Int. J. Heat Mass Transfer. Vol. 33, No. 6, pp. 1347–1357, 1990
Printed in Great Britain

0017–9310/90 $3.00+0.00

Natural convection heat transfer characteristics of a protruding thermal source located on horizontal and vertical surfaces

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(*Received* 13 *January* 1989 *and in final form* 11 *September* 1989)

Abstract—An experimental study of natural convection heat transfer from a heat source module of finite thickness, mounted on a vertical or horizontal surface, is carried out. This problem is of particular interest in the removal of thermal energy from heated elements in manufacturing and electronic systems. A wall plume, adjacent to the surface, arises if the heat source module is mounted on a vertical surface. A freely rising thermal plume is generated by the heat source when it is located on a horizontal surface. The temperature distributions in the flow and along the surface are measured. The results obtained indicate that the natural convection flow and the associated heat transfer characteristics vary strongly with the rate of energy input and the source thickness. A comparison between the free and wall plume circumstances indicates very different temperature distributions. While the resulting isotherms for the free plume case are symmetric about the source, those for the wall plume case are swept downstream of the source. It is found that conduction along the plate length is of considerable importance in the transport process. The downstream locations of the plate come under the effect of the wake for the wall plume case, whereas the region far from the source is free from the wake effect in the free plume circumstance which arises in the horizontal orientation. The interaction between the wakes generated by the three exposed surfaces of the heat source module is also found to have a significant effect on the flow. Thus, the heat transfer from a protruding heat source module cannot be accurately determined by assuming three heated, isolated surfaces. The results obtained are compared with those for a source of negligible thickness and the effect of a significant module thickness on the convective flow and transport is determined. The heat transfer coefficient for a protruding heat source module is found to depend on the ratio of the vertical to the horizontal surface areas of the module, exposed to the ambient medium. Several other interesting trends are observed in this investigation and discussed in terms of the underlying flow and thermal transport processes.

INTRODUCTION

MANY PRACTICAL situations involve the generation of a buoyancy-driven wake above an isolated heat source. These include devices such as heaters and electronic components which dissipate energy at a constant rate and which are often mounted on surfaces. The removal of this energy is frequently only by natural convection, where an externally induced flow is not provided. This circumstance is important in several areas of practical concern, such as the cooling of electronic equipment, positioning of heated elements in furnaces and safety considerations in enclosure fires. The flow arising from a protruding heat source located on a surface is of particular interest in many of these and similar problems. The buoyancy-driven flow arising due to the input of thermal energy alone is called a free plume if the flow is not constrained by the presence of walls, and a wall plume if the flow arises adjacent to a wall. The buoyant flow above a protruding heat source mounted on a horizontal surface and a vertical surface may, therefore, be referred to as a free plume and a wall plume, respectively. In these flows, the conjugate heat transfer from the heat source module, due to the coupled effects of conduction in the plate and convection in the fluid, the plume characteristics, and the effect of the thickness of the module on the flow and on the heat transfer are important aspects to be considered in detail.

Considerable work has been done on the important basic mechanisms that arise in these flows, employing several idealizations. The flow due to a single line thermal source on a vertical, adiabatic wall was first considered by Zemin and Lyakhov [1], who carried out an experimental and analytical study of the flow. Jaluria and Gebhart [2] conducted a theoretical investigation of this problem for a wide range of fluid Prandtl numbers, using a similarity technique. Recently, Milanez and Bergles [3] presented a theoretical study of this problem and compared their results with experimental data. They pointed out that heat conduction in the fluid along the main flow direction as well as in the plate near the heat source must be included in the analysis. The similarity variable approach could not be applied for studying the flow over a finite size thermal source and the corresponding laminar boundary layer equations were solved by finite difference methods by Sparrow *et al.* [4].

Some experimental work has also been done on this problem by several investigators. The turbulent wall

NOMENCLATURE

A	total surface area of the module exposed to the fluid
g	gravitational acceleration
Gr	Grashof number, defined in equation (2)
h	local convective heat transfer coefficient
$\bar{h}$	average convective heat transfer coefficient
H	thickness of the heated module
k	thermal conductivity of the fluid
L	length of the heated module, shown in Fig. 1
Nu	local Nusselt number for heat transfer from the source, hL/k
Nu_c	local Nusselt number at the mid point of the module surface BC, see Fig. 1
$\overline{Nu}$	mean Nusselt number for the heat source
Pr	Prandtl number of the fluid
q	uniform heat flux input at the module surface due to the electrical power dissipation
q_{cond}	heat flux conducted from the heat source to the test plate over the surface AD, see Fig. 1
q_{conv}	heat flux convected from the heat source to the flow over the exposed surfaces of the module
Q	total heat input, per unit width, into the module, $q(L+2H)$
Ra	Rayleigh number, $Gr\ Pr$
s	distance along the module surface, see Fig. 6
T	local temperature in the fluid
T_a	ambient temperature
T_s	local surface temperature
ΔT	local temperature excess above the ambient temperature
ΔT_m	maximum source surface temperature excess above the ambient temperature
W	width of the heated module
x	coordinate distance along the surface from the edge of the module, see Fig. 1
X	dimensionless x coordinate, defined in equation (1)
y	normal coordinate distance from the plate surface, see Fig. 1
Y	dimensionless y coordinate, defined in equation (1).

Greek symbols

β	coefficient of thermal expansion of the fluid
θ	dimensionless local temperature, defined in equation (1)
ν	kinematic viscosity of the fluid.

plume was considered by Grella and Faeth [5], who reported velocity and temperature measurements. Carey and Mollendorf [6] measured the temperature field in the vicinity of isolated, circular and square heat sources mounted on an adiabatic wall in water. They discussed the increased complexity due to the three-dimensionality of the flow associated with a heat source of finite dimensions. Robinson and Liburdy [7] investigated the natural convection transport above a heated disk, facing upward and located on a horizontal plate.

Some work has also been done to investigate several other aspects related to these flows. Zinnes [8] and Kishinami and Seki [9] considered the heat conduction effects in the plate, employing finite difference methods. They found that the most important factor for this conjugate problem was the plate–fluid thermal conductivity ratio. Goel and Jaluria [10] studied experimentally the two-dimensional natural convection flow arising from an isolated finite size heat source located on vertical and inclined surfaces. They showed that flow separation from the surface occurs at large inclinations and affects the thermal transport near the separation region substantially. In all these investigations, the heat source was mounted flush on the surface, i.e. the source was of negligible thickness. Not much attention has been given to the natural convection transport from a protruding heat source module on a surface. Recently, experimental and numerical studies have been carried out on the natural convection heat transfer from a protruding heat module mounted on an adiabatic, vertical wall [11, 12].

The present study concerns the natural convective thermal transport from a protruding heat source module, whose thickness is varied and which is mounted on a vertical or a horizontal surface. A detailed experimental investigation is carried out. The temperature distributions in the flow and at the surface are obtained by measurement. The effects of the heat input rate and of the thickness of the heat source module on the convective heat transfer coefficient and on the thermal field, particularly in the vicinity of the source, are studied. The conjugate transport effects, arising due to conduction in the plate, are also considered. Both free and wall plume circumstances are investigated in terms of the underlying physical mechanisms and comparisons are made between the results obtained for the two configurations.

EXPERIMENTAL ARRANGEMENT

An experimental arrangement was designed for detailed measurements of the thermal field near the heat source module. The geometry and the coordinate system, corresponding to a protruding heat source module, are shown in Fig. 1. The width of the test plate and of the heat source module in the transverse direction is taken as 40 cm, which is much larger than the expected boundary layer thickness, so that a two-dimensional flow situation is obtained. The test plate is made of masonite (thermal conductivity around 0.14 W m^{-1} K^{-1}) and is 3 mm thick. The flow from the sides is avoided by using side walls, again to maintain the two-dimensionality of the flow. The heat source module consists of a highly polished stainless steel foil (0.03 mm thick) in close contact with the three exposed surfaces of a stack of bakelite strips, each strip being 2.2 cm wide, 1.5 mm thick, and 40 cm long. The fourth surface is held in good contact with the plate by force exerted at the ends of the thermal source. Similarly, a good contact is maintained between the foil and the bakelite stack. An epoxy cement was employed to ensure good contact between the strips. The thickness of the module could be varied by changing the number of strips in the stack. Similarly, the length L of the source could be changed by using strips of different widths. The stainless steel foil is electrically heated to provide a uniform heat flux input condition over the entire module surface exposed to the ambient fluid.

The heat source module is mounted firmly on the test plate by means of a specially designed clamp. In order to reduce the conduction heat loss from the heat source, enclosed air gaps are used in the design of the plate. The detailed cross-section of the test plate is

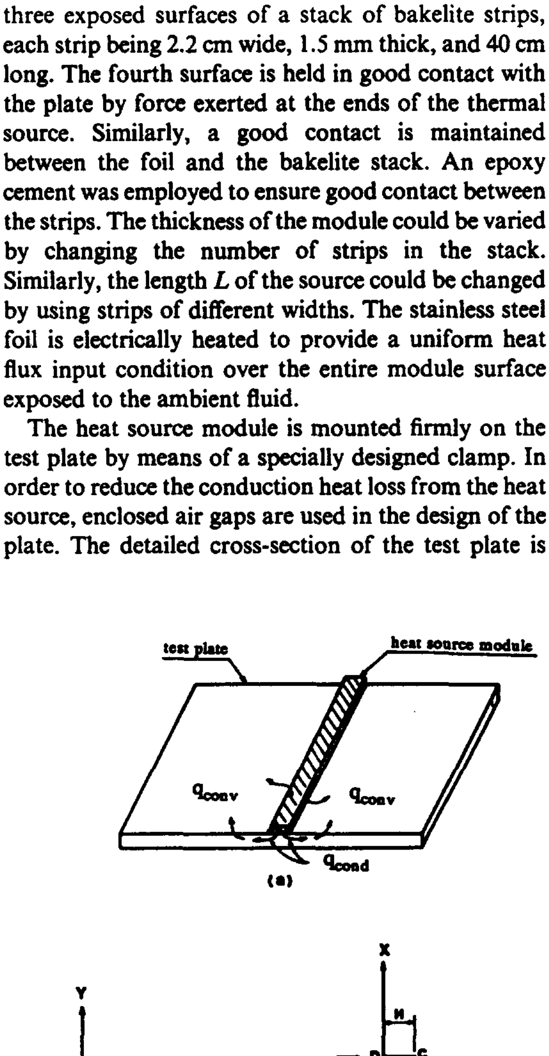

FIG. 1. Sketches of the configurations under consideration and the corresponding coordinate systems. (a) Schematic drawing. (b) Free plume arising from the source on a horizontal surface. (c) Wall plume arising from the source on a vertical surface.

shown in Fig. 2. The thickness of the air gaps is kept small, so that the Rayleigh number based on the thickness of the layer of air is less than 1000 in order to avoid the onset of natural convection. Then, the energy transfer through the air gaps is largely by conduction. This energy loss from the source to the backside of the plate is estimated to be less than 10% of the total electrical heat input and is, thus, lost from the flow. However, most of the thermal energy conducted along the plate is not lost from the system but is transferred to the flow downstream of the source [10]. The conduction from the source to the plate involves transport through the air gaps and along the plate length. Only the former is lost from the system, since the latter finds its way back into the flow downstream. Also, the former component is small whereas the latter affects the flow substantially, as will be seen later. Radiative losses are also kept small by employing a polished stainless steel foil, whose emissivity was measured as around 0.1. The radiation heat loss to the ambient medium was calculated by means of a simple analytical model to be less than 3% of the total electrical power dissipation for the heat flux range considered. Therefore, considering these conduction and radiation effects, about 90% of the total energy input to the thermal source is transferred to the ambient medium by natural convection.

The electric power dissipation at the heat source surface is obtained by measuring the voltage drop across the strip and the current through it by means of precision digital voltmeter and ammeter, respectively. Three high temperature heat flux gages are mounted between the heating foil and the bakelite strips to measure the conductive heat flux from the foil to the test plate on the three surfaces AB, BC, and CD (see Fig. 1). In addition to these gages, high sensitivity heat flux gages are also mounted at various locations on the vertical plate in order to obtain the local convective heat flux from the plate. The heat flux data are recorded by means of a data acquisition system (Keithley, Series 500). The conduction along the heated foil was also estimated and was found to have a negligible effect on the heat flux measurements.

The surface temperature is measured using a set of 30 thermocouples (copper–constantan, 0.025 mm diameter wire), located at the backside of the foil and attached to the module and the vertical surface by means of a high thermal conductivity cement. The temperatures in the flow are measured by an array of thermocouples (copper–constantan, 0.05 mm diameter wire). These were calibrated and the signals were measured by means of a data acquisition system, which employed an A/D converter with an Apple microcomputer. Data analysis and graphics were carried out on a Sun computer system. In most cases, the temperature difference from the ambient was measured. The error was estimated to be within about $\pm 3\%$ of the measured value of the temperature difference ΔT [13]. Temperature data included measurements of the plate surface temperature, tem-

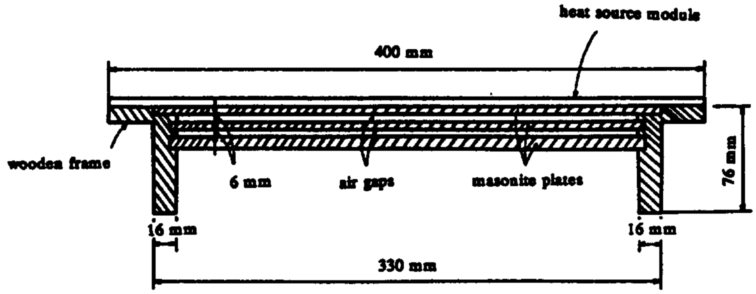

FIG. 2. Cross-section of the test plate, indicating air gaps to reduce conductive transport to the plate.

perature distribution over the surface of the heat source module and temperature distributions in the flow. The heat transfer coefficients were obtained from the measurements of the temperature difference ΔT and of the surface heat flux. An error of less than 5% was estimated in the values of convective heat transfer coefficients determined in the study.

EXPERIMENTAL RESULTS AND DISCUSSION

The coordinate systems for the two configurations under consideration are shown in Fig. 1. The characteristic length for this problem is taken as the length L of the heat source module. This dimension characterizes the flow in the vicinity of the source and determines the vigor of the flow, indicating whether nonboundary layer effects are important, as considered in detail by Jaluria [14, 15] for $H = 0$. Far downstream, the effects due to source dimensions are expected to become small and the distance from the source will be a more appropriate characteristic length. The total heat input Q into the heated element, per unit width, is given by $Q = q(L+2H)$. The measured temperature is normalized by a characteristic temperature difference Q/k, instead of $T_s - T_a$, where T_s is the surface temperature, since T_s is nonuniform over the surface and is not known at the onset. Therefore, the experimental results are presented in terms of the following nondimensional variables, which are similar to those used frequently for natural convection boundary layer flows over heated surfaces with the uniform heat flux condition [10, 14–16]. Thus, the nondimensional temperature θ and coordinate distances X, Y are defined as

$$\theta = \frac{(T-T_a)}{Q/k} Gr^{1/5}, \quad X = \frac{x}{L}, \quad Y = \frac{y}{L} Gr^{1/5} \quad (1)$$

where Gr is the Grashof number, defined as

$$Gr = \frac{g\beta Q L^3}{k\nu^2}. \quad (2)$$

Here T is the local temperature, T_a the ambient temperature, k the thermal conductivity of the fluid, x the physical distance measured from the edge of the source as shown in Fig. 1, y the normal distance from the plate, g the magnitude of gravitational acceleration, β the fluid coefficient of thermal expansion and ν the kinematic viscosity. The properties of the fluid are evaluated at the film temperature $T_f = [(T_s)_{max} + T_a]/2$, using the property data given for air in ref. [17]. Various values of the Grashof number and of the module thickness were used to obtain a range of experimental conditions. Results were obtained for H/L ranging from 0 to 1.0 and for Gr ranging from 10^5 to 10^7. Only a few characteristic results are presented in this paper for brevity. It was found that the results were well correlated in terms of the above dimensionless variables.

Typical isotherms for the vertical configuration, as well as those for the horizontal configuration, are shown in Fig. 3 for $H/L = 0.4$. These isotherms are obtained by employing interpolation on the measurements of the thermal field to determine the locations in X and Y where the specific values of θ shown in the figure are obtained. As seen in this figure, the flow over a heated module located on a horizontal surface is simply a vertically rising, buoyancy-driven plume. Consequently, the thermal effects due to the presence of the source are fairly localized and the plate temperature is not affected by the thermal wake far from the source, i.e. distances larger than about twice the source length L from the source middle point. The plate surface is heated by conduction of thermal energy from the module through the plate. The consequent isotherms are symmetric, as expected. However, in the wall plume case which arises for the vertical orientation, the thermal wake from the heat source module rises along the vertical plate. Thus, the downstream locations come under the effect of the wake, as seen from the thermal field that arises downstream in Fig. 3(b).

The variation of the surface temperature is important in the interpretation and understanding of the thermal transport mechanisms operating in the flow and within the plate. The surface temperature is also relevant to the consideration of the downstream behavior of the flow and the thermal effects due to the buoyancy-induced wake. The surface temperature distribution, measured by means of embedded thermocouples at various locations on the surface and at the heat source, is shown in Fig. 4 for $H/L = 0.4$ at various Grashof numbers, for the two orientations. As

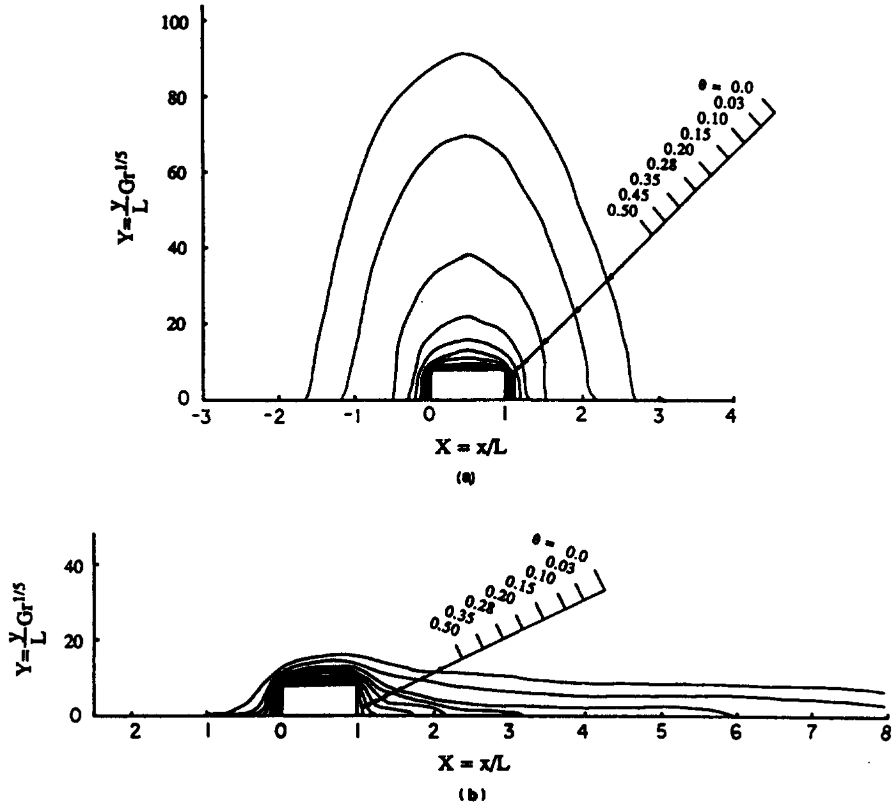

FIG. 3. Experimentally obtained isotherms in the buoyancy-induced flow. (a) Horizontal configuration, $H/L = 0.4$, $Gr = 2.6 \times 10^6$. (b) Vertical configuration, $H/L = 0.4$, $Gr = 2.3 \times 10^6$.

seen in Fig. 4(a), the surface temperature distributions are symmetric about the source for the horizontal circumstance. The maximum temperature is observed at the middle point of the top surface BC of the module, i.e. at $X = 0.5$. This is the expected behavior for a vertically rising free plume generated by a finite size horizontal source. These trends are expected to be quite different for the vertical circumstance, as seen below. The nondimensional value is found to increase slightly with an increase in Gr, indicating the expected behavior for an increased heat input Q. However, the change in the physical temperature is much larger and the nondimensionalization used reduces the corresponding change in θ, as desired for a good correlation of the data in terms of the dimensionless variables. While the idealized distribution used in analysis has often been a step variation in the surface temperature or in the heat flux over the module surface, the surface temperature distribution is experimentally found to be quite different and to spread out over a distance which is much larger than the module length L. This implies that the effects due to conduction along the plate, i.e. in the X direction, are significant and must be included in the analysis. These aspects are again considered in detail later.

The surface temperature distributions for the vertical configuration are shown in Fig. 4(b). The maximum is found to be shifted downstream, as expected. The maximum temperature is observed on the top surface CD of the module (see Fig. 1(c)). This maximum temperature is seen to be only weakly affected by the Grashof number, indicating a good representation of the results in terms of the chosen dimensionless variables. However, the surface temperature in the downstream region ($X > 1.0$) is found to be affected more strongly by the Grashof number Gr. This indicates that the thermal wake contributes to an increase in the surface temperature downstream, a higher Gr leading to a higher surface temperature. It is interesting to note that the surface temperatures upstream of the source ($X < 0.0$), for the vertical configuration, are found to be lower than the corresponding values for the horizontal configuration. This indicates that the vertical natural convection flow largely heats up the downstream portion of the plate, leaving the upstream portion largely unheated. This is physically expected. However, the strong asymmetry of the distribution indicates the dominant effect of the wake downstream, unlike the horizontal circumstance.

The magnitude of the temperature rise above the ambient temperature, $\Delta T_m = (T_s)_{max} - T_\infty$, is a very important parameter in the design of thermal systems, such as a cooling system for electronic components. The reliability and operation of electronic equipment depend on the maximum temperature attained by the components. The temperature rise for $H/L = 0.4$ is shown in Fig. 5 as a function of the Grashof number

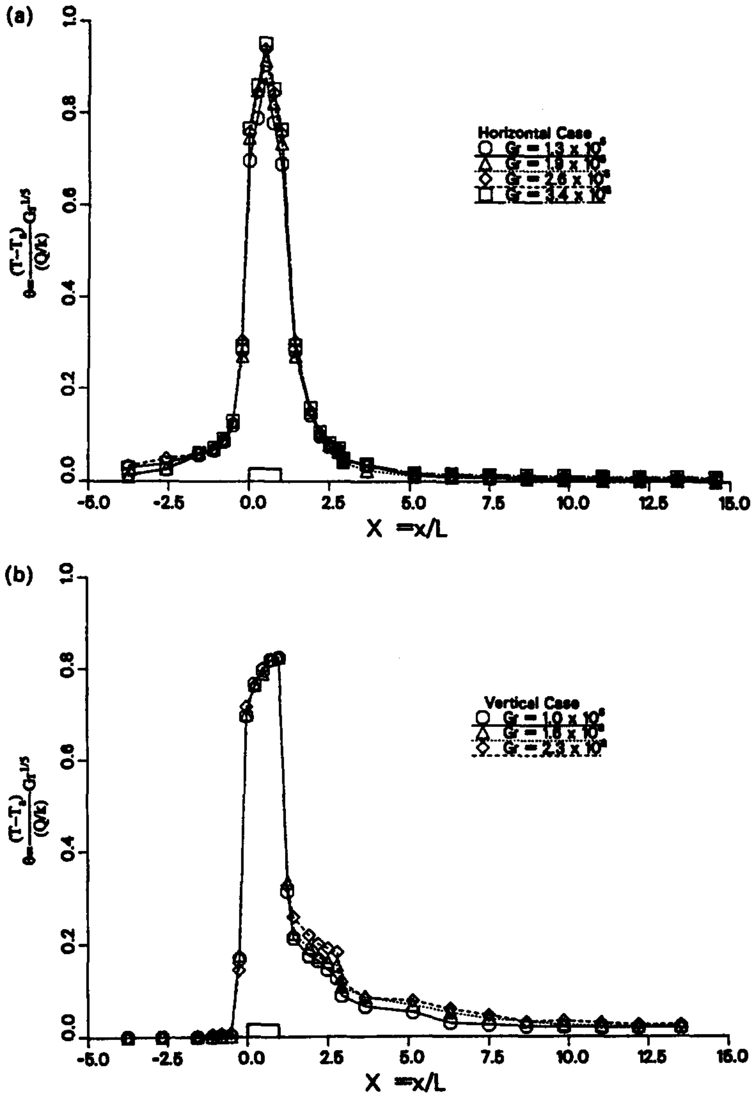

FIG. 4. The downstream variation of the temperature at the plate surface for various values of the Grashof number Gr. (a) Horizontal configuration, $H/L = 0.4$. (b) Vertical configuration, $H/L = 0.4$.

Gr. Again, Gr, in terms of the source length L, represents the buoyancy input at the source and is, therefore, the appropriate parameter for characterizing the thermal transport in the neighborhood of the source. Far downstream, a Grashof number based on the downstream distance from the source is expected to provide a better characterization of the flow [2, 14]. Also shown are the numerical results of Afrid and Zebib [12] for the vertical orientation. A fair agreement is observed between the experimental and numerical results. The temperature rise is found to increase rapidly with an increase in the Grashof number Gr for both the free and wall plume circumstances. However, the temperature rise for a free plume is about 10% larger than that for a wall plume at the same Grashof number. Clearly, this is due to the flow in the vicinity of the source being more vigorous for the vertical orientation owing to the longer side being aligned with the vertical buoyancy force. This leads to a higher heat transfer coefficient and a lower surface temperature in the vertical case. The numerical results show similar trends, but the gradient of ΔT_m versus Gr in the numerical results is somewhat higher than that in the experiments. The difference between the two is found to be about 10% of the measured values over the range of Grashof number considered. This difference can be attributed to the slight difference in the module thickness. The experimental results are for a module thickness $H/L = 0.4$ while the numerical results were for $H/L = 0.5$.

The heat transfer rate can be expressed in terms of the dimensionless local Nusselt number Nu, which is defined as $Nu = hL/k$. This definition again employs the source length L as the characteristic dimension, since the local natural convection flow generated by the source is strongly affected by this dimension [14]. Here, h is the convective heat transfer coefficient and is given by $q_{conv}/(T_s - T_a)$, where q_{conv} is the heat flux

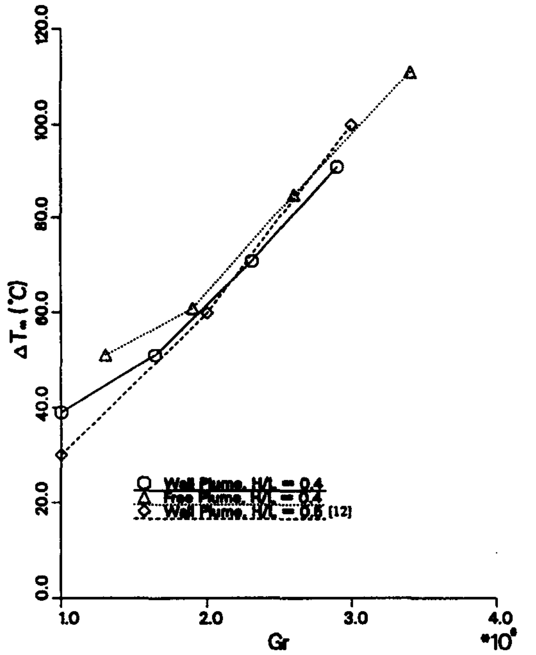

FIG. 5. Surface temperature rise ΔT_m, above the ambient temperature, as a function of the Grashof number Gr for the two orientations.

convected from the module surface to the ambient. This heat flux is obtained from

$$q_{\text{conv}} = q - q_{\text{cond}}. \qquad (3)$$

The heat flux conducted from the module to the plate, q_{cond}, is measured by the heat flux gages mounted at the bottom surface of the heat source, between the stainless steel foil and the bakelite strips. The mean Nusselt number $\overline{Nu}$ for the module surface is defined as

$$\overline{Nu} = \frac{\bar{h}L}{k} \qquad (4a)$$

where

$$\bar{h} = \frac{1}{A}\int_A h\,\mathrm{d}A = \frac{1}{L+2H}\int_{(L+2H)} h(s)\,\mathrm{d}s. \qquad (4b)$$

Here, A is the total surface area of the module exposed to the fluid, i.e. $A = (L+2H)W$, and s is the distance along the module surface, as shown in Fig. 6.

The variation of the local Nusselt number Nu with the location on the heat source module is shown in Fig. 6(a) for the horizontal configuration at $H/L = 0.4$ and $Gr = 2.6 \times 10^6$. The s-coordinate employed for the local Nusselt number distribution is measured around the source and includes both the vertical and horizontal surfaces of the module, as shown. The heat transfer rate is seen to increase along the vertical surfaces (AB and DC) until the top corners (B and C) of the module are reached. A maximum in the heat transfer rate is observed near these corners, followed by a gradual decay along the horizontal surface (BC) to the middle point of the module surface

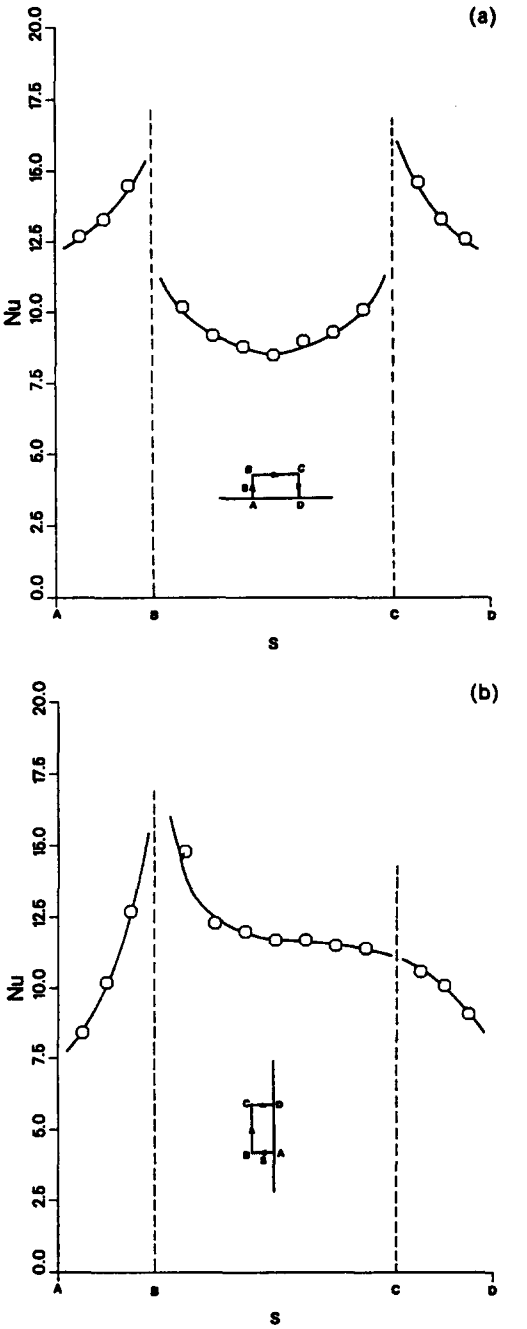

FIG. 6. The variation of the local Nusselt number Nu along the surface of the heat source module. (a) Horizontal configuration, $H/L = 0.4$, $Gr = 2.6 \times 10^6$. (b) Vertical configuration, $H/L = 0.4$, $Gr = 2.3 \times 10^6$.

BC. A minimum in Nu is observed at this point. It is expected that the maximum heat transfer rate would occur at the leading edges (A and D) of the vertical surfaces if the horizontal plate were perfectly insulated and no conduction occurred from the module to the plate. However, in actual practice, the flow does not start at A and D but on the horizontal plate, since the plate does get heated due to the conduction from the module to the plate. This conduction effect is significant. Also, the induced flow due to the top surface BC results in the shift of the local maximum heat transfer rate toward the top corners (B and C). The heat transfer rate from the vertical surfaces (AB and DC) is seen to be higher than that from the horizontal surface (BC) by about 50%. This is clearly due to the buoyancy force being aligned to the heated surface in the former case and being normal to the surface in the latter.

Figure 6(b) shows the variation of the local Nusselt number Nu for the vertical configuration at $H/L = 0.4$. Nu is seen to increase until the left bottom corner (B) of the module is reached. A maximum in Nu is observed near this corner, followed by a gradual decay. Nu decreases rapidly along the module surface CD. The total heat transfer rate from the horizontal surfaces facing upward and downward (AB and CD) is found to be about 70% of the value for the vertical surface (BC). It is also interesting to note that the heat transfer rate, in terms of the Nusselt number, from the horizontal surfaces (AB and CD) for the vertical configuration is about 15% higher than the corresponding value for the horizontal surface (BC) for the horizontal configuration. Also, the value for the vertical surface (BC) for the vertical orientation is seen to be about 10% lower than the value for the vertical surface (AB or BC) for the horizontal orientation. It is also seen that the Nusselt number on face AB increases much more rapidly with distance for the vertical case, as compared to the horizontal orientation. In the vertical orientation, separation arises near point A, resulting in a much smaller value of Nu, as compared to the horizontal case. Similarly, separation arises near point D, lowering the Nusselt number in this region. Thus, the two orientations yield very different heat transfer results because of the difference in the flow fields that arise.

These experimental results for the heat transfer are found to be higher than the calculated values from the empirical expressions given by Steinberg [18] by about 5–10% for the vertical surface and by about 15–40% for the horizontal surface. While the formulae given in ref. [18] are based on an isolated vertical or horizontal surface, the geometry in this experiment has a combination of vertical and horizontal surfaces. In this configuration, each surface influences the buoyancy force due to the other surfaces and the observed trends arise due to the orientation of the module surfaces with respect to gravity.

The experimental data are also compared with the correlation of the laminar boundary solution for a vertical surface with a flush heat source given by Fujii and Fujii [19], as shown in Fig. 7. The solid line in this figure is the correlation equation:

$$Nu_c = 3.48\left(\frac{Pr}{4+9Pr^{0.5}+10Pr}\right)^{1/5}(Gr\,Pr)^{1/5}. \quad (5)$$

Here, Nu_c is defined as the local Nusselt number at the middle point of the heat source module. The present

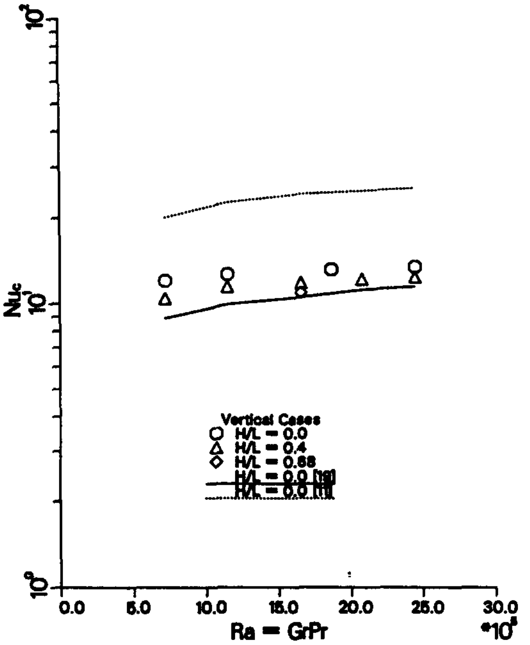

FIG. 7. Comparison of the measured local Nusselt number at the middle point of the vertical surface element (BC) of the module for the vertical configuration with the results from earlier studies.

experimental data for the flush surface are seen to be higher than the predicted values. The experimental data of Park and Bergles [11] for water ($Pr = 5$) and for a source width of 2.0 cm are also shown. These are found to be much higher than the predicted values from equation (5). The data in the present study are found to lie between the experimental results of Park and Bergles [11] and the correlation equation of Fujii and Fujii [19]. This difference between the experimental and numerical results is attributed to the conjugate heat transfer effects due to the conduction in the plate. Also, three-dimensional effects become important at small source widths, as considered in ref. [11]. The leading edge for the flow is no longer the edge of the source but moves vertically downward due to the conduction effects in the plate.

It is also found that, for the vertical configuration, a protruding heat source has a smaller value of Nu_c than a flush heat source. The geometry for the protruding heat source has a combination of vertical and horizontal surfaces. Therefore, the wake induced by one surface interacts with and influences the buoyancy-induced flows at the other surfaces. For instance, the air in the vicinity of the surface AB of the module becomes heated due to the heat input and generates a natural convection wake, which affects the flow of the air near the vertical surface (BC) of the module, leading to a thicker flow region near this surface. Thus, the temperature gradient in the fluid at the middle point of the module surface BC is expected to become lower than that for a flush surface. These results indicate that the wake interaction among the module surfaces is so significant that the heat transfer

CONCLUSIONS

An experimental study on the thermal transport associated with a protruding heat source module mounted on a vertical or a horizontal surface has been carried out. The natural convection flow simulated is taken as two-dimensional due to the experimental arrangement employed. The buoyant flow above a protruding heat source module mounted on a horizontal surface and a vertical surface gives rise to a free plume and a wall plume, respectively. The thermal field generated by the heat source module was studied in detail, particularly in the neighborhood of the module. The effects of the module thickness and of the Grashof number on the thermal transport were also investigated.

It is found from the isotherms obtained that the plate surface is heated by conduction from the module to the plate. The downstream locations of the plate come under the effect of the wake of the wall plume in the vertical circumstance whereas the region far from the source is free from the wake effect of the free plume in the horizontal circumstances. These phenomena strongly affect the surface temperature distribution and different trends in the surface temperature variation in the two cases are obtained. While isotherms in the flow and at the surface for the horizontal configuration are symmetric about the thermal source, those for the vertical configuration are swept downstream. The temperature rise above the ambient is seen to increase rapidly with an increase in the Grashof number for both the free and the wall plume cases. However, the temperature rise for a free plume is found to be larger than that for a wall plume at the same Grashof number.

The results obtained indicate that the interaction of the wakes generated by the surface elements of the heat source module, exposed to the fluid, is very significant and that the natural convection heat transfer characteristics are strongly affected by such an interaction. Thus, the heat transfer rate should not be calculated assuming three isolated surfaces in the design of the thermal system with a protruding heat dissipating module. It is also found that the thermal transport from the protruding heat source module greatly depends on the ratio of the area of the vertical surface element to that of the horizontal surface element of the module exposed to the ambient medium. As H/L approaches 0.5, the vertical and the horizontal surface areas become equal and the total heat transfer for the two configurations is almost identical.

Acknowledgements—The authors acknowledge the support of the National Science Foundation through Grant No. CBT-84-15364 for this work and the help of Dr S. Tewari during the experimental work.

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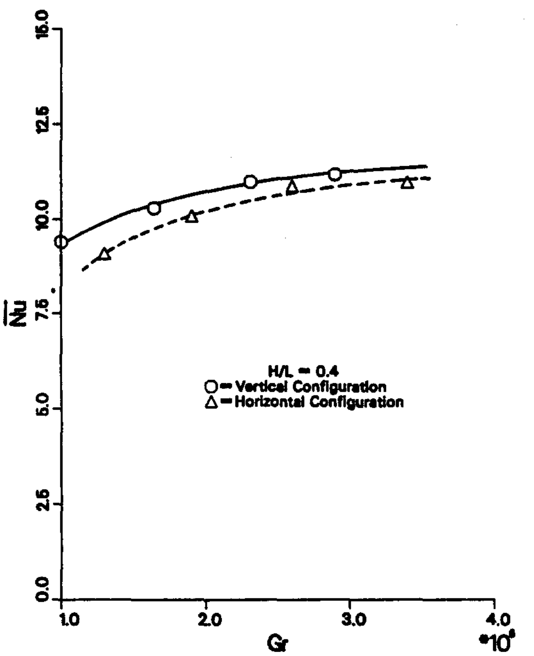

FIG. 8. The variation of the mean Nusselt number $\overline{Nu}$ with Gr at $H/L = 0.4$ for the two flow configurations considered.

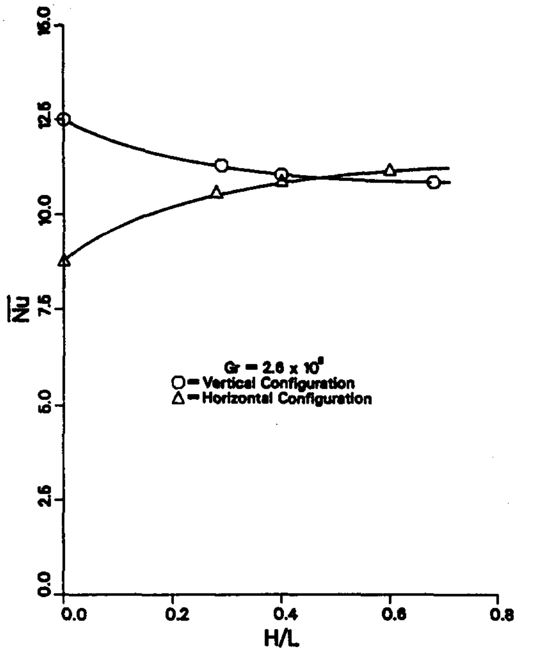

FIG. 9. The variation of the mean Nusselt number $\overline{Nu}$ with H/L at $Gr = 2.6 \times 10^6$ for the two flow configurations considered.

rate should not be calculated assuming an isolated surface in the design of the thermal system which includes a protruding heat dissipating device.

The mean Nusselt number $\overline{Nu}$ for $H/L = 0.4$ is plotted in Fig. 8 against the Grashof number Gr. Correlations were also obtained to indicate the dependence of $\overline{Nu}$ on Gr for various module thicknesses. The correlations for $H/L = 0.4$ were obtained as

$$\overline{Nu} = 0.900Gr^{0.170} \quad \text{for the vertical configuration} \tag{6a}$$

$$\overline{Nu} = 0.431Gr^{0.217} \quad \text{for the horizontal configuration.} \tag{6b}$$

The correlation coefficients, which indicate how well the experimental data are correlated by the above equations, were found to be larger than 0.98. These correlating equations are expected to apply for Gr ranging from about 5×10^5 to 10^7. Similar correlating equations were derived for other H/L values.

It is useful to compare these correlations with the analytical solution for a semi-infinite, uniform heat flux surface. According to the analytical solution, $\overline{Nu}$ is proportional to $Gr^{0.25}$ for the vertical configuration and to $Gr^{0.2}$ for the horizontal configuration [16]. However, for the analysis, a semi-infinite horizontal or vertical surface with a single leading edge is assumed. In experimentation, since finite surfaces are employed, there would be flow over the other edges of the module as well. For the horizontal configuration, two side surfaces act independently as leading edges and the flow proceeds over the surface towards the middle. These flows merge and rise as a buoyant flow. Also, interactions between these individual wakes arising from the various module surface elements will affect the resulting flow. When resulting mean Nusselt numbers are compared with the analytical values for the idealized semi-infinite surfaces, it is found that these effects give rise to a larger $\overline{Nu}$ for the horizontal configuration and a smaller $\overline{Nu}$ for the vertical configuration. It is also noted from this figure that the mean Nusselt number $\overline{Nu}$ for the horizontal configuration is about 10% smaller than that for the vertical configuration at $H/L = 0.4$.

The effect of the module thickness ratio H/L on the mean Nusselt number $\overline{Nu}$ is shown in Fig. 9 for $Gr = 2.6 \times 10^6$. As the module thickness increases, $\overline{Nu}$ decreases for the vertical configuration and increases for the horizontal configuration. These results are expected, since with increasing thickness the horizontal surface area of the module is increased, with a fixed vertical surface area of the module for the vertical configuration. Also, the vertical surface area of the module is increased, with a fixed horizontal surface area for the horizontal configuration, as the module thickness increases. It is found that $\overline{Nu}$ for the vertical configuration is higher than that for the horizontal configuration with $H/L < 0.5$ and the reverse is true with $H/L > 0.5$. As H/L approaches 0.5, the areas of the vertical and the horizontal surfaces of the module become equal and the total heat transfer from the source for the two configurations is almost identical. These results indicate that the thermal transport from a protruding heat source strongly depends on the ratio of the area of the vertical surface to that of the horizontal surface of the module exposed to the fluid, i.e. on the module geometry.

CONVECTION THERMIQUE NATURELLE D'UNE SOURCE EN SAILLIE SUR DES SURFACES HORIZONTALES ET VERTICALES

Résumé—On étudie expérimentalement la convection thermique naturelle d'une source de chaleur d'épaisseur finie montée sur une surface verticale ou horizontale. Ce problème est intéressant dans l'enlèvement de chaleur pour des éléments chauffés de systèmes électroniques. Un panache pariétal, adjacent à la surface, s'élève pour une source chaude montée sur une paroi verticale. Un panache libre est créé par une source de chaleur sur une surface horizontale. On mesure les distributions de température dans l'écoulement de convection naturelle et les caractéristiques de transfert thermique varient fortement avec la puissance de charge et l'épaisseur de la source. Les distributions de température sont différentes pour les panaches pariétaux et libres. On trouve que la conduction le long de la plaque est importante dans le mécanisme de transport. L'interaction entre les sillages générés par les trois surfaces exposées de la source de chaleur a un effet important sur l'écoulement. Ainsi le transfert de chaleur à partir d'une source thermique en saillie ne peut pas être déterminé avec précision en supposant trois surfaces chaudes indépendantes. Les résultats sont comparés avec ceux pour une source d'épaisseur négligeable et on détermine l'effet d'une épaisseur de la source sur l'écoulement et le transport. Le coefficient de transfert thermique pour une source saillante dépend du rapport des aires de surface verticale et horizontale. Plusieurs autres tendances sont observées dans cette étude et elles sont discutées.

NATÜRLICHE KONVEKTION AN EINER HERVORSTEHENDEN WÄRMEQUELLE BEI WAAGERECHTER UND SENKRECHTER GRUNDFLÄCHE

Zusammenfassung—Die natürliche Konvektion an einer Wärmequelle endlicher Dicke, die an einer senkrechten oder waagerechten Grundfläche befestigt ist, wird untersucht. Dieses Problem tritt insbesondere bei der Wärmeabfuhr von Heizelementen in der Fertigungstechnik und in elektronischen Systemen auf. Im Fall einer vertikalen Grundfläche tritt nahe der Oberfläche eine Auftriebsfahne auf. Im Fall einer waagerechten Grundfläche ergibt sich eine freie Auftriebsfahne. In der Strömung und entlang der Oberfläche werden die Temperaturverteilungen gemessen. Die Ergebnisse zeigen, daß die natürliche Konvektion und die damit verbundene Wärmeübertragung stark von der Wärmezufuhr und von der Dicke der Wärmequelle abhängen. Ein Vergleich zwischen der frei aufsteigenden und der wandanliegenden Auftriebsfahne zeigt sehr unterschiedliche Temperaturverteilungen. Während die Isothermen bei freier Auftriebsfahne symmetrisch um die Quelle verteilt sind, werden diejenigen bei wandanliegender Auftriebsfahne in stromabwärtiger Richtung deformiert. Es zeigt sich, daß die oberflächenparallele Wärmeleitung in der Platte von beträchtlicher Bedeutung für den Übertragungsvorgang ist. Bei der wandanliegenden Auftriebsfahne zeigen sich deutliche Nachlaufeffekte, die bei waagerechter Grundfläche fehlen. Die Wechselwirkung zwischen den Nachlaufgebieten der drei verwendeten Oberflächen erweist sich als sehr bedeutsam für die Strömung. Daraus folgt, daß der Wärmeübergang an der betrachteten Anordnung nicht einfach dadurch bestimmt werden kann, daß man drei beheizte einzelne Oberflächen getrennt betrachtet. Die Ergebnisse werden mit denen für eine Quelle von vernachlässigbarer Dicke verglichen, daraus ergibt sich der Einfluß einer endlichen Dicke des Heizelements auf die Konvektionsströmung. Es zeigt sich, daß der Wärmeübergangskoeffizient für die Anordnung vom Verhältnis der senkrechten und der waagerechten Einzelflächen der Anordnung abhängt.

ОСОБЕНОСТИ ЕСТЕСТВЕННОКОНВЕКТИВНОГО ТЕПЛОПЕРЕНОСА ОТ ИСТОЧНИКА ТЕПЛА В ФОРМЕ ВЫСТУПА НА ГОРИЗОНТАЛЬНОЙ И ВЕРТИКАЛЬНОЙ ПОВЕРХНОСТЯХ

Аннотация—Экспериментально исследован естественноконвективный теплоперенос от источника тепла в форме выступа конечной толщины на вертикальной или горизонтальной поверхности. Решение этой задачи представляет особый интерес в связи с отводом тепловой энергии от нагретых элементов в производственных и электронных системах. При наличии выступа на вертикальной поверхности в непосредственной близости от нее возникает пристенная струя. В случае же его расположения на горизонтальной поверхности образуется свободно восходящая тепловая струя. Измерены распределения температур в потоке и вдоль поверхности. Полученные результаты показывают, что особенности естественноконвективного потока и соответствующие характеристики теплопереноса сильно зависят от скорости подвода энергии и толщины источника тепла. В обеих указанных ситуациях температурные распределения носят совершенно различный характер. Если в случае свободной струи изотермы симметричны относительно источника тепла, то в случае пристенной струи они смещены по отношению к нему вниз по потоку. Обнаружено, что существенное влияние на процесс теплопереноса оказывает теплопроводность по длине пластины. При наличии пристенной струи расположенные по потоку участки пластины испытывают влияние спутного следа, тогда как в случае свободной струи он не влияет на удаленную от источника тепла обкасть. Установлено также, что взаимодействие спутных следов, порожденных тремя обтекаемыми поверхностями выступа, существенно изменяет картину течения. Поэтому особенности теплопереноса при наличии выступа с источником тепла нельзя точно определить по трем (нагретым, изолированным) поверхностям. Сравнение полученных результатов с соответствующими данными для источика тепла малой толщины позволило сделать вывод о существенном влиянии толщины выступа на конвективный поток и теплоперенос. Найдено, что коэффициент теплопереноса при наличии выступа с источником тепла зависит от отношения площадей его вертикальной и горизонтальной поверхностей, обтекаемых окружающей средрй. В ходе исследования получено несколько других интересных закономерностей поведения основного потока и процессов теплопереноса.