INT. J. NUM. METH. HEAT FLUID FLOW, VOL. 5, 839–851 (1995)

COMBINED FORCED AND FREE LAMINAR CONVECTIVE HEAT TRANSFER FROM A VERTICAL PLATE WITH COUPLING OF DISCONTINUOUS SURFACE HEATING

KOKI KISHINAMI, HAKARU SAITO AND JUN SUZUKI

Department of Mechanical Engineering, Muroran Institute of Technology, 27-1 Mizumoto-cho, Muroran, 050, Japan

ABSTRACT

Combined free and forced laminar air convective heat transfer from a vertical composite plate with isolated discontinuous surface heating elements has been studied numerically and experimentally. The problem has been simplified by neglecting heat conduction in unheated elements of the plate to accomplish a better understanding of the complicated combined/complicated convection problem. In this study, it is most important in explaining the heat transfer behaviour to clarify the interactions between buoyancy and inertia forces in the convective field and also the coupling effects of unheated elements upon the combined flow fields. Therefore, the temperature distributions of the wall surface and local Nusselt number, obtained by numerical calculations and experiments, have been discussed based on the various parameters associated with the present convection problem, i.e., Grashof number Gr_L, Reynolds number Re_L, geometry factor D/L and stage number N. Heat transfer characteristics $Nu_t/Re_L^{1/2}$ of this combined and coupled convection of air are presented as a function of a generalized coupling dimensionless number Gr_L/Re_L^2, and stage number N for certain values of the geometry factor of D/L.

KEY WORDS Laminar flow Convection Heat transfer Discontinuous heating

NOMENCLATURE

D	length of unheated element
L	length of heated element as characteristic length
Gr_L	Grashof number based on the temperature difference between the heated element and surroundings, $(\theta_w - \theta_\infty)$, and L
Re_L	Reynolds number based on the free velocity U_∞ and L
h	heat transfer coefficient based on $(\theta_w - \theta_\infty)$
N	stage number from the leading edge
Nu	local Nusselt number defined by (5)
Nu_t	mean Nusselt number defined by (6)
Pr	Prandtl number
Θ	dimensionless temperature defined by (1)
U, V	dimensionless vertical and transverse velocities defined by (1)
U_∞	free stream velocity
X, Y	dimensionless vertical and transverse coordinates

INTRODUCTION

Combined/coupled convective heat transfer has been becoming important in recent years, both in academic and in practical fields. A problem considered here has become the subject of considerable interest in several technological applications, particularly in electronic semi-conductor devices, any industrial manufacturing systems and heat transfer elements in heat exchangers appearing in engineering. The convection problem considered in this paper should be treated exactly as a combined problem of convection due to buoyancy and inertia forces,

0961–5539/95/090839–13$2.00
© 1995 Pineridge Press Ltd

Received March 1994
Revised September 1994

© Emerald Backfiles 2007

and also coupled with thermal conduction in unheated elements though it is very difficult to solve the combined/coupled free and forced convection because of the many parameters including.

Problems with flow situations similar to those considered here have been studied by many investigators[1,2]. For example, an effect of buoyancy on forced convection flow and heat transfer is reported by Mori[2], and combined laminar convection in horizontal rectangular channels by Cheng *et al.*[3]. Most of the papers presented up to now have treated heat transfer by considering the heated surface to be isothermal and have paid little attention to the case of a composite plate with discontinuous surface heating in which peculiar heat transfer may appear[4,5].

In this study, combined forced and free laminar convective heat transfer of air from a composite plate with isolated discontinuous surface heating has been studied, where buoyancy and inertia forces are in the same direction and the induced convective flow is affected by the existence of unheated elements. Numerical calculations are carried out for laminar, forced and free mixed convection air flow along a vertical composite wall with isolated heating elements. Numerical results such as Nusselt number and velocity/temperature distributions in boundary layers are presented for the purpose of illustrating the effects of these parameters on the convective heat transfer. Experimental corroboration is provided by measuring the temperatures of the plate surface and boundary layers with ϕ 0.1 mm C–C thermocouple under the conditions of temperature difference between the heating element and the surroundings $\Delta\ \theta = 15 \sim 45$°C, free stream velocity $U_\infty = 0.2$, 0.5 and 1.0 m/s, geometry factor $D/L = 0.5$, 1.0 and 1.3 for the stage number $N = 4$, on the basis of the characteristic length of $L = 100$ mm. Heat transfer behaviour predicted by the numerical calculations and observed in the experiments are discussed based on the dimensionless number for coupling forced and free convection Gr_L/Re_L^2, stage number N, and the geometry factor D/L.

The results are summarized by proposing the generalized expressions of $Nu_l/Re_L^{1/2}$ for predicting the combined and coupled convective heat transfer by using these parameters of Gr_L/Re_L^2, D/L and N.

PHYSICAL MODEL AND ANALYSIS

The schematic diagram for the physical situation of forced-free combined air convection under consideration is shown in *Figure 1* where convective heat transfer and thermal diffusion are taking place on the surface of the heating and adiabatic elements, respectively. The composite vertical plate, located in the opposite direction to the gravity, is set up in the parallel flow of the free stream velocity U_∞, in which x-axis originated from the leading edge and y-axis measured from the surface of the plate are taken as the longitudinal and transverse length coordinates. Isolated heating elements with the length of L mounted on an adiabatic surface are separately located by a prescribed distance D, i.e., unheated element length. The vertical composite plate is composed of N stages from the leading edge, counting a pair of combinations with one heated element and one adiabatic element as one stage.

The combined/coupled convection heat transfer is treated as a steady, laminar boundary layer flow developed by the cooperation of the buoyancy and inertia forces over the vertical plate. In treating the multi-stage heated elements, the following assumptions are made; the heating elements with the length of L are kept at the identical isothermal heating temperature θ_w, and unheated elements with the length of D are regarded to be adiabatic.

Governing equation

The following dimensionless variables, commonly used for the combined free and forced convection on the multi-stage heating plate[5], can be defined, by taking the heating element

© Emerald Backfiles 2007

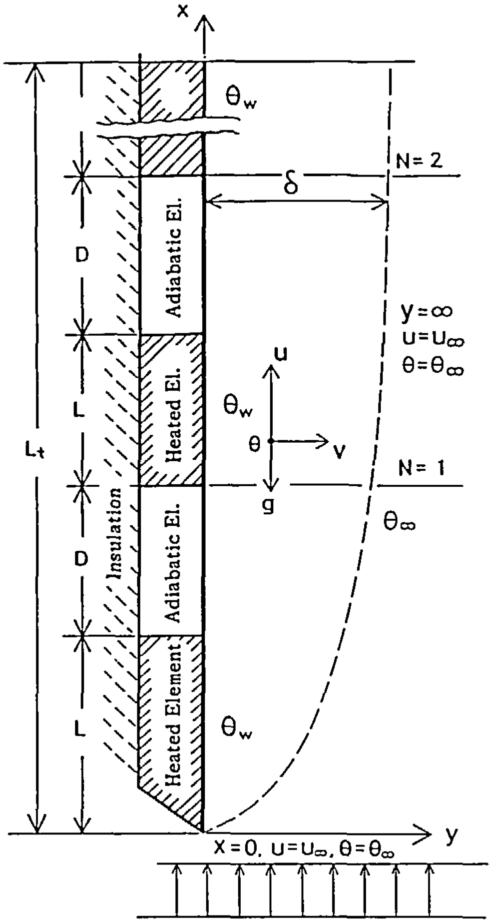


Figure 1 Physical model

length L and free stream velocity U_∞ as the characteristic length and velocity,

$$X=x/L, \quad Y=y/L, \quad U=u/U_\infty, \quad V=v/U_\infty, \quad \Theta=(\theta-\theta_\infty)/(\theta_w-\theta_\infty)$$

$$Re_L=U_\infty L/\nu, \quad Gr_L=g\beta L^3(\theta_w-\theta_\infty)/\nu^2, \quad Pr=\nu/\alpha \tag{1}$$

The governing equations for the laminar boundary layer are given in the following dimensionless form by applying the variables of (1),

Continuity equation

$$\frac{\partial U}{\partial X}+\frac{\partial V}{\partial Y}=0 \tag{2}$$

X-momentum equation

$$U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=\frac{Gr_L}{Re_L^2}\cdot\Theta+\frac{1}{Re_L}\cdot\frac{\partial^2 U}{\partial Y^2} \tag{3}$$

Energy equation

$$U\frac{\partial \Theta}{\partial X}+V\frac{\partial \Theta}{\partial Y}=\frac{1}{Pr\cdot Re_L}\frac{\partial^2 \Theta}{\partial Y^2} \tag{4}$$

where the standard boundary layer assumptions for the forced convection have been employed.

© Emerald Backfiles 2007

Based on the difference in temperature between the heated surface and surroundings $(\theta_w-\theta_\infty)$, the heat transfer coefficient on the surface, regardless of heating or unheating, can be expressed as follows, by using the dimensionless variables of (1),

$$Nu=\frac{h_x L}{\lambda_f}=-\frac{\partial\Theta}{\partial Y}\bigg|_0 \tag{5}$$

In the present problem, it is most important to estimate the total heat delivery from the entire system. Then, the mean/accumulated Nusselt number, Nu_t, on every stage are defined as follows,

$$Nu_t=\frac{1}{L}\int_0^x Nu\,\mathrm{d}x=\int_0^X Nu\,\mathrm{d}X \tag{6}$$

where the mean/accumulated Nusselt number, Nu_t, on the N stages is interpreted as an inegration of the local Nusselt number Nu with respect to the nondimensional length of X in which the upper integration limit implies the distance from the leading edge to the corresponding stage to be equivalent to $x=N(L+D)$ or $X=N(L+D)/L$.

Only the boundary conditions on the surfaces of heating and unheated elements are expressed as follows since the rest parts of the boundary conditions are the same as in the case of forced convection,

$$\text{For the heating elements:} \quad Y=0; \quad \Theta=1 \tag{7}$$

$$\text{For the adiabatic elements:} \quad Y=0; \quad \frac{\partial\Theta}{\partial Y}=0 \tag{8}$$

Outline of numerical solution technique

In the numerical calculation for a multi-stage combined/coupled convection flow, the governing equations (2), (3) and (4) for convection in the boundary layer are written in the form of finite difference, by taking an up-wind difference for a non-linear terms and a central difference form for other derivatives, based on the control volume technique. Hence, non-uniform node spacings are used in both X- and Y-directions in order to guarantee the accuracy of the numerical results, especially fine node spacings in the regions near the surface of the wall and the leading edge. The difference forms of the governing equations are solved numerically by the iterative technique of the SOR method. The numerical results obtained are estimated to involve the relative error less than 3% for Nusselt number on the 1st stage[5,6].

EXPERIMENTAL APPARATUS AND PROCEDURE

Figure 2 shows an outline of the experimental apparatus with a composite vertical plate set up in the opposite direction to the gravity. The heat transfer composite vertical plate was sharpened at the lower leading edge and comprised $N=4$ stages with isothermal heating elements in length $L=100$ mm and adiabatic elements in length $D=50$ or 100 or 130 mm. Heating elements were made of aluminum plate of 3 mm thick and warmed from the rear by an electrical heater (Nichrome wire of ϕ 0.26 mm) set in the wooden plate with the thickness of 10 mm, and controlled at a heating temperature between 35~65°C within the accuracy of 0.2°C. Unheated elements were made from a balsa plate with thermal conductivity $\lambda_b=0.55$ W/(mK) assumed to be nearly adiabatic. Thin aluminum foil with emissivity of 0.09 and 15 μm in thickness was bonded tightly to the surface of the unheated balsa elements in order to minimize the thermal radiation loss. A low velocity wind tunnel ($U_\infty=0.05\sim4.0$ m/s), with a 500 mm dia. propeller driven by a 150 W inverter controlled motor, was installed at the bottom of the composite vertical plate to regulate the free stream velocity U_∞ from 0.2 m/s to 1.0 m/s through the 2 stages honeycomb with dimension 400 x 420 x 30 mm (hole dia. ϕ 3 mm). The side walls, made of plywood with

© Emerald Backfiles 2007

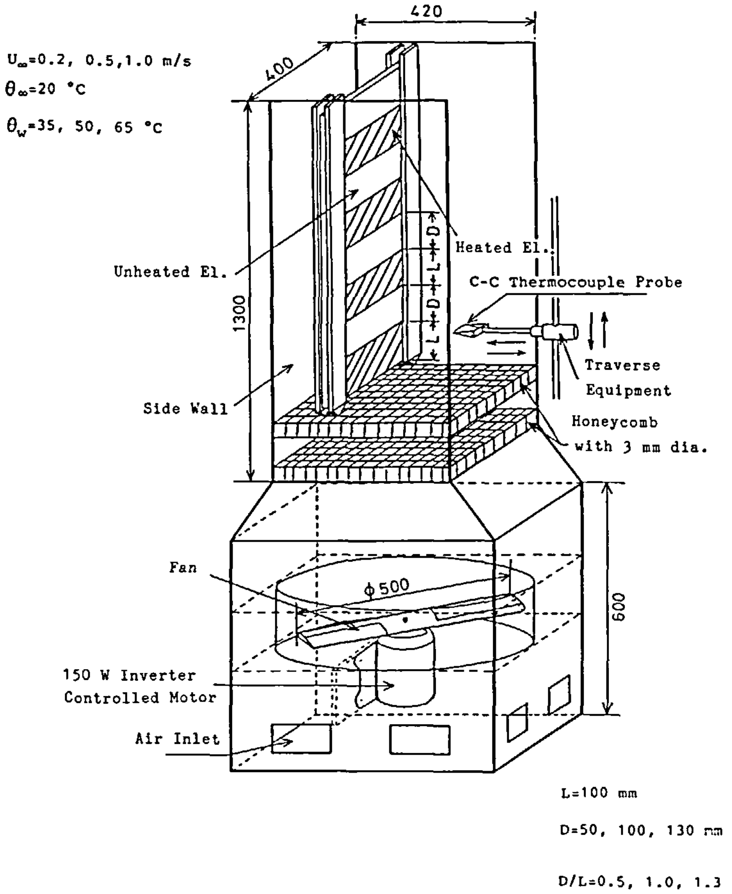


Figure 2 Experimental apparatus

1300 mm in height and 420 mm in width, were installed on the both sides of the convection plate to prevent external disturbance. The heated and adiabatic elements could be replaced to change the condition of the number of the stage N and the geometry factor of D/L. The temperature profiles in the boundary layer and surface temperature distribution on the plate were measured by a ϕ 0.1 mm C–C thermocouple probe which was traversed precisely by a remote-controller, and the free stream velocity by thermistor anemometer. Experimental conditions were as follows: the difference in temperatures between the heated surface and the surroundings $\Delta\theta = 15$, 30 and 45°C, the free stream velocity $U_{\infty} = 0.2$, 0.5 and 1.0 m/s and the geometry factor $D/L = 0.5$, 1.0 and 1.3. A steady state condition was attained at about 1 hour after the stating of the run. Then, temperatures of the boundary layer were measured at $52 \sim 64$ points along the surface by remote-controlling C–C thermocouple probe. Local mean temperatures at each point were measured at five/six points distant 0.05, 0.1, 0.15, 0.2 and 0.3 cm from the surface. The data obtained was directly linked to a personal computer to estimate the local Nusselt number by means of the 'least squares' procedure, including 10% experimental error, and to draft a figure in real time.

© Emerald Backfiles 2007

RESULT AND DISCUSSION

The basic behaviour of combined free and forced convective heat transfer in this model is considerably different from that of an ordinary single isothermal plate because of the existence of the intermittent adiabatic elements.

Vectorial dimensional and numerical analyses

By performing a vectorial dimensional analysis for the mixed/coupled forced-free convection heat transfer, the following expressions are obtained for the wall temperature and heat transfer coefficient when the heat conduction in unheated element is neglected through its length D is considered.

$$Nu/Re_L^{1/2}=f_1(X, Pr, Gr_L/Re_L^2, D/L) \tag{9a}$$

$$\Theta_w=f_2(X, Pr, Gr_L/Re_L^2, D/L) \tag{9b}$$

where $X=x/L=f(N)$.

The main parameter (Gr_L/Re_L^2) explaining this combined/coupled problem can be provided by other aspect of the dimensionless transformation of the governing equations applying new dimensionless variables as follows,

$$X=\frac{x}{L} \qquad Y^*=\frac{y}{L}Re_L^{1/2} \qquad U=\frac{u}{U_\infty} \qquad V^*=\frac{v}{U_\infty}Re_L^{1/2} \tag{10}$$

where the above variables are based on the forced convection boundary layer assumption while the dimensionless temperature Θ, Reynolds number Re_L, Grashof number Gr_L are the same forms defined by (1).

The dimensionless governing equations and local Nusselt number are given as follows,

$$\frac{\partial U}{\partial X}+\frac{\partial V^*}{\partial Y^*}=0 \tag{11}$$

$$U\frac{\partial U}{\partial X}+V^*\frac{\partial U}{\partial Y^*}=\frac{Gr_L}{Re_L^2}\cdot\Theta+\frac{\partial^2 U}{\partial Y^{*2}} \tag{12}$$

$$U\frac{\partial \Theta}{\partial X}+V^*\frac{\partial \Theta}{\partial Y^*}=\frac{1}{Pr}\frac{\partial^2\Theta}{\partial Y^{*2}} \tag{13}$$

$$\frac{Nu_L}{Re_L^{1/2}}=-\frac{\partial\Theta}{\partial Y^*}\bigg|_{y=0}=f\left(X, Pr, \frac{Gr_L}{Re_L^2}\right) \tag{14}$$

where the heat conduction in X-direction in energy equation is neglected because of little effect on the velocity and temperature fields.

These relations, (11), (12), (13) and (14), indicate that the results obtained by numerical calculation should be generalized solution for the every parameter $Gr_L/Re_L^{1/2}$, regardless of the values of Gr_L and Re_L. Consequently, it is concluded that the obtained results of U, Θ_w and $Nu/Re_L^{1/2}$ should be treated as the coupling parameter Gr_L/Re_L^2, geometry factor D/L and stage number N from both analyses.

Surface temperature and local Nusselt number

The distributions of the surface temperature on the plate with $N=4$ stages predicted by the numerical calculations are shown in *Figure 3*, for the geometry factor $D/L=0.5$, 1.0 and 1.3, indicated by a broken line, a solid line, and a chained line, respectively. Also in the figure indicated are the free stream velocity $U_\infty=0.2$ m/s, and temperature difference $\Delta\theta=30$°C, i.e.,

© Emerald Backfiles 2007

D/L	EXP.	F.D.S.	Gr_L	Re_L	U_∞
0.5	□	- - - - - -	3.1281×10^6	1214.8	0.2 m/s
1.0	○	———	3.1281×10^6	1214.8	0.2 m/s
1.3	●	— — —	3.1281×10^6	1214.8	0.2 m/s

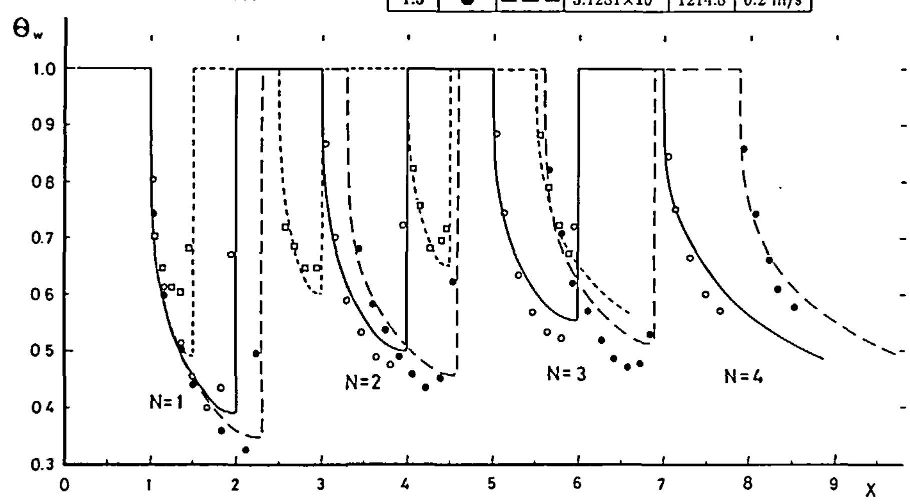


Figure 3 Surface temperature distribution on plate for $D/L=0.5$, 1.0, 1.3

EXP.	F.D.S.	Gr_L	Re_L	U_∞
	———	3.1281×10^6	5799.1	1.0 m/s
	—·—	3.1281×10^6	2899.6	0.5 m/s
●	—··—	3.1281×10^6	1214.8	0.2 m/s

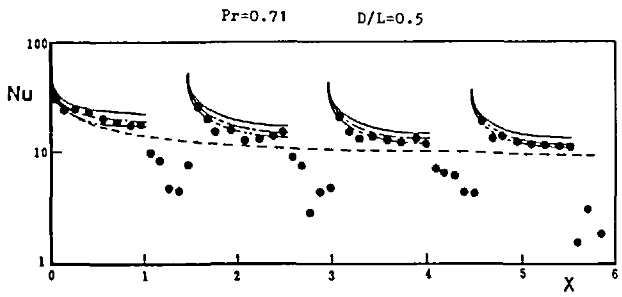


Figure 4 Local Nusselt number for $D/L=0.5$

$Gr_L/Re_L^2=2.325$. The numerical results are compared with the experimental data of corresponding conditions expressed as symbols of □, ○ and ●, respectively. The surface temperature distributions on the unheated elements are found to fall close to the lower contact point and then gradually decrease downstream. Also the surface temperature of the unheated elements increases in accordance with the stage number N. From the figure, the predicted numerical results are found to be in good agreement with the corresponding experimental data except the contact points region. Paying attention to the effect of the geometry factor D/L on the distribution, the surface temperature for small value of $D/L=0.5$ is found to be considerably higher than that for a large value of D/L while the distributions are very similar to each other. The small effect of heat conduction in unheated elements (balsa wood) are also recognized on their distribution near the contact region between the heated and unheated elements in the range of small D/L.

Figure 4 shows the local Nusselt number on every heated element under the same condition as above for $D/L=0.5$, with slightly changed free stream velocity $U_\infty=0.2$, 0.5 and 1.0 m/s for

© Emerald Backfiles 2007

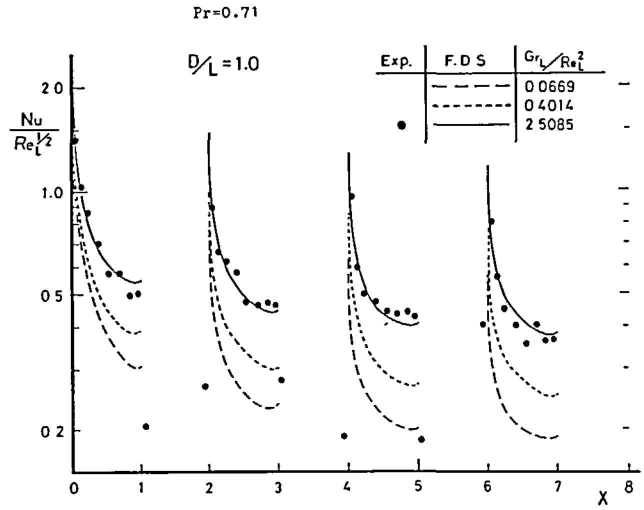


Figure 5 Local heat transfer characteristics $Nu/Re_L^{1/2}$ for $D/L=1.0$

the constant $Gr_L=3.125\times10^6$ ($\Delta\theta=30°C$). *Figure 5* indicates the relation of the local heat transfer characteristics $Nu/Re_L^{1/2}$ and X with Gr_L/Re_L^2 as the parameter for $D/L=1.0$ under the same condition of Gr_L and Re_L as above. In these figures, the numerical corresponding results are expressed by a solid line, a dotted chained line and a double-dotted chained line in *Figure 4*, and a broken line, a dotted line and a solid line in *Figure 5*, comparing with the experimental data for the $U_\infty=0.2$ m/s expressed by a symbol ●, respectively. According to the results in these figures, Nusselt number on the first heated element changes in the same manner as an ordinary isothermal single plate, and Nusselt numbers on the following heated elements abruptly increases at every of the lower contact point due to the edge effect while the Nusselt number becomes larger with the increase in the free stream velocity U_∞. The effect of the free stream velocity changed from $U_\infty=0.2$ m/s to 1.0 m/s at constant Gr_L is found to take place to 15~30% promotion of Nu on the $N=1$ stage and 10~20% on the following stages. The numerical results of Nusselt number are recognized to be consistent with the corresponding experimental data in *Figure 4* ($D/L=0.5$) and *Figure 5* ($D/L=1.0$), but are not always in agreement with the data in the region of unheated element due to the heat conduction effect from the upper and lower contact regions, especially in the range of small D/L. From the *Figure 5*, small range of Gr_L/Re_L^2 such as 0.067 is interpreted to be the small effect of the buoyancy, i.e., pure forced convection flow. So, it is concluded that the buoyancy effect on Nu at constant U_∞ plays an important role of heat transfer promotion in the range of $0.07<Gr_L/Re_L^2<2.5$.

The heat transfer enhancement due to the edge effect is clearly recognized occurring on every heated element and trending toward weakening its effect in the region of small D/L. For reference, a pure natural convection heat transfer for a single isothermal plate at the same condition are compared with the present results expressed by a broken line.

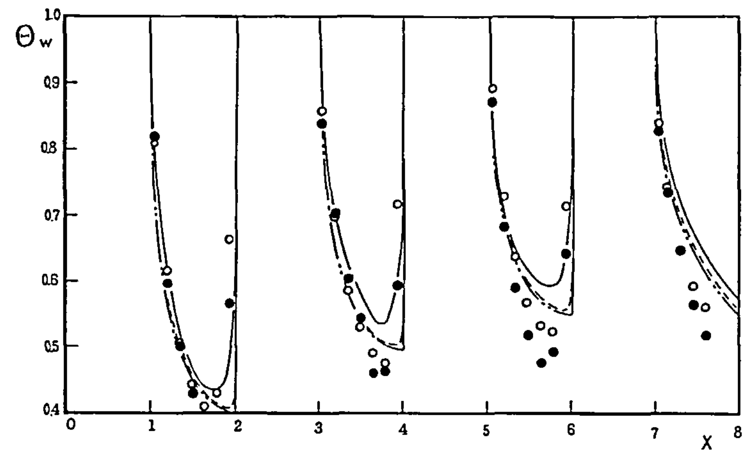
Further consideration for various thermal conditions

Figure 6 shows the temperature distributions along the surface of the wall and local Nusselt number Nu estimated through the numerical calculations for various parameters and experiments under the specified conditions of $\Delta\theta=30°C$, $U_\infty=0.2$ m/s, i.e., the coupling parameter $Gr_L/Re_L^2=2.325$ for the $D/L=1.0$. The results illustrated in the figure includes the some cases taking the combination effect of the buoyancy on the inertia forces and the existence of the

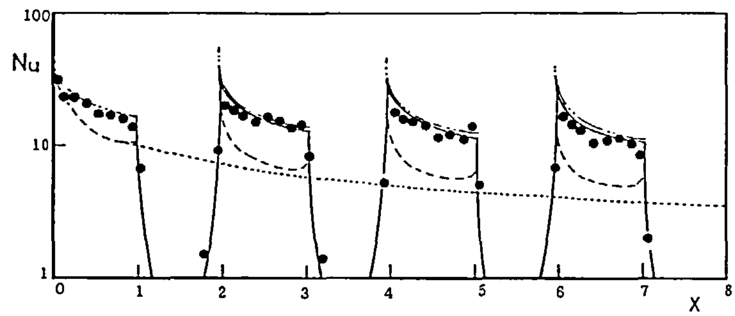
© Emerald Backfiles 2007

Pr=0.71 D/L=1.0

EXP.	F.D.S.	Gr_L	Re_L	U_∞	λ_s/λ_f
	— — —	0.0	1214.8	0.2 m/sec.	0.0
	—··—	3.1281×10^6	1214.8	0.2 m/s	0.0
●○	———	3.1281×10^6	1214.8	0.2 m/s	2.03



(1) Distribution of Wall Surface Temperature



(2) Local Nusselt Number

Figure 6 Results for various thermal conditions, 1) distribution of wall surface temperature, 2) local Nusselt number

unheated elements into consideration; one is the result obtained by considering the effect of heat conduction in unheated balsa wood elements (solid line), the others are by neglecting the thermal conduction in the unheated elements as adiabatic (double-dotted line), and the others by only the inertia forces, i.e., pure forced convection (broken line). It seems that there is little difference in both numerical results indicated by a solid line and a double-dotted chained line. This is due to the insulation effect of the balsa wood and the results indicated by the solid line are found to be in better agreement with the experimental data than the results by the double-dotted chained line, especially in the vicinity of the contact region of the unheated elements. Therfore, it can be said that the simplification employed in this calculation (the unheated elements as

© Emerald Backfiles 2007

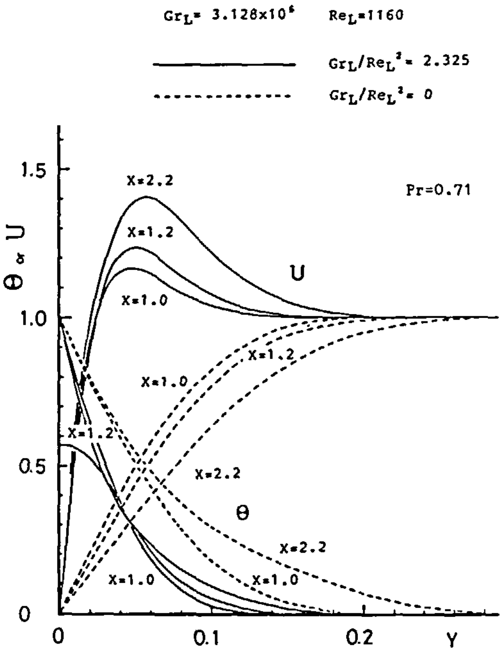


Figure 7 Behaviour of boundary layer affected by buoyancy for $Gr_L/Re_L^2 = 2.325$

adiabatic) is reasonably applicable to the present problem for the case of large D/L. In the figure, the broken line corresponds to the case of pure forced convection by neglecting the buoyancy term in the momentum equation. Accordingly, the local Nusselt number indicated by the broken line is found to be reduced by about $2/3 \sim 1/2$ in its value in accordance with the stage number N, comparing with the solid line for the combined free-forced convection. Contrary to this, little difference in the surface temperatures can be seen between both results. The fine dotted line, i.e., local Nusselt number for the case of the pure forced convection on a continuous isothermal heating plate, indicates that Nu is considerably lower than the solid line for the combined convection. Hence, it is expected that the strong effects of the buoyancy to the boundary layer exists in the vicinity of the wall surface.

Figures 7 and *8* show the temperature and velocity profiles of the boundary layers in the transition region from $N=1$ to $N=2$ stages for the conditions of $Gr_L/Re_L^2 = 2.325$ ($\Delta\theta = 30°C$, $U_\infty = 0.2$ m/s) and $Gr_L/Re_L^2 = 0.372$ ($\Delta\theta = 30°C$, $U_\infty = 0.5$ m/s). Also, the solid line and dotted line correspond to the cases of the combined free-forced convection (buoyancy and inertia forces being considered) and the pure forced convection (buoyancy force being neglected), respectively. The results represented by the solid line for the $Gr_L/Re_L^2 = 2.325$ reveal that the strong effects of the buoyancy on the velocity and temperature distributions exist in the vicinity of the wall surface and become more evident with an increase in the X-location, in comparison with the case of the dotted line. Subsequently, the local Nusselt number of the combined free-forced convection is promoted by $3 \sim 5$ times in comparison with that of the pure forced convection as shown in *Figure 6*. On the other, for the case of $Gr_L/Re_L^2 = 0.372$, the effects of the buoyancy on the velocity and temperature boundary layers are found to be not so strong as that of $Gr_L/Re_L^2 = 2.325$, even though the velocity and temperature distributions near the wall surface are considerably affected by the buoyancy.

An observation of the flow patterns and the measurement of the temperature reveal that the stable laminar flow covers the entire surface of the plate. It is also recognized that the combined

© Emerald Backfiles 2007

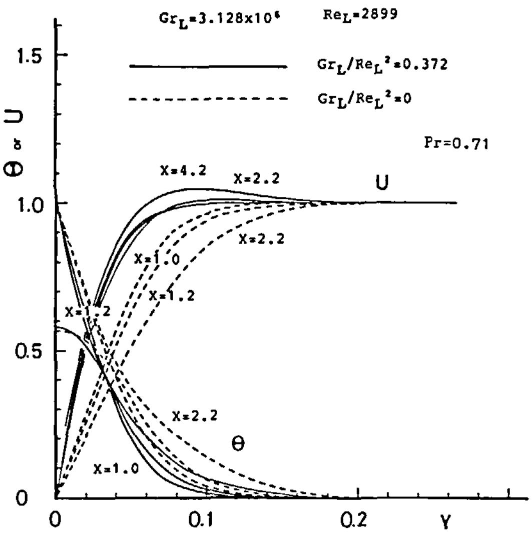


Figure 8 Behaviour of boundary layer affected by buoyancy for $Gr_L/Re_L^2=0.372$

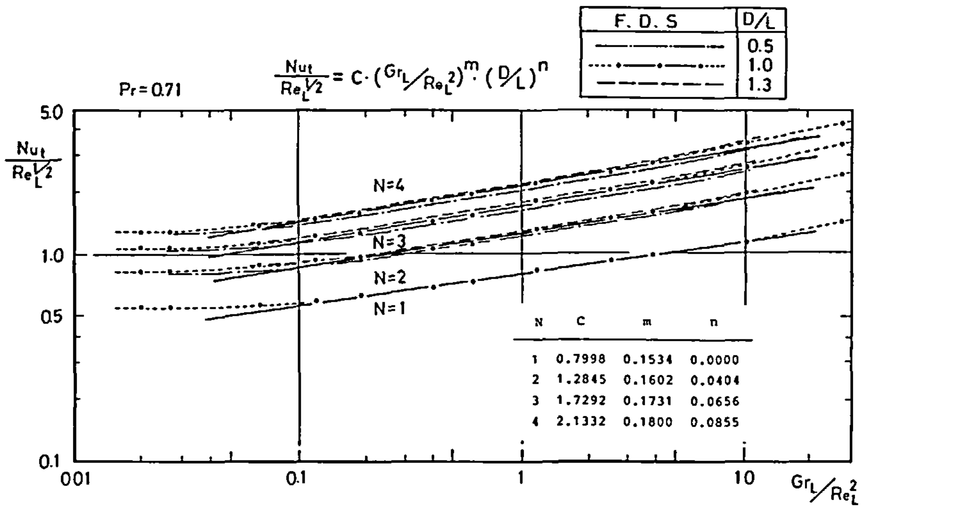


Figure 9 Mean heat transfer characteristics $Nu_t/Re_L^{1/2}$ versus generalized coupling number Gr_L/Re_L^2 with N and D/L as parameters

free and forced convection flow considered here is more stable than the pure natural convection flow.

Heat transfer characteristics, $Nu_t/Re_L^{1/2}$

Figure 9 shows the relationship between the heat transfer characteristics $Nu_t/Re_L^{1/2}$ at every heating element ($N=1\sim4$) and the generalized coupling number Gr_L/Re_L^2 of the mixed convection based on the results of dimensional and numerical analyses for the cases of $D/L=0.5$, 1.0 and

© Emerald Backfiles 2007

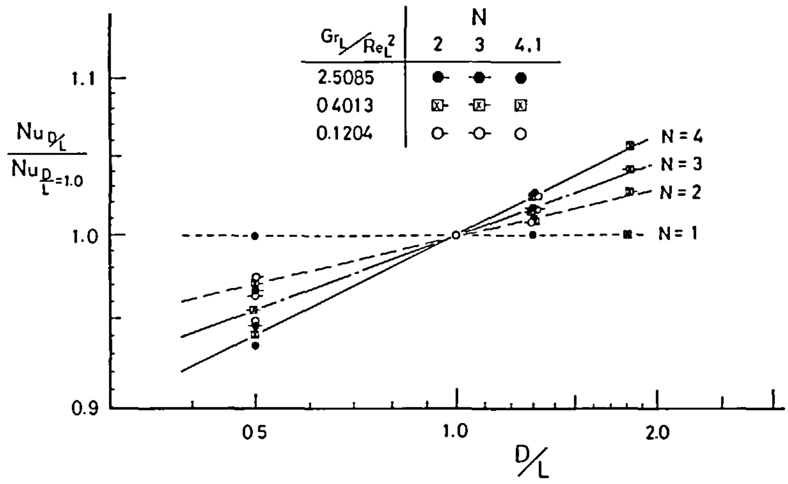


Figure 10 Dependence of D/L on mean heat transfer Nu_t expressed in values relative to a standard of case for $D/L = 1.0$

1.3. It can be concluded that the heat transfer characteristics $Nu_t/Re_L^{1/2}$ are completely dependent on the generalized coupling number Gr_L/Re_L^2 and the geometry factor D/L in the following relation:

$$Nu_t/Re_L^{1/2} = C(Gr_L/Re_L^2)^m(D/L)^n \tag{15}$$

where C is a constant and an exponent m is the slope of the relation at every stage. For convenience, the constant C and exponents m,n of the every stage N determined by the least squares method for multi variables are tabulated in the figure for the range of $0.07 < Gr_L/Re_L^2 < 7.0$. (15) is interpreted as the generalized relation of the heat transfer characteristics of the present combined/coupled free and forced convection and within the 3% error in the prescribed range.

Taking the geometry factor's contribution to the heat transfer characteristics into consideration, *Figure 10* precisely illustrates the dependence of D/L on the average heat transfer Nu_t expressed in relative rate to a standard of the case for $D/L = 1.0$. The heat transfer promotion due to the existence of the unheated elements is recognized to increase in its value with the increasing D/L and stage number N although the increasing rate is to be slight.

Decision of mixed convection range by coupling parameter

From the result of *Figure 9*, it is predicted that the mixed convection area is dominated by the coupling parameter range, $0.07 < Gr_L/Re_L^2 < 7.0$. Out of the range, i.e., for the range of $Gr_L/Re_L^2 < 0.07$, the exponent m asymptotically approaches zero, i.e., Nu_t is proportional to $Re_L^{1/2}$. On the other, for the range of $Gr_L/Re_L^2 > 7$, the constant m gradually approaches 0.25, i.e., Nu_t is proportional to $Gr_L^{1/4}$. Consequently, forced convection heat transfer predominates for the range of < 0.07 and natural convection heat transfer for the range of $Gr_L/Re_L^2 > 7$, even though the transition range of Gr_L/Re_L^2 seems to become narrow in accordance with the stage number N as shown dotted line in *Figure 9*.

CONCLUSION

In this paper, a combined and coupled convective heat transfer from the composite plate with an isolated heating surface was studied by numerical analysis and experiments for the wide range of thermal conditions. It is clear that the heat transfer characteristics $Nu_t/Re_L^{1/2}$ at every stage

© Emerald Backfiles 2007

are strongly dependent on the existence of the unheated section, i.e., D/L, and on the generalized coupling nondimensional number, Gr_L/Re_L^2.

REFERENCES

1 Mabuti, I. Heat transfer from a heated vertical plate in flow field of low Reynolds number, *Trans. JSME*, **27**, No. 180, 1299–1305 (1961)

2 Mori, Y. Buoyancy effects in forced laminar convection flow over a horizontal flat plate, *Trans. ASME, J. of Heat Transf.* **83**, 479–482 (1961)

3 Cheng, K. C. Numerical solution for combined free and forced laminar convection in horizontal rectangular channels, *Trans. of ASME, J. of Heat Transf.* **91**, 59–66 (1969)

4 Kishinami, K. A fundamental investigation of laminar natural convective heat transfer from a vertical plate with discontinuous surface heating, *29th Series of Archival Publishings of the ICHMT*, 139–153 (1991)

5 Kishinami, K. Natural convective heat transfer on a vertical plate with discontinuous surface-heating, *1987 ASME-JSME Thermal Eng. Joint Conf.*, 61–68 (1987)

6 Roache, Patrick J. *Computational Fluid Dynamics*, Hermosa Publishers Inc (1976)

© Emerald Backfiles 2007