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Effect of surface geometry and orientation on laminar natural convection heat transfer from a vertical flat plate with transverse roughness elements

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Abstract—An interferometric technique was used to determine local heat transfer coefficients for surfaces with repeated ribs and steps. The effects of parameters such as protuberance height-to-spacing ratio, conductivity of ribs, and angle of inclination were studied. It was found that heat transfer enhancement, relative to a plain vertical surface of equal projected area, was possible in laminar natural convection using transverse roughness elements of proper size and shape. In general, the stepped surfaces helped improve the heat transfer. The maximum increase in average heat transfer coefficient was 23.2% with a step pitch-to-height ratio of 16. The study indicated the presence of an optimum step pitch-to-height ratio. All of the ribbed surfaces resulted in degraded heat transfer performance.

INTRODUCTION

NATURAL CONVECTION from vertical surfaces with large scale surface roughness elements is encountered in several technological applications. Of particular interest is the dissipation of heat from electronic circuits, where component performance and reliability are strongly dependent on operating temperature. Natural convection represents an inherently reliable cooling process. Further, this mode of heat transfer is often designed as a back-up in the event of the failure, due to fan breakdown, of a forced convection system.

In other applications where the heat dissipating surface is normally smooth, it may be necessary to enhance the surface to achieve the desired temperature level or rate of heat transfer. The traditional solution is to add vertical fins; however, roughening the surface would be a more attractive solution if the heat transfer coefficient increase is substantial. Few studies have been carried out to determine the effect of surface roughness elements on free convection heat transfer. The results obtained seem to conflict with each other as to the increase of the heat transfer coefficient.

Free convection

Since the pioneering experimental work of Ray [1] in 1920, natural or free convection has developed into one of the most studied topics in heat transfer. However, relatively little information is available on the effect of complex geometries on natural convection. Several studies have examined the effect of roughness on average heat transfer coefficients for vertical isothermal surfaces in a large enclosure (unbounded free convection).

A composite of published data for rough surfaces in free convection is shown in Fig. 1. Comparisons have been made using the projected base area as a reference. Jofre and Barron [2] obtained data for heat transfer to air from a vertical surface roughened with triangular grooves. At $Ra_L \approx 10^9$ they quoted an improvement in the average Nusselt number of about 200% relative to the turbulent predictions of Eckert and Jackson [3]. However, the flow was certainly not turbulent over the whole smooth plate. Compared to other solutions which consider upstream laminar flow, the improvement is closer to 100%. Furthermore, it appears that the reported Nusselt numbers are too high due to underestimation of the radiation correction [4].

Ramakrishna *et al.* [5] developed an analogy correlation of heat transfer data as obtained by Sastry *et al.* [6] for a vertical cylinder. Sastry *et al.* roughened the cylinder by wrapping a 0.45–1.45 mm diameter wire around it; the wrapping was done with a pitch equal to the wire diameter (i.e. no gap between wires). The enhancement in air was typically about 50%.

Heya *et al.* [7] conducted interferometric experiments on horizontal cylinders of 35 and 63 mm diameter with dense pyramid-type, streak-type, and check-type roughness elements of height varying from 0.15

NOMENCLATURE

g	gravitational acceleration [m s^{-2}]
Gr	Grashof number, $g\beta\Delta T x^3/\nu^2$
h	heat transfer coefficient [W m^{-2} K^{-1}]
k	thermal conductivity [W m^{-1} K^{-1}]
Nu	Nusselt number, hx/k
$\overline{Nu}$	average Nusselt number for test surface
p	rib pitch (step pitch) [m]
Pr	Prandtl number
q	height of rib (step) [m]
Ra	Rayleigh number, $Pr \cdot Gr$
s	width of rib [m]
s	distance from leading edge along profile [m]
T	temperature [°C]
ΔT	temperature difference, $T_w - T_a$ [°C]
x	longitudinal coordinate
x	distance from leading edge [m]
y	transverse coordinate.

Greek symbols

β	volumetric coefficient of thermal expansion [°C^{-1}]
θ	angle of inclination [deg]
λ	wavelength of light [m]
ν	kinematic viscosity [m^2 s^{-1}].

Subscripts

a	ambient
D	diameter
f	film
L	characteristic length
s	based on profile length
w	wall
x	local.

All properties are evaluated at the film temperature, $(T_w + T_a)/2$, unless otherwise noted.

to 0.72 mm and spacing varying from 0.76 to 2.00 mm. The streak-type roughness was formed by milling longitudinal grooves on the dense pyramid-type surface and the check-type roughness was formed by milling circumferential grooves on the streak-type surface. The tests were conducted using both water and air. No increase in average heat transfer coefficient was reported for the range of $4 \times 10^4 < Ra < 10^7$ for air and $3 \times 10^6 < Ra < 2 \times 10^8$ for water.

Fujii *et al.* [8] roughened large-diameter vertical cylinders with repeated ribs, dispersed protrusions, and closely spaced pyramids. Heat transfer to water and spindle oil was studied. Maximum increases in the average Nusselt number of 10% with water were observed relative to their previous data for a smooth cylinder. The latter data are only slightly higher than McAdams' correlation [9] for smooth vertical plates.

Prasolov [10] presented some results of an experimental investigation of the influence of machined roughness on a horizontal cylinder on heat transfer

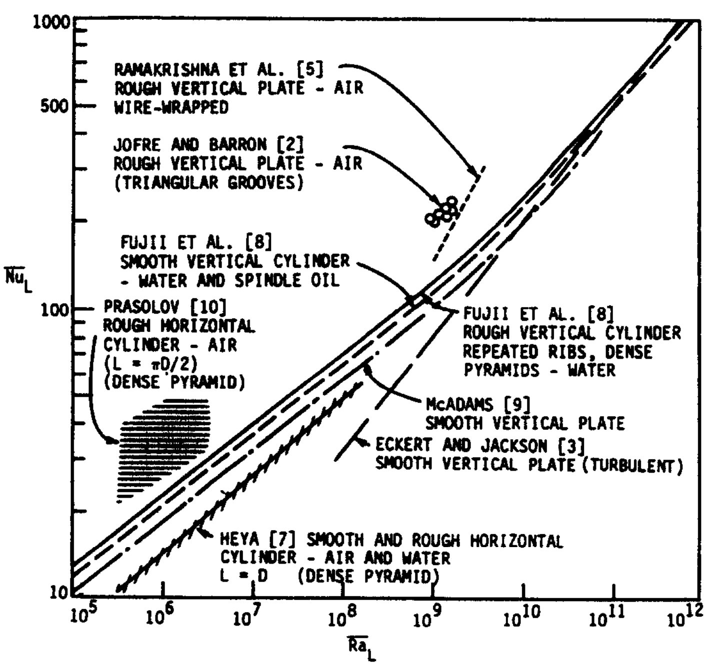


FIG. 1. Composite of data reported in the literature on rough surfaces in free convection.

to air. Densely packed pyramids of height varying from 0.08 to 0.36 mm and spacing varying from 1 to 2 mm were used. A very significant increase in the average Nusselt number was observed in the Rayleigh number range $5 \times 10^4 < Ra_D < 10^6$. For lower or higher values of the Rayleigh number, little enhancement was observed. A qualitative explanation is offered in terms of intensification of turbulence in the transition region. It is difficult to accept this, however, as turbulence is normally considered to occur at $Ra_D \approx 10^9$. The transition Grashof number for an isothermal vertical flat plate in natural convection in air, summarized by Godaux and Gebhart [11] from the results of various researchers, is between 1.1×10^8 and 1.0×10^{10}.

Several studies have been made on the effect that the angle of inclination has on heat transfer from a flat plate. Fujii and Imura [12] conducted experiments on a vertical plate inclined at arbitrary angles of inclination. They found that for the laminar regime the expression for the vertical plate, $Nu = K\,Ra^{0.25}$, is applicable to the inclined plate if the gravitational force term in the Rayleigh number is altered to the component parallel to the inclined surface. Their plate dimensions were 300 mm height and 150 mm width. The expression suggested for correlating the Nusselt number is

$$Nu = K(Gr \cdot Pr \cdot \cos\theta)^{0.25}$$

where the constant K is from the solution for the vertical plain plate. The applicable range of the angle of inclination in this expression, for hot plates facing upward, is limited by the occurrence of the plume that is characteristic of horizontal surfaces.

The same conclusions were drawn by Rich [13] in an earlier study. He carried out an interferometric study on an isothermal plain plate with angles of inclination varying between 0 and 40 deg to the vertical. The Grashof number range investigated was 10^6–10^9. Data were predicted to within 10% by modifying the vertical plate expression in the manner mentioned above.

Natural convection in enclosures

Although the study described here is one of free (unbounded) convection, a better understanding of the effect of roughness on the heat transfer characteristics would result if other flow regimes were also studied. With this in mind, studies of the effect of roughness on natural convection in enclosed spaces (bounded convection) have also been reviewed. There are several studies on the effect of roughness on natural convection in large enclosures.

Bohn and Anderson [14] recently studied natural convection heat transfer from large machine-roughened vertical surfaces in an enclosure. The study was carried out to simulate the interior of a passively solar heated building. They concluded that for an isothermal surface, the rough texture produced fully turbulent behavior at Ra_L about half that characterizing turbulence for a smooth surface. This produced increases in section average heat transfer of up to 40% and in the surface average heat transfer of about 16%. The results were complicated by recirculating flow in the enclosure. They found no enhancement in the laminar regime, stated to be $Ra_L < 2 \times 10^{10}$.

Note, however, that there are two differences between enclosure flow and infinite medium flow. In enclosure flow, due to the presence of a horizontal wall, fluid rising along a warm wall has different initial conditions than if the wall were an isolated plate in an infinite medium. In the latter case, both the velocity and the thermal boundary layers start at the leading edge of the plate. Moreover, the fluid has no component of momentum normal to the surface. In an enclosure, fluid approaches the vertical wall and must turn the corner to flow upward. Thus, the velocity boundary layer is somewhat developed before it reaches the corner.

In another recent study, Shakerin *et al.* [15, 16] determined the heat transfer characteristics of an enclosure with square-cross-sectioned ribs on the heated wall. The height of their roughness elements was about the order of the boundary layer thickness. The study was conducted up to Rayleigh numbers at which the smooth wall boundary layer normally undergoes transition. The study was conducted experimentally using a Mach–Zehnder interferometer and supported by numerical calculations. Nearly stagnant regions were observed between ribs. The surface heat flux was reportedly reduced in these regions to the extent that the total heat transferred from the surface was no larger than that from a smooth surface even though the surface area was increased. They attributed local increases at the rib surface to local flow acceleration in the 'attached' regions. The increase in average Nu_L was reported to be around 12% with one rib although the increase in surface area was about 32%. The enhancement obtained with two ribs was 16%. It was suggested that the spacing of roughness elements may be important.

Experiments conducted by Bohn *et al.* [17] with three-dimensional natural convection with water in a cubical enclosure showed that the mechanism of heat transfer is laminar boundary layer convection from one wall to the bulk fluid. They also concluded that the boundary layers are unaffected by the core flow. Flow visualization studies showed that these boundary layers are not greatly affected by the presence of adjacent walls, at least for aspect ratios near unity. This permits better comparison with two-dimensional flat plate studies. Rayleigh numbers, based on cell height, up to 6×10^{10} were tested.

Objectives

Based on the previous discussion of the literature, it appears likely that natural convective heat transfer coefficients on vertical surfaces can be enhanced using various types of surface roughness elements. The literature seems somewhat contradictory, however, with some researchers finding increases of 100% and others finding no increases or even decreases. If it is indeed possible to enhance heat transfer, then it is important to understand how various surface modifications produce this enhancement. This would then help determine the best type of enhancement to use. It is of equal importance to understand the behavior of rough surfaces on which the roughness occurs naturally, such as electronic circuit boards and microelectronic chip modules.

To resolve these issues, several repeated ribbed and stepped surfaces were studied. The particular shapes were selected, based on a study of the literature, as being the ones most likely to help achieve the dual objectives of gaining an insight into the nature of heat transfer from rough surfaces in laminar natural convection and suggesting a surface geometry that might enhance the heat transfer performance. The effects of parameters such as protuberance height-to-spacing ratios, conductivity of ribs, and angle of inclination were studied. Some preliminary results have been reported in Bhavnani and Bergles [18]. In addition, an earlier paper [19] reported the local heat transfer characteristics of wavy surfaces.

An optical technique was used to facilitate a nonintrusive analysis of local characteristics. This study is one of the first to experimentally determine the local heat transfer characteristics from a macroscopically roughened surface in unbounded free convection.

EXPERIMENTAL APPARATUS AND PROCEDURE

Interferometer

A Mach–Zehnder interferometer (MZI) with 100 mm optics was used in the experimental study. A schematic view is shown in Fig. 2. The light source used was a 2 mW helium–neon laser, $\lambda = 6328$ Å. The paths of the light beam from the laser to the camera are indicated in Fig. 2. Beam splitters SP2 and SP3, along with plane mirrors M1 and M2, constituted the basic MZI. The configuration was the same as that developed and used for a previous study of natural convection heat transfer [20]. Interferograms were recorded by a Canon TX camera on 35-mm fine grain Kodak Panatomic-X film having a speed of ASA 32.

All the components of the MZI were so positioned that the light beam was incident on each of them at an angle of 30 deg. This is different from the conventional MZI setting of 45 deg. The reason for using 30 deg is that a wider beam can be obtained with this angle, thereby increasing the field of view.

The sides of the interferometer were shrouded with a sheet of clear polyethylene to eliminate the effects of external disturbances caused by the movement of the operator. This also helped keep the optical components free from dust. Windows were cut in appropriate locations to facilitate entry of the beam from the laser and the exit of the beam to the camera.

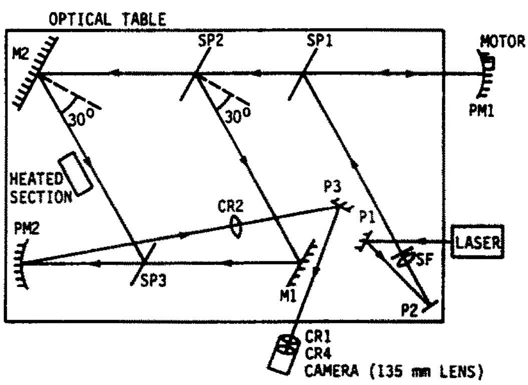
The interferometer was housed in a windowless room located in an internal part of the building so as to maintain a controlled atmosphere, an essential requirement for free convection studies in air. The supply from the building air-handling units to the room was shut off a few hours before experiments were conducted. The walls of the room were well insulated.

Test sections

Test sections were fabricated from 6.35-mm-thick aluminum plate. Care was taken during the machining of the base plate to ensure a very high degree of flatness and a mirror finish with sharp and burr-free edges. This helped improve the quality of the results and served as a very useful alignment tool.

Several different types of macroscopic surface contours were analyzed. The profiles selected evolved during the course of the study. Because the purpose of the study was to study the effect of roughness on natural convection heat transfer, ribbed surfaces were a natural first choice. The experience obtained with ribbed surfaces led to experimentation with stepped surfaces.

The test sections had a width of 127 mm parallel to the direction of the light beam and a length of 178 mm. The aluminum ribs were press-fit into the base plate to minimize contact resistance. The choice of aluminum was made because of its high conductivity and easy machinability. Studies were also conducted on surfaces with ribs made from low thermal conductivity material in order to assess the conjugate conduction–convection interaction. The plastic ribs used in the study were made from extruded acrylic



P1, P2, P3 - SMALL PLANE MIRRORS

SP1, SP2, SP3 - SPLITTER PLATES

PM1, PM2 - PARABOLOIDIAL MIRRORS

SF - SPATIAL FILTER

M1 - 0.102 m MIRROR

M2 - 0.152 m MIRROR

CR1, CR2, CR4 - CLOSE-UP RINGS #1, 2, AND 4

FIG. 2. Plan view of the ISU Mach–Zehnder interferometer.

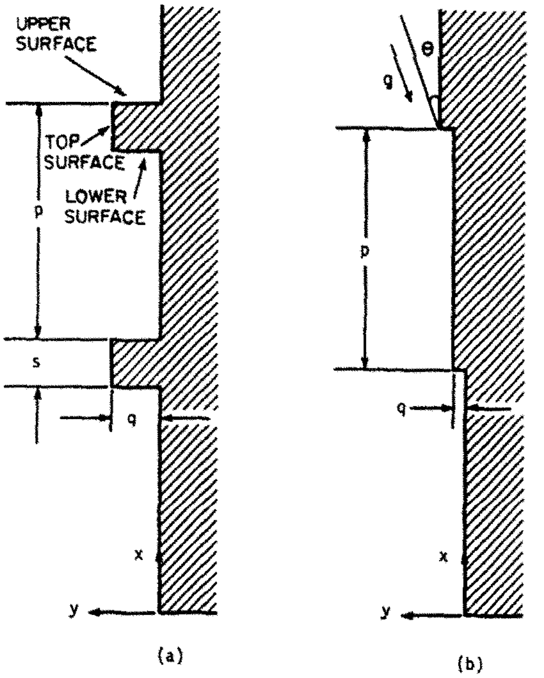


FIG. 3. Schematic of test geometry: (a) ribbed; (b) stepped.

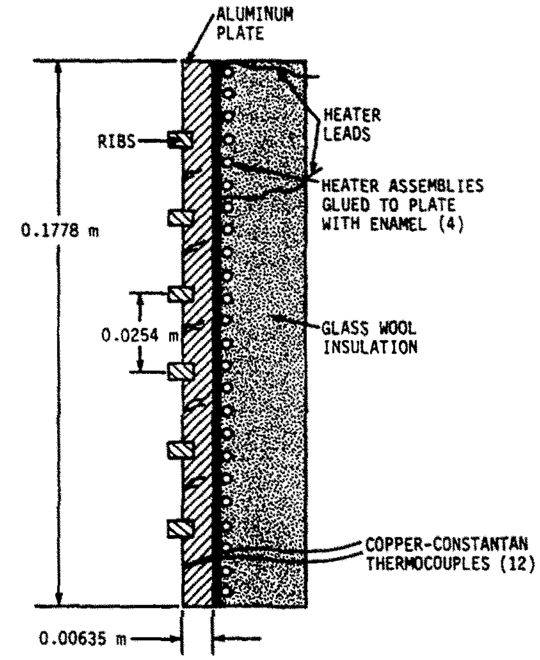


FIG. 4. Cross-sectional view of a ribbed test section showing fabrication details.

(Plexiglas) rods. These were attached to the base plate using an epoxy-based adhesive.

The ribbed surfaces tested had square cross-sectioned ribs of height (and width) equal to 6.35 and 3.18 mm, spaced at intervals of 25.4 mm, resulting in spacing-to-height ratios of 4 : 1 and 8 : 1, respectively. The stepped test sections had steps with a constant height of 1.59 mm at spacings of 12.7, 25.4, and 50.8 mm, yielding step spacing-to-height ratios of 8 : 1, 16 : 1, and 32 : 1, respectively. Schematics of the ribbed and stepped test sections are shown in Fig. 3.

Twelve 1.58-mm-diameter holes were drilled into the rear surface of the plates. These holes were in two rows 31.75 mm from the lateral edges at a spacing of 25.4 mm in the x-direction. Care was taken to drill the holes as close to the front surface as possible. Thirty-gauge copper–constantan (T-type) thermocouples were placed in these holes and fixed in place using an epoxy adhesive. To minimize lead wire conduction error, the leads were run about 40 mm along the plate before they emerged into the ambient air.

The rear surface of the plate was then coated with a layer of Glyptal Red Enamel electrical insulating paint. Four 26 gauge Nichrome electrical resistance heater assemblies were custom fabricated and secured to the surface using another layer of Glyptal Red Enamel. A cross-sectional view of a typical ribbed test section is shown in Fig. 4 as an example. The rear of the test section was insulated using 38-mm-thick glass wool to minimize heat loss. It is worth noting here that the actual magnitude of heat loss from the rear of the assembly is of no consequence since local measurements of heat flux are made entirely from the interferograms. For the same reason, measurements of radiation heat loss are not required.

Procedure

An isothermal boundary condition was established with the four individually controlled heater assemblies and monitored by the thermocouples. The typical variation in temperature over the plate was $\pm 0.3°C$ for a plate-to-ambient temperature difference of 30°C with plate surface temperatures of about 55°C, giving a percentage variation of $\pm 1.0\%$ based on plate-to-ambient temperature difference and $\pm 0.55\%$ based on plate wall temperature.

The test section assembly was suspended in the measuring path of the interferometer. Suspension from above was chosen in preference to support from below since bottom support would affect the natural convection boundary layer at the leading edge. Vertical movement was facilitated by the use of a laboratory jack. Indicator pins located at intervals of 25.4 mm (nominal) served a dual purpose, providing a convenient length scale for the interferograms and as position indicators. These were placed outside the region of interest and did not interfere with the boundary layer being studied.

The actual data collection procedure was preceded by a careful alignment of the interferometer to obtain a good infinite fringe pattern. The alignment procedure followed the suggestions given by Eckert and Goldstein [21] and Hauf and Grigull [22]. Once the interferometer was aligned, major readjustment of the components was never required. The minor adjustments occasionally required were accomplished by rotating beam splitter SP3 and mirror M1.

After initial alignment was completed, power to the heater assemblies was switched on. Steady state was achieved in about 3–4 h. The experiments were conducted at a temperature difference that was selected to result in a light fringe being the one closest to the wall. This greatly increased the accuracy with which the wall profile could be identified on the interferogram.

Care was taken to focus the camera–lens combination on an object at the center of the test section. Several interferograms had to be made to cover the entire length of the plate. The plate was traversed vertically about 25 mm using the jack, and the camera was kept stationary. After each traverse, the alignment was checked and slight corrections were made, if required. Several interferograms were made at each location. It was found that a shutter speed of 1/4 s and an f-stop of 3.5 gave the best results.

Composite interferograms of typical ribbed (high conductivity ribs) and stepped test sections are shown in Figs. 5 and 6.

The plates were tested at temperatures that varied from plate to plate between 48 and 75°C. Note that the temperature was not used as a variable, the variation coming about merely to obtain a conveniently readable interferogram. Grashof numbers up to 8×10^7 were tested.

The data reduction procedure involved analyzing the interferogram at several different x-locations. This was accomplished using a tool maker's microscope. The data consisted of fringe shift values in terms of number of wavelengths and fringe relative displacements. Based on operator experience only the dark fringes (complete destructive interference) were used in the data reduction since the error in reading the dark fringes is considerably lower than that in reading the light fringes (complete constructive interference). This is because the dark fringes are better defined. The fringe shift information obtained was then used in conjunction with equations of the variation of index of refraction of air to evaluate both a predicted wall temperature, T_w, using the ambient temperature as a reference, and the slope at the wall, dT/dy. A curve fitting program was used to fit a second order polynomial to the data obtained for fringe temperatures and locations, in order to accomplish this.

Note that for the ribbed surfaces with 3.2 mm square ribs, one series of temperature and fringe shift measurements was made at the middle of the lower surface (upstream side), the top surface, and the upper surface (downstream side) of the rib. Each of these sets of measurements was considered to be a representative average for that surface. For the larger ribs (7.9 mm × 6.4 mm), four series of measurements were made on the top surface of the rib because the heat transfer coefficient varied considerably over that surface. However, just one series of measurements was made on the lower and upper surfaces because the coefficient variation over these surfaces was small. Several sets of measurements were made on the base surface at different x-locations. All fringe shift measurements were made perpendicularly outward from the respective surfaces. The distance along the profile was considered as the x-coordinate distance for all test sections.

FIG. 5. Composite interferogram of a typical ribbed test section.

For the ribbed and stepped surfaces, the heat flow is clearly two-dimensional near the protuberances. However, it is still legitimate to carry out the measurements indicated even though only components of that heat flux are being measured. This is because the domi-

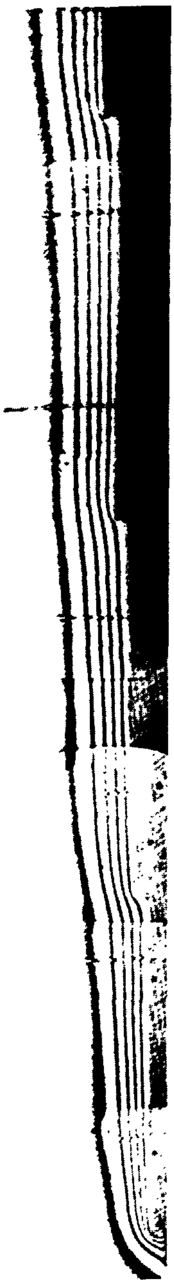


FIG. 6. Composite interferogram of a typical stepped test section.

nant component of the flux direction can be quite readily identified by looking at the isotherms on the interferogram.

The local heat transfer coefficients and Nusselt numbers were then calculated as follows:

$$h_x = -k \cdot \frac{dT}{dy} \cdot \frac{1}{(T_w - T_a)} \tag{1}$$

and

$$Nu_x = -\frac{dT}{dy} \cdot \frac{1}{(T_w - T_a)} \cdot x. \tag{2}$$

The local Grashof number was evaluated at the film temperature

$$T_f = \frac{T_w + T_a}{2}. \tag{3}$$

Additional details of the experiment and data reduction are available in ref. [23].

RESULTS AND DISCUSSION

All experimental results were obtained from the interferograms. The interferograms gave the complete air temperature distribution and local heat transfer in the vicinity of the heated surface. The overall surface-to-ambient temperature difference calculated from the interferograms generally differed from the thermocouple measurements by only 3–4%. This, however, did not influence the calculations because both temperatures and temperature gradients were calculated from fringe data [24].

Plain plate

A check was performed on the experimental method by obtaining interferograms and overall data for the vertical plain plate. Figures 7 and 8 show the local heat transfer coefficient plotted against distance from the leading edge and local Nusselt number plotted against local Grashof number, respectively. The solid line is a plot of the Ostrach [25] boundary layer analytical equation

$$Nu = 0.355Gr^{0.25}.$$

The constant in the above equation is a function of the Prandtl number and has been evaluated at a Prandtl number of 0.709. All experiments were conducted at temperatures that yielded this Prandtl number at the film temperature (to three significant figures).

There is a difference of 3.4% in mean integrated heat transfer coefficient between the data obtained from interferograms for a plain plate and those obtained using the Ostrach solution. It should be noted that points very near the leading edge do not agree too well with the theory because the boundary layer assumptions made in the theoretical development imply that the distance along the plate is large compared with the boundary layer thickness. This assumption is obviously invalid near the leading edge. Axial conduction, in both the fluid and the plate, is responsible for the lower experimental coefficients.

Experimental uncertainty and external disturbances are other possible causes for the difference between the Ostrach results and the experimental results obtained in the present study. Overall, the comparison with the analytical solution is good, indicating that the experimental method is adequate.

Ribbed surfaces

1. *High thermal conductivity ribs.* Figure 9 shows the variation of local heat transfer coefficients for a

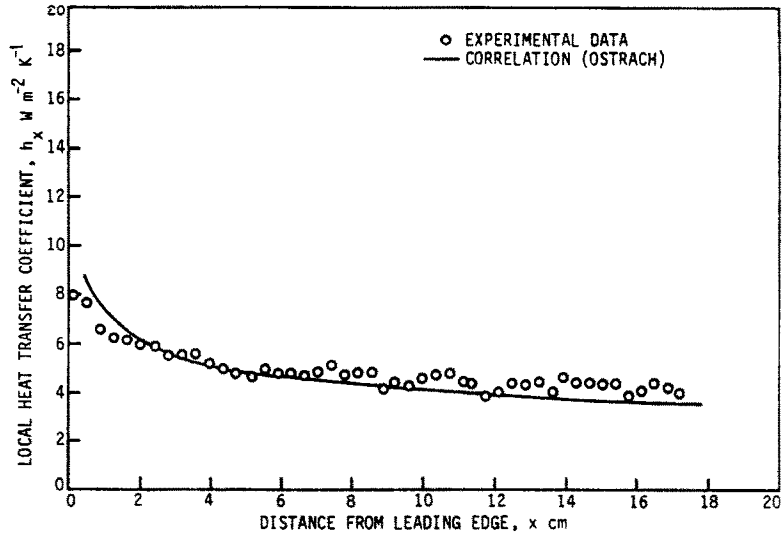


FIG. 7. Local heat transfer coefficient for a vertical isothermal plain plate.

ribbed test section (high thermal conductivity ribs) with rib pitch-to-height ratio $p/s = 8:1$ and rib pitch-to-width ratio $p/q = 8:1$. It can be seen that most of the coefficients are rather low with the exception of points located on the top of the rib. The coefficients are especially low just downstream and upstream of the rib. This is attributed to reduction in flow velocity due to the obstruction and the resulting thickening of the boundary layer at these locations.

A second set of clearly defined peaks in the heat transfer coefficient, of lesser magnitude, occurs between ribs. This is where the velocity of flow in the vertical direction in the boundary layer increases in magnitude, before once again encountering the retarding effect of the next rib downstream. This is clearly brought out in the composite interferogram (Fig. 5). The heat transfer from this ribbed surface was 10.1% lower than that for a plain plate of equal projected area (and ΔT) in spite of an increase in surface area of 25.0%, due to the upper and lower surfaces of the ribs. When compared with a plain plate of equivalent surface area, the performance was 23.1% lower for the ribbed surface.

The stagnation zone effect was even more pronounced in the test section with larger ribs having $p/s = 16:5$ and $p/q = 4:1$, as indicated by the data given in Fig. 10. For this geometry, the reduction in heat transferred was 26.3% below that for a plain plate of equal projected area in spite of a surface area increase of 50.0%. The reduction in heat transfer was

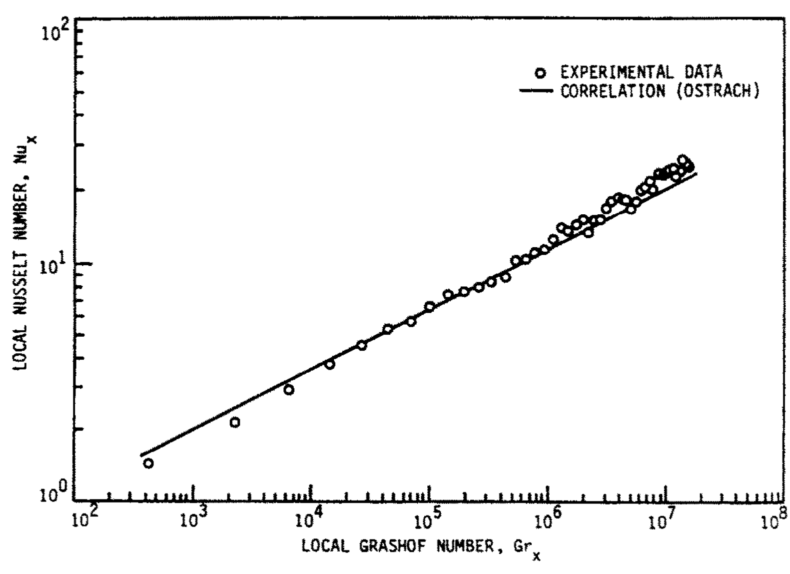


FIG. 8. Local Nusselt number for a vertical isothermal plain plate.

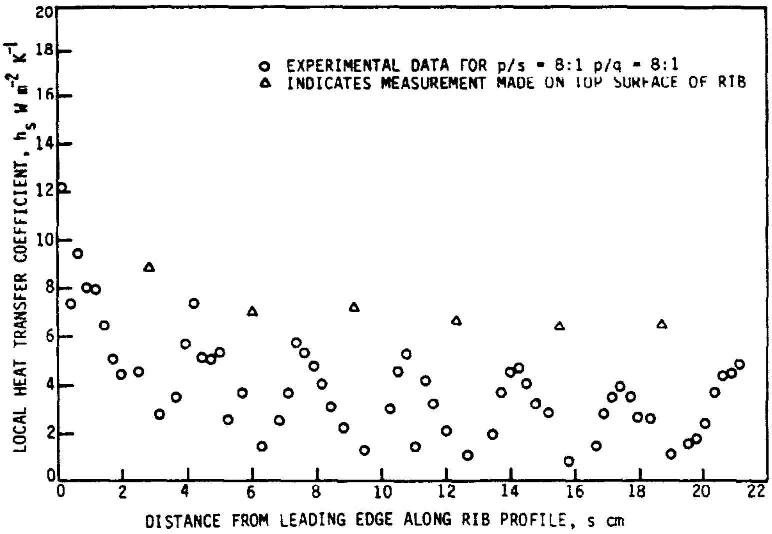


FIG. 9. Local heat transfer coefficient for ribbed plate with $p/s = 8:1$ and $p/q = 8:1$ (high conductivity ribs).

44.3% when an equivalent surface area plain plate was used as the reference. It is interesting to note the variation of the heat transfer coefficient on the top surface of the ribs. It is highest at the edge which is closer to the leading edge, then drops off towards the middle of the top of the rib and increases slightly as it approaches the downstream edge.

A summary of all experimental results is presented in Table 1.

2. *Low thermal conductivity ribs.* Experiments were also conducted on ribbed plates to study the effect of low thermal conductivity ribs on the heat transfer characteristics. Plexiglas with a thermal conductivity of ≈ 0.134 W m^{-1} K^{-1} was used. This is approximately three orders of magnitude lower than aluminum which has a thermal conductivity of ≈ 119 W m^{-1} K^{-1}. Note, in this regard, that the thermal conductivity for air is ≈ 0.027 W m^{-1} K^{-1}. This configuration is actually the reverse of the situation encountered in electronics applications, where heat-generating components are mounted on a low-conductivity base.

Data for the test section with $p/s = 8:1$ and $p/q = 8:1$ are plotted in Fig. 11. The composite interferogram is shown in Fig. 12. As in the case of the high thermal conductivity ribs, an overall decrease in heat transfer performance is observed. This is again due to the local thickening of the boundary layer just

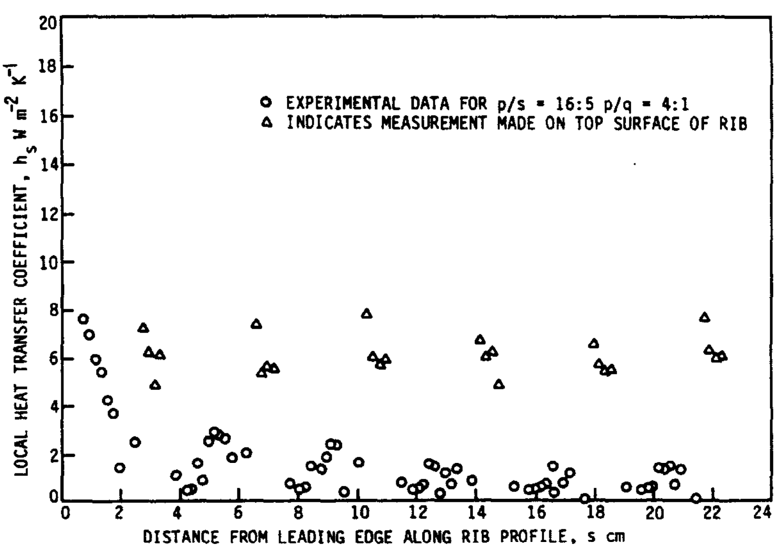


FIG. 10. Local heat transfer coefficient for ribbed plate with $p/s = 16:5$ and $p/q = 4:1$ (high conductivity ribs).

Table 1. Summary of experimental results

Description of surface geometry	Experimental q [W]	Reference q [W] (based on profile length)	Percentage change (%)	Reference q [W] (based on projected length)	Percentage change (%)
Ribs, $p/q = 8:1$, $p/s = 8:1$, high k	$0.826(L)\Delta T$	$1.072(L)\Delta T$	−23.06	$0.926(L)\Delta T$	−10.15
Ribs, $p/q = 16:5$, $p/s = 4:1$, high k	$0.568(L)\Delta T$	$1.020(L)\Delta T$	−44.30	$0.771(L)\Delta T$	−26.31
Ribs, $p/q = 8:1$, $p/s = 8:1$, low k	2.816	3.490	−19.31	2.746	−4.54
Ribs, $p/q = 16:1$, $p/s = 16:1$, low k	2.701	2.905	−7.02	2.629	2.75
Steps, $p/q = 8:1$, vertical	$0.651(L)\Delta T$	$0.678(L)\Delta T$	−4.03	$0.613(L)\Delta T$	6.14
Steps, $p/q = 16:1$, vertical	$0.867(L)\Delta T$	$0.732(L)\Delta T$	18.50	$0.704(L)\Delta T$	23.21
Steps, $p/q = 32:1$, vertical	$0.876(L)\Delta T$	$0.812(L)\Delta T$	7.94	$0.794(L)\Delta T$	10.30
Steps, $p/q = 8:1$, inclined	$0.757(L)\Delta T$	$0.804(L)\Delta T$	−5.78	$0.740(L)\Delta T$	2.31
Steps, $p/q = 16:1$, inclined	$0.694(L)\Delta T$	$0.713(L)\Delta T$	−2.61	$0.686(L)\Delta T$	1.24
Steps, $p/q = 32:1$, inclined	$0.846(L)\Delta T$	$0.780(L)\Delta T$	8.30	$0.769(L)\Delta T$	10.03

upstream and downstream of a rib. However, from the interferogram it can be seen that this thickening is not as pronounced as in the case of high thermal conductivity ribs, and local coefficients just upstream and downstream of the ribs are not as low as the ones measured for the metallic ribs. In general, one can observe that wall heat conduction in the rib tends to reduce the average temperature difference between the rib and the adjacent fluid. This, combined with the decrease in flow velocity, decreases natural convection heat transfer from these horizontal surfaces.

Local heat transfer coefficients on the upper and lower surfaces of the rib are almost zero in magnitude. This leads to the conclusion that these surfaces are essentially adiabatic in nature. This conclusion is further reinforced by the shape of the isotherms seen on the composite interferogram shown in Fig. 12. The dark isotherm closest to the plate surface is almost perpendicular to the upper and lower surfaces of the ribs. This was not so in the case of the metallic ribbed surfaces. When metallic ribs were used, the dark isotherm closest to the plate surface curved around the

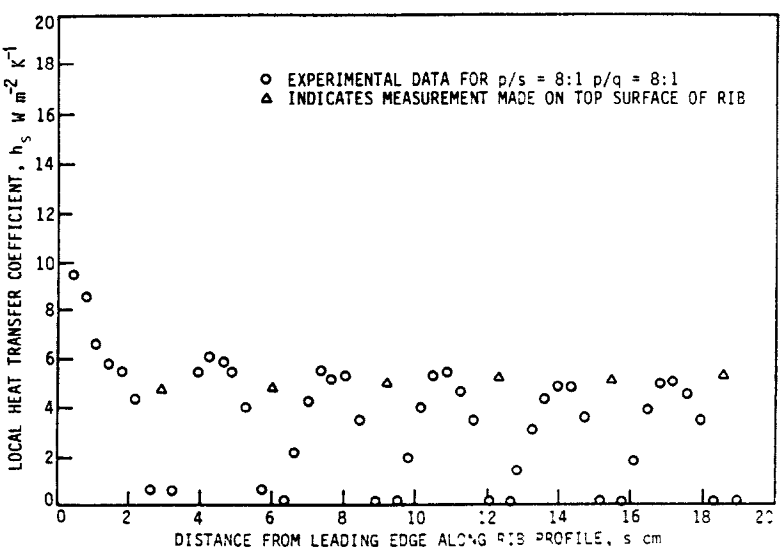


FIG. 11. Local heat transfer coefficient for ribbed plate with $p/s = 8:1$ and $p/q = 8:1$ (low conductivity ribs).

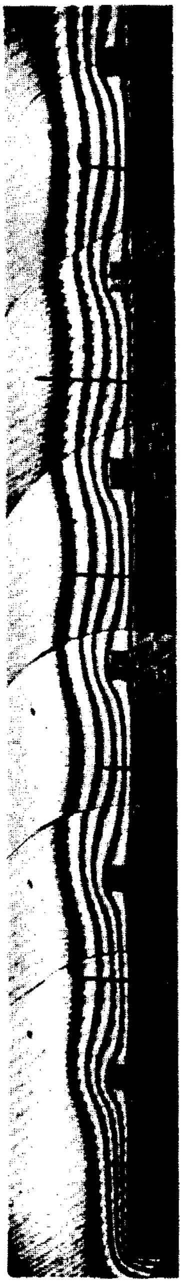


FIG. 12. Composite interferogram of a ribbed plate with $p/s = 8:1$ and $p/q = 8:1$ (low conductivity ribs).

rib. For the low-conducting ribs there is a pronounced temperature gradient in the y-direction within the rib material along the upper and lower surfaces.

Note also that the heat transfer coefficient at the top of the rib is not as high as with the metallic ribs. This is only to be expected since the temperature difference between the top of the rib and the fluid is lower than that at the baseplate. The secondary peak between ribs, where the boundary layer thickness is reduced, is much more pronounced and the heat transfer coefficients are much higher at these locations. This is because the thermal boundary layer is not thickened at the ribs as much as it was in the case of metallic ribs because of the low thermal conductivity of the non-metallic ribs.

The net result of this complex thermally induced convection is that the heat transfer is not as low as for conducting ribs. The reduction in heat transfer in this case is 4.5% compared to 10.1% for metallic ribs with the same geometry, with the surface area being increased by 25.0% in both cases.

The results for the surface with low-conducting ribs with $p/s = 16:1$ and $p/q = 16:1$ are presented in Fig. 13. The results are somewhat better than those for the case of the surface with rib pitch-to-height and pitch-to-width ratios of 8 : 1. The heat transfer performance is actually 2.8% higher than that for a plain plate of equal projected area. However, note that the increase in area is 12.5% with this geometry. In this case, the peaks in the value of local heat transfer coefficient occur just downstream of the rib after which there is a steady decrease in magnitude to nearly zero just before the next rib. Again, the heat transfer coefficient on the top surface of the rib is not as dramatically increased as in the metallic ribbed surfaces. The adiabatic nature of the upper and lower surfaces of the rib was observed in this case, too.

Stepped surfaces

A stepped profile was considered next in order to reduce the effect of the dead zones observed with ribbed surfaces. The effects of step pitch-to-height ratio and plate orientation were studied in an attempt to find a surface that actually enhanced the heat transfer.

1. *Vertical orientation.* Figures 14–16 illustrate the variation of local coefficients for the stepped test sections with step pitch-to-height ratios, $p/q = 8:1$, $16:1$, $32:1$, respectively. These surfaces have been oriented such that the plate, in effect, consists of a series of vertical segments where each segment is offset from the preceding segment by a short horizontal step.

It is readily apparent that the heat transfer performance of this type of surface is far superior to that of the ribbed plates studied. One obvious reason is the complete elimination of the downstream dead zone caused by this configuration. For the test sections with pitch-to-height ratios 16 : 1 and 32 : 1, almost all the data points lie above the exact solution for a plain plate. This results in a 23.2% increase in heat transfer for the test section with $p/q = 16:1$ and 10.3% for the test section with $p/q = 32:1$ when compared with a plain plate of an equal projected area. The increase in surface area was 6.3 and 3.1%, respectively. As before, the peaks occur just after the steps as was the case with the ribs. Also, there is no readily observable second set of peaks, as is to be expected.

The data for the test section with $p/q = 8:1$ exhibit wide fluctuations similar to the variations for the ribbed test sections, with many data points falling well below the exact solution for a plain plate. The percentage increase in integrated mean heat transfer

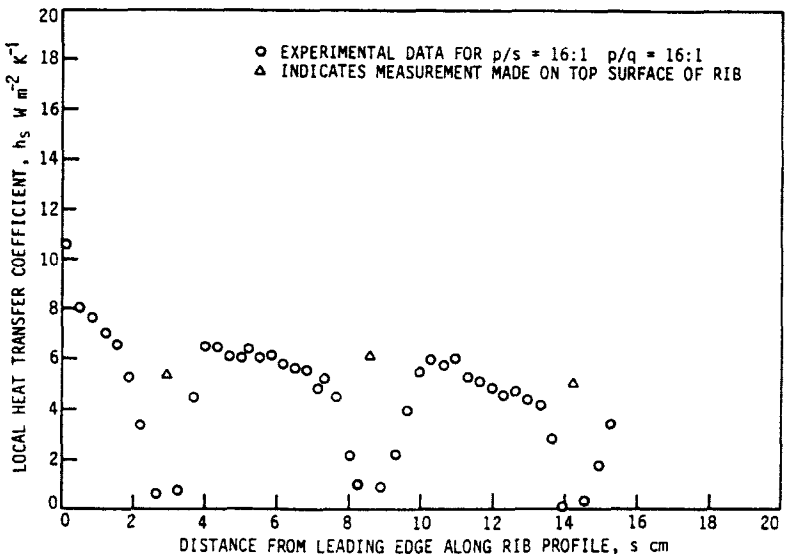


FIG. 13. Local heat transfer coefficient for ribbed plate with $p/s = 16:1$ and $p/q = 16:1$ (low conductivity ribs).

coefficient for this case was 6.1% while the area increase was 12.5%. This suggests that the steps may be too frequent, causing a reduction of flow velocity, and therefore heat transfer coefficient, upstream of the rib, thereby offsetting the local increase in heat transfer coefficient obtained downstream of the step location.

The results obtained indicate the presence of an optimum step pitch-to-height ratio. High values of p/q permit the boundary layer to thicken considerably between steps, thereby reducing the improvement in heat transfer performance. Low values of p/q, on the other hand, cause the same effect by reducing the flow velocity.

The surface of the step perpendicular to the x-direction is essentially adiabatic, as is apparent from the shape of the isotherms in Fig. 6. Therefore, the heat transfer from this surface is negligible.

The increase in heat transfer coefficient is more pronounced as the upper edge of the plate is approached. This is reasonable as the ribs or steps should have a greater influence as the transition Grashof number ($Ra_L \approx 10^9$) is approached, especially in light of the increased sensitivity to low magnitude external disturbances. This suggests that both ribbed and stepped surfaces may be useful in improving heat transfer performance in the transition regime, in accordance with the results presented in Bohn and

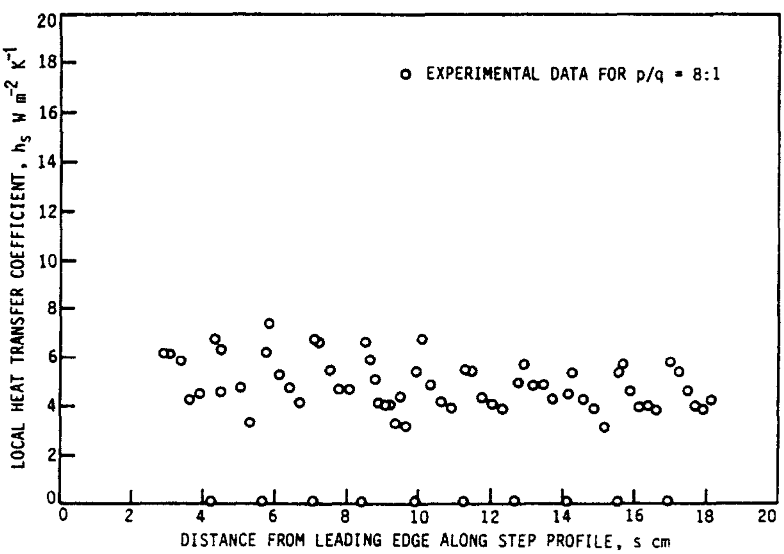


FIG. 14. Local heat transfer coefficient for stepped plate with $p/q = 8:1$.

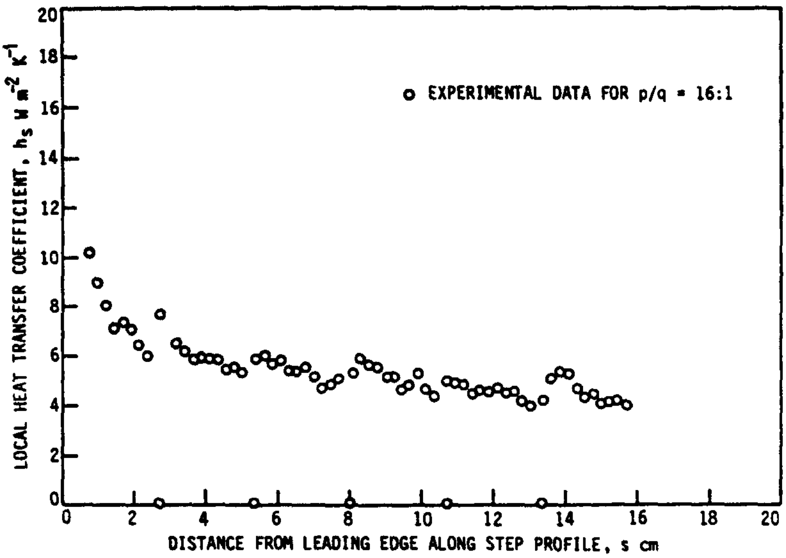


FIG. 15. Local heat transfer coefficient for stepped plate with $p/q = 16:1$.

Anderson [14]. Recall that the same behavior was also exhibited by all the ribbed surfaces.

2. *Inclined surfaces*. The effect of small angles of inclination was studied with the stepped surfaces. In these tests the plates were oriented such that instead of short vertical segments as before, the tips of each segment were in a vertical line. This yielded angles of inclination of 1.9°, 3.7°, and 7.2° for the plates with $p/q = 32:1, 16:1$, and $8:1$, respectively. These results are shown in Figs. 17–19.

The overall performance for the plates decreased compared to that for the plates with vertical segments (zero angle of inclination). The plate with $p/q = 32:1$ did not exhibit much difference in performance, but this result is to be expected since the angle of inclination in this case was very low (1.9°). The increase in heat transfer above a plain plate of equal projected area was reduced slightly to 10.0%. There was a much greater decrease in the performance of the plates with $p/q = 16:1$ and $32:1$. The values dropped to almost equal the heat transfer for a plain plate.

In general, the thickening of the boundary layer just upstream of a step was more pronounced for the inclined orientation for all cases. It was also noted that the expression suggested by Fujii and Imura [12] for laminar free convection flow along a vertical plain plate at small angles of inclination

$$Nu = K(Gr \cdot Pr \cdot \cos\theta)^{0.25}$$

was not adequate for the stepped surface.

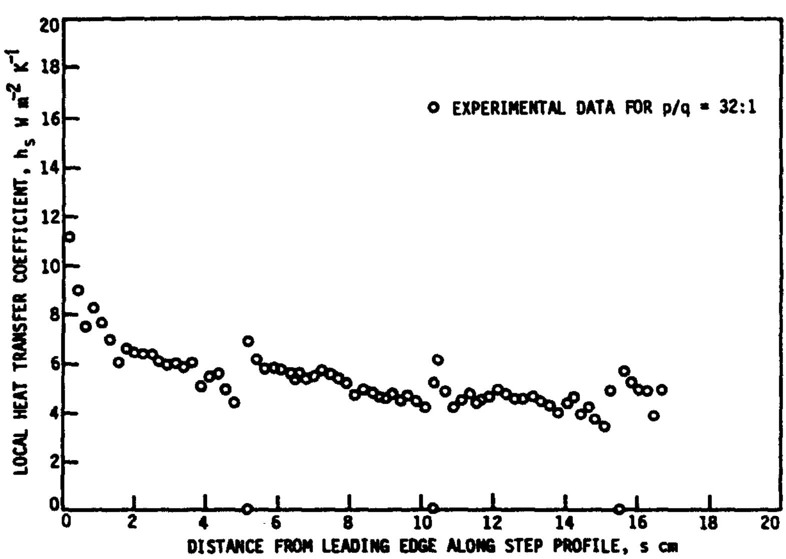


FIG. 16. Local heat transfer coefficient for stepped plate with $p/q = 32:1$.

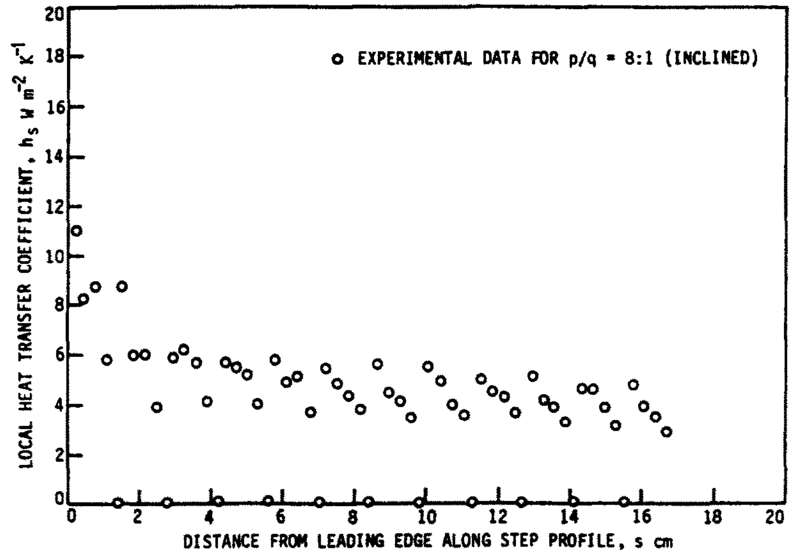


FIG. 17. Local heat transfer coefficient for stepped plate with $p/q = 8:1$, inclined at an angle of 7.1°.

In these experiments again, almost all the data points for $p/q = 32:1$ fell above the correlation for the plain plate. However, for $p/q = 8:1$ and $16:1$, almost half the data points fell below the correlation. This serves to show that the local increases in heat transfer coefficient just downstream of the steps are completely offset by the coefficient decrease upstream of the steps.

The local experimental results obtained in this study suggest that both the height of the roughness and the spacing between elements are of great importance. It is interesting to qualitatively examine the previously reported work with regard to this point, and attempt to explain the reasons for the contradictory results that were alluded to earlier.

Discussion of results reported in literature

The increase in heat transfer coefficient reported by Jofre and Barron [2] may be attributed to the effect the triangular grooved surface (height 0.76 mm and spacing 0.89 mm) had on promoting an early transition to turbulence, because the data are at relatively high Rayleigh numbers of about 10^9.

Fujii *et al.* [8] do not report significant increases in the Nusselt number in their experiments, which can be attributed to the fact that the dense pyramid-type roughness used is not conducive to increasing heat transfer in the laminar region. This is because individual roughness elements would lie in the downstream dead zone caused by the preceding roughness element. For the Fujii *et al.* tests conducted using

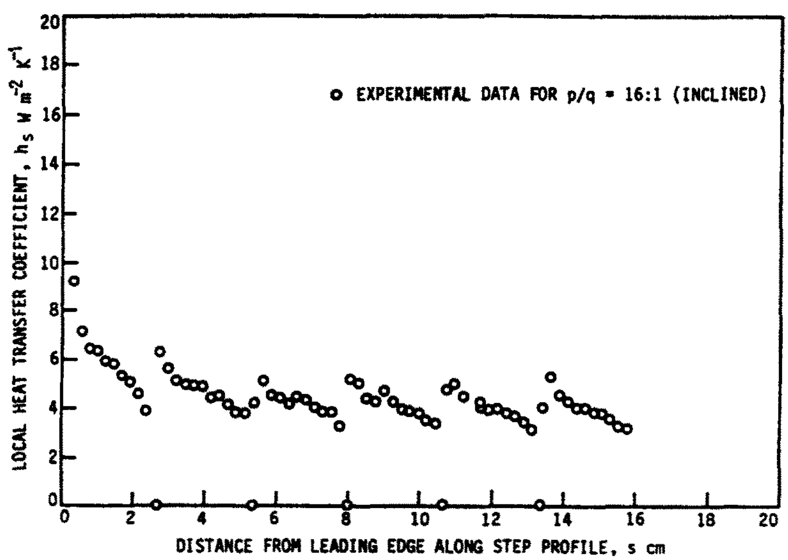


FIG. 18. Local heat transfer coefficient for stepped plate with $p/q = 16:1$, inclined at an angle of 3.6°.

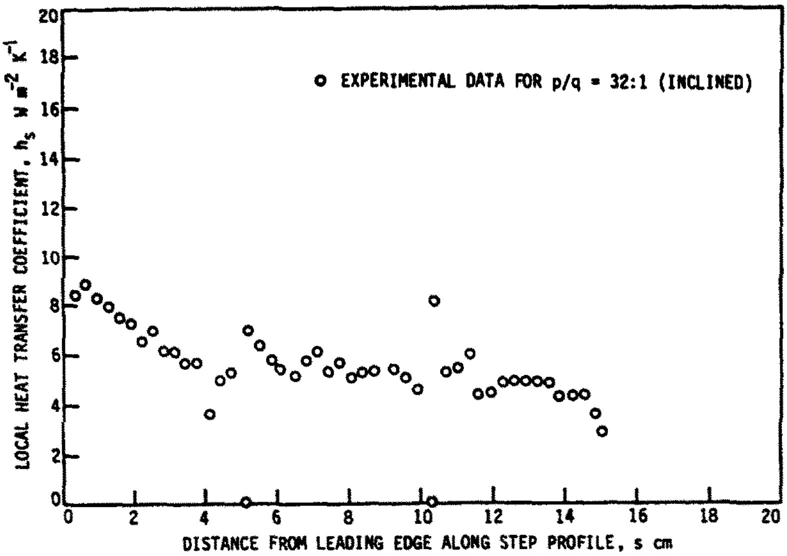


FIG. 19. Local heat transfer coefficient for stepped plate with $p/q = 32:1$, inclined at an angle of 1.9°.

repeated ribs, it is likely that their rib height of 0.5 mm was insufficient to disturb the boundary layer but enough to cause a dead zone; therefore, a thickening of the boundary layer occurred.

The data of Heya *et al.* [7] show no increase in the Nusselt number using dense pyramid-type, streak-type, and check-type roughness elements. Because the testing was done in the laminar region, this again supports the hypothesis that this type of roughness is not effective in this flow regime, because the boundary layer is stable enough to resist tripping.

The experiments of Bohn and Anderson [14] were performed using a machine-roughened plate consisting of two sets of grooves 1 mm deep and 1 mm wide cut at right angles to each other. This type of surface, for reasons similar to those expressed for Fujii's data, may not be best suited for laminar natural convection enhancement. This is suggested by the results reported in ref. [14], where increases in Nusselt number in the laminar region are only about 4% and those in the transition region are about 16%.

Shakerin *et al.* [15, 16] in their experiments also found that heat transfer coefficients were lower just above and below a rib element compared to the value on a smooth wall. They also found that the temperature gradients were higher at the top surface of a rib, especially at the leading edge. The type of roughness element used in their study was similar to the ribs used in the present study, the only difference being that their study was made in an enclosure. It was also reported that the effect of the roughness was mainly localized to within about two rib heights above and below the rib. An improvement in heat transfer performance is reported that is less than the increase in surface area. This study supports the findings in the present study.

Thus, it is evident that with proper sizing and shape selection, enhancement of heat transfer from a vertical plate is possible in natural convection, even well into the laminar region. This could be an important consideration in the positioning of electronic components since the heat transfer characteristics of an element are strongly affected by the flow generated around it by the preceding element. It must be emphasized that the detailed local measurements made possible by interferometry cannot be obtained by calorimetric methods.

CONCLUSIONS

An experimental investigation using a Mach–Zehnder interferometer to study structured, isothermal vertical surfaces is reported here. The purpose of the study was to determine the effect of transverse roughness elements on natural convection heat transfer. The major results of the experiments are as follows.

(1) Enhancement of heat transfer can be accomplished in the laminar flow regime in natural convection using transverse roughness elements with proper sizing and shape selection. The maximum increase observed in the present study was 23%, when a plain plate of equal projected area was used for comparison. This was obtained with a stepped configuration with a step pitch-to-height ratio of 16:1.

(2) The performance of ribbed surfaces was below that of a plain plate of equal projected area. The heat transfer performance decreased as the rib pitch-to-height ratio decreased.

(3) The ribbed surfaces with low thermal conductivity ribs had a heat transfer performance that was slightly improved (about 5%) relative to a geometrically identical configuration with ribs of high

thermal conductivity. This was because of a less pronounced stagnation zone.

(4) Stepped surfaces improved the heat transfer performance relative to a plain plate of equal projected area. This was accomplished with an orientation that consisted of a series of vertical segments, each projecting out from the previous upstream segment. The local heat transfer coefficient in each segment was highest at the upstream edge and gradually decreased as the downstream edge was approached. The greatest improvement observed was with a step pitch-to-height ratio of 16:1.

(5) Altering the orientation of the stepped surface to keep all the step tips in a vertical plane reduced the heat transfer performance relative to the orientation consisting of a series of vertical segments. This reduction was most significant in the section with a pitch-to-height ratio of 16:1. The surface with a pitch-to-height ratio of 32:1 was least affected since the angle of inclination resulting from this orientation was very low.

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EFFET DE LA GEOMETRIE DE LA SURFACE ET DE L'ORIENTATION SUR LA CONVECTION NATURELLE THERMIQUE LAMINAIRE POUR UNE PLAQUE PLANE VERTICALE AVEC DES ELEMENTS TRANSVERSAUX

Résumé—Une technique interférométrique est utilisée pour déterminer les coefficients locaux de transfert de chaleur pour des surfaces avec des nervures et des créneaux. On étudie les effets des paramètres tels que le rapport hauteur/pas des protubérances, la conductivité des nervures et l'angle d'inclinaison. On trouve que l'augmentation du transfert thermique, par rapport à une surface verticale lisse de même surface projetée, est possible en convection laminaire en utilisant des éléments transversaux de dimension et forme convenables. En général, les surfaces avec créneaux améliorent le transfert thermique. L'accroissement maximal du coefficient moyen de convection est de 23,2% pour un rapport pas/hauteur de 16. L'étude indique la présence d'un rapport optimal pas/hauteur. Toutes les surfaces avec nervures correspondent à des performances de transfert thermique dégradées.

EINFLUSS DER OBERFLÄCHENGEOMETRIE UND -AUSRICHTUNG AUF DIE LAMINARE NATÜRLICHE KONVEKTION AN EINER SENKRECHTEN EBENEN PLATTE MIT QUERLIEGENDEN KÜNSTLICHEN AUFRAUHUNGEN

Zusammenfassung—Mit Hilfe einer interferometrischen Versuchstechnik wurde der örtliche Wärmeübergang an Oberflächen mit wiederholten Rippen und Stufen bestimmt. Dabei wurde der Einfluß folgender Parameter untersucht: Verhältnis von Höhe zu Abstand der Aufrauhungen, Wärmeleitfähigkeit der Rippen und Neigungswinkel. Es zeigt sich eine Verbesserung des Wärmeübergangs—in Bezug auf eine glatte, senkrechte Oberfläche gleicher Größe—wenn bei laminarer natürlicher Konvektion querliegende Rauhigkeitselemente geeigneter Größe und Form verwendet werden. Ganz allgemein verbessern gestufte Oberflächen den Wärmeübergang. Die maximale Erhöhung des mittleren Wärmeübergangskoeffizienten beträgt 23,2% bei einem Verhältnis des Stufenabstands zur Stufenhöhe von 16. Die Untersuchung zeigt, daß es ein optimales Verhältnis gibt. Im Gegensatz dazu weisen berippte Oberflächen stets einen verminderten Wärmeübergang auf.

ВЛИЯНИЕ ГЕОМЕТРИИ И ОРИЕНТАЦИИ ПОВЕРХНОСТИ НА ТЕПЛОПЕРЕНОС ОТ ВЕРТИКАЛЬНОЙ ПЛОСКОЙ ПЛАСТИНЫ С ЭЛЕМЕНТАМИ ПОПЕРЕЧНОЙ ШЕРОХОВАТОСТИ ПРИ ЛАМИНАРНОЙ СВОБОДНОЙ КОНВЕКЦИИ

Аннотация—Для определения локальных коэффициентов теплопереноса для поверхностей с оребрением используется интерферометрический метод. Исследуются эффекты таких параметров, как отношение высоты ребер к шагу, теплопроводность ребер и угол наклона. Найдено, что интенсификация теплопереноса по сравнению со случаем плоской вертикальной поверхности с равной площадью проекции возможна при ламинарной свободной конвекции с использованием элементов поперечной шероховатости соответствующего размера и формы. В целом шероховатые поверхности интенсифицируют теплоперенос. Максимальное увеличение среднего значения коэффициента теплопереноса составляло 23,2% при отношении шага ребер к высоте, равном 16. Исследование показало, что существует оптимальное значение данной величины. Для всех оребренных поверхностей характеристики теплопереноса ухудшены.