Es wurde eine experimentelle Untersuchung durchgeführt bezüglich der Wärmeübergangs-Eigenschaften eines turbulenten, abwärts gerichteten, zweidimensionalen Wandstrahls. Der örtliche Wärmeübergang vom Strahl an die isotherme Wandoberfläche wird entlang dieser Oberfläche für verschiedene Werte von Wand-, Strahl- und Umgebungstemperatur gemessen. Es wurde herausgefunden, daß der Gesamt-Wärmeübergang an der isothermen vertikalen Oberfläche mit zunehmendem Misch-Konvektionsparameter kleiner wird. Der Einfluß der Temperaturdifferenz zwischen Strahl und Wand auf die abwärtsgerichtete Strahllänge und auf das Wärmeübertragungsvermögen wurde ebenfalls untersucht. Korrelations-Gleichungen werden abgeleitet.

ТЕПЛОПЕРЕНОС ОТ ПРИСТЕННОЙ СТРУИ С ОТРИЦАТЕЛЬНОЙ ПЛАВУЧЕСТЬЮ

Аннотация—Экспериментально исследуются характеристики теплопереноса турбулентной двумерной пристенной струи с отрицательной плавучестью. Теплоперенос к поверхности от струи, температура которой при истечении превышает температуру поверхности, рассматривается как функция расстояния вдоль изотермической поверхности для нескольких значений температуры стенки, струи и окружающей среды. Найдено, что суммарная интенсивность теплопереноса к изотермической ветикальной поверхности уменьшается с увеличением коэффициента смешанной конвекции. Изучено также влияние избыточной температуры поверхности на распространение струи вниз и на интенсивность теплоперноса. Выведены соответствующие корреляционные зависимости.