HEAT TRANSFER FROM AN ELECTRONIC MODULE PLACED DOWNSTREAM OF A FENCE

S. A. ABDEL-MONEIM*, A. R. EL-SHAMY*, E. F. ATWAN* and A. M. ESMAEEL**

ABSTRACT

The present work investigates experimentally the heat transfer and flow characteristics for an electronic module mounted on a simulated printed circuit board placed downstream of a fence on the bottom wall of a duct with an aspect ratio of 4. The module height (B) to channel height (H) ratio is fixed at B/H=0.317. Three different values for the fence height (b) and four values for the spacing between the fence and the module (S) were investigated in such a manner that the ratio b/B=1, 1.5 and 2, and the ratio S/B=1, 2, 3, and 4. Reynolds number, based on the streamwise length of the module (L), was ranged from 8000 to 40000. The results for the module without fence displayed some noticeable differences as compared to the smooth duct flow due to the existence of the separation-reattachment flow patterns. Secondary vortex tubes were existed at the module sides due to the presence of the fence and these vortices have significant effects on both the heat transfer and flow friction. Both Nusselt number and Fanning friction factor are strongly dependent on Reynolds number while they are critically dependent on both the fence height and spacing. A maximum Nusselt number enhancement ratio (Nu/Nu_{N-F}) of about 1.94 was obtained corresponding to a friction factor ratio (F/F_{N-F}) of about 6.62 at the condition (Re=8700, b/B=1.5 and S/B=2). It was found that the fence with b/B=1 provides the best performance based on the criterion of equal mass flow rate. New correlations were obtained for the average Nusselt number and the Fanning friction factor utilizing the present measurements within the investigated range of the different parameters.

Keywords: Heat transfer, enhanced cooling, turbulence generator, electronic modules, fence

NOMENCLATURE

SI system of units is used for the whole parameters within the present paper.

Symbols:

B	module height	L	module length
b	fence height	ℓ	test section length
c_p	specific heat at constant pressure	P	flow pressure
D	duct hydraulic diameter	q	heat flux
H	duct height	S	the spacing between fence and
h	heat transfer coefficient		the module
k	thermal conductivity	T	temperature
		V	flow velocity

*   *Mech. Eng. Dept., Faculty of Eng. (Shoubra), Zagazig University,*
      E-mail: Sayed_moneim@ hotmail.com
**   *M.Sc. Student*

<u>Subscripts:</u>
f	fluid inlet
N-F	case of no fence
s	mean surface

<u>Greek letters:</u>
Δ	difference value
ρ	density
ν	kinematic viscosity

<u>Dimensionless terms:</u>
F	Fanning friction factor
Nu	average Nusselt number
Re	Reynolds number based on the module streamwise length

1. INTRODUCTION

The trend toward higher power density in electronic packages coupled with temperature limitations on device and package components for reliability and operability purposes has placed increased emphasis on thermal control of electronic components. The heat flux levels within the modern devices have increased by several orders of magnitude. Recently, electronic circuit densities continuing to increase at a rate of 30% per year and the problems associated with the thermal control of these devices will continue to multiply. In the high capacity power components the heat flux, the required thermal dissipation rate, frequently reaches 50 W/cm^2 and in some cases as high as 200 W/cm^2.

Therefore, thermal control of electronic components has one principal objective, to maintain relatively constant component temperature equal to or below the manufacturers maximum specified service temperature (typically between 85 and 100 °C). It have been demonstrated that components operate at 10 °C beyond this maximum temperature can reduce the reliability of the system by as much as 50%. For this reason, good thermal control schemes must be capable of eliminating hot spots within the electronic devices, removing heat from these devices, and dissipating this heat to the surrounding environment. Recent developments of thermal control schemes include various forms of cooling such as liquid cooling or conduction enhancement; however, many manufactures prefer air as a coolant because of its ready availability and the mechanical simplicity of the handling systems.

Several experimental studies of heat transfer from three dimensional modules resembling electronic packages, alone or in arrays have been discussed. Forced convection data from single electronic packages mounted on a circuit board were presented by Buller and Killburn [1], both with and without attached finned heat sinks. These data were correlated successfully to within 15%.

Anderson and Moffat [2], discussed forced convection heat transfer in a channel populated with discrete components similar to those found in electronics cooling situations. The temperature rise of each component is expressed as the sum of two parts; its adiabatic temperature rise, due to the thermal wakes of upstream components; and its self heating temperature rise due to its own power dissipation. Nakayama [3] defined a temperature scaling factor relating the local adiabatic temperature rise to the mean temperature rise, and showed that it could be derived

from the ratio of the heat-transfer coefficient from an element in a heated array, and one in an unheated array.

Heat-transfer measurements on an array of modules with only one active (heated) element were carried out by Moffat and Ortega [4]. It was found that the value of heat-transfer coefficient can be properly applied in a fully heated array by considering the adiabatic temperature rather than to the local mixed mean temperature. These values were found to be quite different depending on the level of diffusion and turbulent mixing. Moffat et al [5] investigated flow and heat transfer from arrays of 12.7 mm cubical elements mounted on adiabatic plates. Cubical elements have four-fifths of their total exposed surface area on the lateral surfaces, whereas electronic packages such as leadless chip carriers and flat packs have most of their exposed area on the top surface, hence, differences were expected in their respective behavior. In the relatively sparse array, Moffat et al. [5] found that heat transfer from an element was dependent on channel spacing and array spacing, and also on both the mean local velocity around an element, which was different than the mean channel velocity, and on the level of turbulence. Piatt [6] measured heat-transfer coefficients for arrays of short cylindrical elements on an adiabatic channel wall and found behavior similar to that of Moffat et al.[5] for cubes.

An indication of the difference in behavior between arrays of tall elements, such as cubes and cylinders, and short, flat elements, such as flat packs, was given by the data of Wirtz and Dykshoorn [7]. The heat transfer coefficients on arrays of aluminum modules, measuring 25.4 × 25.4 mm on the top surface, and 6.35 mm high, with spacing equal to element length were measured. A strong dependence on channel flow rate was found and only a weak dependence on channel height was apparent. The heat transfer coefficient was found to vary only 10% for channel height to module height ratio varying from 1.25 to 4.6.

Sparrow et al [8] used the naphthalene sublimation technique to estimate the Sherwood number and deduce Nusselt number for modules of 26.7 x 26.7 x 10 mm with 6.7 mm spacing within a channel height of 26.7 mm. Also the effect of channel height on the heat transfer was investigated.

Morris and Garimella [9] developed empirical correlations applicable for single-phase forced-convection heat transfer from inline arrays of three-dimensional heated elements placed in channels. These correlations can be applied to the cooling of miniaturized electronic components, computer chips, turbine-blade internal flow passages, and other heat transfer enhancement situations. The proposed correlations are based on data from 633 experiments that cover wide ranges of Reynolds numbers Re_L from 600 to 70000 and Prandtl numbers from 0.7 to 25.2 for different geometries and different array spacings. The data include those from a number of studies in the literature e.g. [10, 11].

Experiments were performed by Garimella and Schlitz [12] on an array of 15 rows and five columns of elements mounted on the bottom wall of the test section with the equal streamwise and spanwise spacings of 6.1 mm. A single heated element was located in the tenth row of the central column where the remaining elements of the array were unheated. The used elements were in the form similar to the flat packs ships and the heat source was constructed of copper and heated with a cartridge

heater installed into a copper stem underneath the element. The experiments were performed at element length-based Reynolds number ranging from 610 to 68580.

Molki et al [13] were carried out an experimental investigation to study heat transfer in the entrance region of an array of rectangular heated blocks. The focus of the work is on the entrance heat transfer coefficients and the associated thermal wake effects. The experiments were performed with air as the working fluid. The geometric parameters of the array were varied in the range identified with B/L=0.5, S/L=0.128-0.33, and H/L=0.128-1. The Reynolds number, based on the height above the blocks and the fluid mean velocity in the bypass channel, ranged from 3000 to 15,000. The adiabatic heat transfer coefficients and thermal wake effects are correlated for the entrance region.

Several investigators have attempted to increase heat transfer in a circuit board channel. Sparrow et al [14] studied the heat transfer enhancement effect of placing long barriers behind components. They used naphthalene sublimation technique to measure the effects of these barriers on the mass transfer coefficient and related this to the heat transfer coefficient. The barriers enhanced heat transfer, with the greatest effect (about a factor of two) in the second row downstream of the barrier. The barriers induced a pressure drop 10 to 150 times that for a barrier free row of components. Sparrow et al. [15] looked at the effectiveness of varying the number and height of the barriers. Again they saw large increases in the heat transfer coefficient, up to a factor of two, with the use of fence-like barriers. The heat transfer coefficient in the presence of multiple barriers correlates well with those for a single barrier. The barrier related pressure drop for a multiple barrier system was less than the corresponding multiple of the pressure drop for a single barrier up to barrier separations of six rows.

Chou and Lee [16] studied the possibility of reducing flow nonuniformities in large-scale integrated packages by vortex generators. Their test configuration consisted of two cubic integrated circuits mounted on a simulated printed circuit board. They attached a rectangular plate as a vortex generator to the front face of the downstream cube. They studied only the effects of the vortex generator on the upstream cube. It was found that the vortex generators reduced both the integrated circuit surface temperature and the temperature nonuniformities in the integrated circuit itself. The vortex generator had a critical height beyond which its effectiveness died off. Ratts et al. [17] conducted an experimental study on internal flow modulation induced by vortex shedding from cylinders perpendicular to an air flow and its effect on cooling an array of chips. They found enhancement effects up to 82 percent in the heat transfer coefficient for cylinders periodically positioned above the back edge of each row of chips. They studied the effects of cylinder position, diameter, length, and number of cylinders on the heat transfer enhancement.

The objectives of the present study are:
  i) to study the effect of fence height and spacing on the heat transfer and flow friction.
  ii) to provide electronic equipment designers with correlations for the thermal performance of electronic modules placed downstream of a fence.

2. EXPERIMENTAL APPARATUS

The experimental facility employed in the present investigation is an open-loop air flow circuit oriented horizontally as shown schematically in Fig.(1-a). It consists of a centrifugal air blower, an orifice flow meter, a main entrance duct, a test section and instrumentations to measure temperature, pressure drop, air flow rate and electrical power input. The details of the apparatus are depicted as follows:

(a) **Air blower:** A centrifugal type air blower of 150 mm inlet section diameter, 600 mm outer blade diameter and 110×110 mm square outlet section, driven by a motor of 3 hp capacity and 3000 rpm normal speed is used to supply the system with air at the required flow rate. The air flow rate is controlled by means of an intake gate. The air is sucked from the room and then blown into the test section via the converging section.

(b) **Test section:** The test section is 320 mm long and has a rectangular cross section of 160 mm by 40 mm which yields an aspect ratio of 4. The test section is made of a plexiglass rectangular duct having hydraulic diameter of 64 mm. A bakelite plate with dimensions of 320 mm length, 160 mm width and 10 mm thickness is used as a smooth adiabatic bottom surface for the test section. The test configuration is a heated rectangular block made of brass and measured 50 mm x 50 mm with 12.7 mm height. The heating element is mounted on a simulated printed circuit board (the bakelite plate) and affixed downstream of a fence. The fence is made of a plexiglass plate with 50 mm width (equal to the module width) and located upstream of the module as shown in Fig.(1-b). A nickel chromium heating tape of 5 mm width and 0.2 mm thickness is wrapped around a mica sheet and sandwiched between two electric insulation sheets. To minimize the heat losses, the bottom side of the heater is insulated with asbestos layers as shown in Fig.(1-c). Pre-calibrated thermocouples made of Chromel-Alumel wires (type K) of 0.2 mm diameter are embedded in grooves cut on the heating element surfaces to measure the temperatures of these surfaces. The heating block is positioned downstream of a fence at different spacing. The heating block height (B) to channel height (H) ratio is fixed at B/H=0.317. Three different values for the fence height (b) and four various values for the spacing between the fence and the block (S) are used as a testing parameters in such a manner that the ratio b/B=1, 1.5 and 2, and the ratio S/B=1, 2, 3, and 4. The working fluid is air and Reynolds number, based on the streamwise length of the heating element (L), ranged from 8000 to 40000.

(c) **Instrumentation:** A digital voltmeter of minimum division 0.1 Volt is used to measure the voltage drop on the electric heater. To measure the resistance of the electric heater, a digital multimeter (division, 1 Ohm) is used. The input power to the heater is regulated with a 5 kW Variac transformer. A compensated digital thermometer accurate to 0.1 °C is used to read the thermocouples output. The air flow rate is measured via a calibrated orifice meter connected to U-tube manometer. The pressure drop across the test section is measured by a digital micromanometer (accurate to±1 %). Also, a dry bulb thermometer (full-scale 50 ° C, and 0.2 °C division) is used to measure the room air temperature.

3. EXPERIMENTAL PROCEDURE AND DATA REDUCTION

Before the onset of a data run, the proper fence height and spacing is selected and then fixed upstream of the element in the test section. Then, to prevent air from leaking into the test section, all suspected joints are sealed by silicon rubber and thoroughly tested for leaks with aid of soap solution. Subsequently, the flow and power to the element heater are turned on and adjusted for predetermined values.

In a typical data run, the input power of the element heater is adjusted at such a level that the element heat flux remains constant. Once the heated element reaches steady state, about 90 minutes, the various parameters are recorded. These parameters are the element surface temperatures, inlet air temperature, pressure drop across the orifice meter, pressure drop across the test section, voltage and resistance of the heater.

The average convective heat transfer coefficient is evaluated from

$$h = \frac{q''}{T_s - T_f} \tag{1}$$

where, the heat flux q" is corrected for radiation and conduction losses as in Abdel-Moneim et al. [18].

The element length was found to be a good choice for the characteristic length in both the Reynolds and Nusselt numbers [9]. Therefore, the average Nusselt and Reynolds numbers are calculated as: $Nu = \frac{hL}{k}$, and $Re = \frac{VL}{\nu}$.

Moreover, the Fanning friction factor (F) was calculated in terms of the frictional pressure drop along the test section of the duct as:

$$F = \left(\frac{\Delta p}{2\rho V^2} \right) \frac{D}{\ell} \tag{2}$$

Air properties were based on the mean film temperature.

Uncertainty Analysis was performed to evaluate the error inherent in the present measurements. Percentages of uncertainties in the measurements of electric voltage, resistance, element surface area, surface temperature, mean film temperature, air flow velocity, and pressure drop were 0.3, 0.2, 0.5, 0.333, 0.666, 1.5, and 2.0, respectively. Therefore, mean uncertainties of Nusselt and Reynolds numbers and the Fanning friction factor were estimated as 2.0, 1.75 and 5.5 percent, respectively.

4. RESULTS AND DISCUSSIONS

To check the validity of the present measurements, a series of experiments were carried out for the module without fence within the range of Reynolds number from 8,000 to 40,000. The no fence results for both the average Nusselt number and the Fanning friction factor are shown in Fig (2). Significant differences in the average

Nusselt number and also in the Fanning friction factor were noticed when comparing the module results with those for the smooth duct. These differences were attributed to the structure of the turbulent velocity field around the module and the strong wall-shear stress associated with the vortex shedding. Figure (2-b) shows a sketch for the vortical flow structure around a module based on the visualization of Meinder et al. [19]. This flow pattern contains vortex tubes as a horseshoe vortex at the two sides of the module, top and side vortices and wake vortex at the trailing face of the module. Therefore, an expected similar vortical flow pattern may be existed around the present module and as a result the heat transfer performance was enhanced.

Also, good agreements were found when comparing the present heat transfer results for the case of no-fence with both the experimental data of Anderson and Moffat [20] and the correlations obtained by Wirtz and Dykshoorn [7] and Lehmann and Pembroke [11] as shown in Fig.(2-a). These better agreements give assurance and confidence in the present experimental data and make it possible to get the following correlations for the case of no-fence:

$$Nu_{N-F} = 0.0468\ Re^{0.8} \tag{3}$$

and

$$F_{N-F} = 0.292\ Re^{-0.283} \tag{4}$$

These correlations are valid with maximum deviations of 2% and 2.8%, respectively, for the case without fence within a range of Reynolds number from 8,000 to 40,000.

The main objective of the present work is to study the effect of the presence of a fence upstream of the module on the heat transfer and flow friction. The effects of the fence height (b) and spacing (S) were investigated. Three different fence-heights such that ratios of b/B=1.0, 1.5 and 2.0 at four different spacing such that ratios of S/B=1.0, 2.0, 3.0 and 4.0 were investigated within the range of Reynolds number from 8,000 to 40,000. The average Nusselt number versus Reynolds numbers for three different S/B ratios at different b/B ratios are shown in Fig.(3). Generally, higher values for the Nusselt numbers were found due to the presence of fence. This in fact confirms the establishment of heat transfer enhancement. This enhancement in the heat transfer is due to the separation-reattachment and recirculation flow pattern with secondary vortices shedding. The shear layer separates upstream of the leading face of the fence and reattaches downstream of the trailing face at reattachment length depends on both Reynolds number and the duct height. The fundamental interest is to find the fence-module configuration that can maximize the heat release from the module and prevent hot spots.

The enhancement in the heat transfer was significant at relatively low ratios of S/B and diminishes with the increase in the spacing due to the decay in the vortices strength. Also, the enhancement in the average Nusselt number decreases with the increase in Reynolds number as shown in Fig.(3-c). This is due to the fact that the increase in Reynolds number causes an enlargement in the recirculation region. At this region the trapped fluid temperature increases and therefore acting as a buffer layer and prevents beneficial heat release.

The present measurements for the pressure drop due to the presence of fence upstream of the module show that the Fanning friction factor is strongly dependent on Reynolds number and both the fence height and spacing while it is independent on the fence height at the higher S/B ratio as shown in Fig.(4).

The average Nusselt number for the module downstream of a fence was correlated with Reynolds number, b/B and S/B ratios and the following correlation was obtained:

$$Nu = 0.4056 \ Re^{0.6086} \left(\frac{b}{B}\right)^{0.1918} \left(\frac{S}{B}\right)^{-0.0008} \tag{5}$$

Also, the present data for Fanning friction factor were correlated as a function of Reynolds number and the ratios b/B and S/B and the following correlation was obtained:

$$F = 617 \ Re^{-0.9725} \left(\frac{b}{B}\right)^{0.6537} \left(\frac{S}{B}\right)^{0.3229} \tag{6}$$

The correlations Eq.(5, 6) are valid for a range of Reynolds number from 8,000 to 40,000 within ranges of (b/B= 1.0 to 2.0) and (S/B= 1.0 to 4.0) with maximum deviations of $\pm13\%$ and $\pm25\%$, respectively. Figure (5) shows the correlated values versus the experimental data for both the average Nusselt number and Fanning friction factor within the investigated ranges of the different parameters.

A performance evaluation criterion based on the condition of equal mass flow rate of the coolant was applied to evaluate the thermal performance benefits of using fence upstream of the module. This was accomplished by determining the ratio $(Nu/Nu_{N-F})/(F/F_{N-F})$ at the different investigated parameters. The ratio $(Nu/Nu_{N-F})/(F/F_{N-F})$ was then plotted against Reynolds number as shown in Fig.(6). It is obvious that the performance ratio decreases with the increase in either b/B or S/B ratio in general. Also, the fence with b/B=1 provides the best performance however it gives the lowest Nusselt number. This due to the lower flow friction associated with this fence configuration as shown in Fig.(4).

5. CONCLUSIONS

The following conclusions can be drawn based on the present study:

1- The results for the module without fence displayed some noticeable differences as compared to the smooth duct flow. These differences demonstrate that separation-reattachment flow patterns with vortical flow structure were existed.

2- The presence of fence upstream of the module enhances the heat transfer and increases the friction factor in general. This is due to the fact that secondary vortex tubes were existed at the module sides and these vortices have significant effects on both the heat transfer and flow friction.

3- A significant enhancement in the heat transfer was found by using fence at the higher b/B ratio and this enhancement decreases with the increase in either Reynolds number or the spacing between the fence and the module.

4- The Fanning friction factor is strongly dependent on Reynolds number and both the fence height and spacing while it is independent on the fence height at the higher S/B ratio.

5- A maximum Nusselt number enhancement ratio (Nu/Nu_{N-F}) of about 1.94 was obtained corresponding to a friction factor ratio (F/F_{N-F}) of about 6.62 at the condition (Re=8700, b/B=1.5 and S/B=2).

6- The fence with b/B=1 provides the best performance based on the criterion of equal mass flow rate.

REFERENCES

1- Buller, M.L. and Kilburn, R.F. "Evaluation of surface heat transfer coefficients for electronic module packages" In "Heat Transfer in Electronic Equipment", ASME, HTD-20, pp.25-28, New York, 1981.

2- Anderson, A. M. and Moffat, R. J. "Direct air cooling of electronic components: Reducing component temperatures by controlled thermal mixing", Trans. of ASME, Vol. 113, pp. 56-62, 1991.

3- Nakayama, W. "Thermal management of electronic equipment", A review of technology and research topics, Appl. Mech. Rev. 39, pp 1847-1868, 1986.

4- Moffat, R.J. and Ortega, A. "Direct air cooling of electronic components" In "Advances in Thermal Modeling of Electronic Components and Systems", pp. 129-282. Hemisphere, Washington, D. C., 1988.

5- Moffat, R. J., Arvizu, D. E. and Ortega, A. " Cooling electronic components: forced convection experiments with an air-cooled array" In "Heat Transfer in Electronic Equipment", ASME HTD-48, pp. 17-27, New York, 1985.

6- Piatt, J. D. "Heat transfer from an array of heated cylindrical elements on an adiabatic channel wall", M. S. Thesis, Naval Postgraduate School, Monterey, California, 1986.

7- Wirtz, R. A. and Dykshoorn, P. Heat transfer from arrays of flat packs in channel flow", Proc. Int. Electron. Packag. Soc. Conf., 4th, New York, pp. 318-326, 1984.

8- Sparrow, E.M., Yanezmoreno, A.A. and Otis, D.R., "Convective heat transfer response to height differences in an array of block-like electronic components", Int. J. Heat Mass Transfer, Vol. 27, pp. 469-473, 1984.

9- Morris, G.K. and Garimella, S.V. "Composite correlations for convective heat transfer from arrays of three-dimensional obstacles", Int. J. Heat Mass Transfer, Vol. 40, pp. 493-498, 1997.

10-Kang, S.S. "The thermal wake function for rectangular electronic modules", J. Electronic Packaging 116, pp. 55-59, 1994.

11- Lehmann, G.L. and Pembroke, J. "Forced convection air cooling of simulated low profile electronic components", J. Electronic Packaging 113, pp. 21-26, 1991.

12- Garimella, S.V. and Schlitz, D.J. "Heat transfer enhancement in narrow channels using two and three dimensional mixing devices", ASME, J. of heat transfer, Vol. 117, pp.590-596, 1995.

13- Molki, M., Faghri, M. and Ozbay, O. "A correlation for heat transfer and wake effect in the entrance region of an n-line array of rectangular blocks simulating electronic components", ASME, J. of heat transfer, Vol. 117, pp. 40-46, 1995.

14- Sparrow, E.M., Niethammer, J. E. and Chaboki, A. "Heat transfer and pressure drop characteristics of arrays of rectangular modules encountered in electronic equipment", Int. J. Heat Mass Transfer, Vol. 25, pp. 961-973, 1982.

15- Sparrow, E.M., Vemur, S.B. and Kadle, D.S. "Enhanced and local heat transfer, pressure drop, and flow visualization for arrays of block-like electronic components", Int. J. Heat Mass Transfer, Vol. 26, pp. 689-699, 1983.

16- Chou, J. and Lee, J. "Reducing flow non-uniformities in LSI packages by vortex generators", Proceedings of the international Symposium on Cooling Technologies for Electronic Equipment, Honolulu, HI, pp. 583-593, 1987.

17- Ratts, E., Amon, C. H., Mikic, B.B. and Patera, A.T. "Cooling enhancement of forced convection air cooled chip array through flow modulation induced by vortex-shedding cylinders in cross-flow", Proceedings of the international Symposium on Cooling Technologies for Electronic Equipment, Honolulu, HI, pp. 651-662, 1987.

18- Abdel-Moneim, S. A., Atwan, E. F. and El-Shamy, A. R., "Heat transfer and flow friction in a rectangular duct with repeated multiple V-ribs mounted on the bottom wall" Proceedings of the 12th Int. Mechanical Power Engg. Conference (IMPEC12) Vol. 2, pp. H11-H25, Oct. 30th – Nov. 1st, 2001.

19- Meinders, E. R. Hanjalic, K., Martinuzzi, R. J., "Experimental study of the local convection heat transfer from a wall-mounted cube in turbulent channel flow", Trans. of ASME, Vol. 121, pp. 564-573, 1999.

20- Anderson, A. M., and Moffat, R. J. "The adiabatic heat transfer coefficient and the superposition kernel function – I. Data from arrays of flat packs for different flow conditions", J. Electronic Packaging 114, pp. 14-21, 1992.
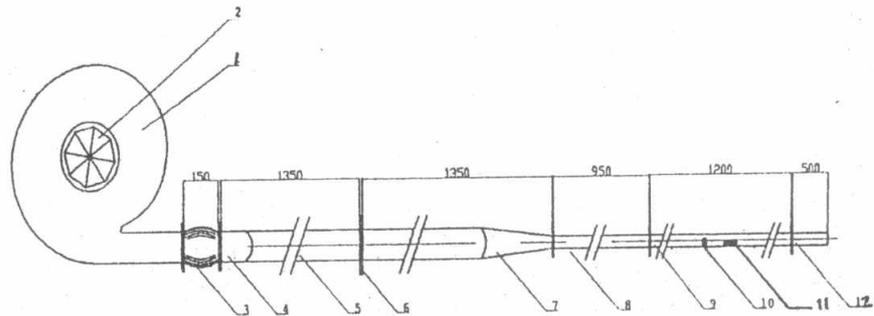
1- Air blower	2- Intake gate
3- Flexible joint	4- Conical section
5- Circular duct	6- Orifice meter
7- Circular/Rectangular converter	8- Rectangular duct
9- Test section	10- Fence
11- Heating element	12- Exit section

Dims in mms

(a): Test rig layout
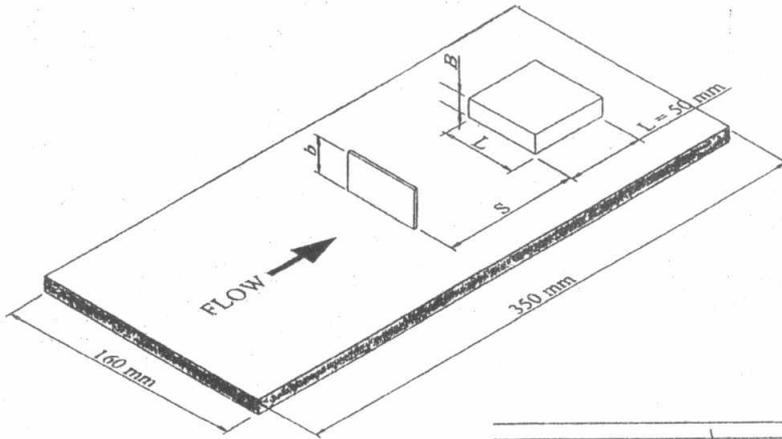
(b): Test section geometry
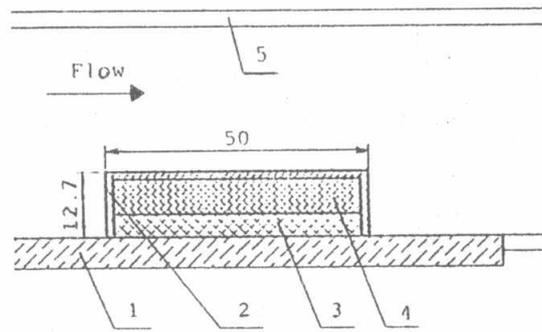
1- Bakelite plate
2- Brass element
3- Asbestos sheet
4- Electrical heater
5-Test duct

(c): Details of the heating element

Fig. (1): The Experimental Set-up

Nu
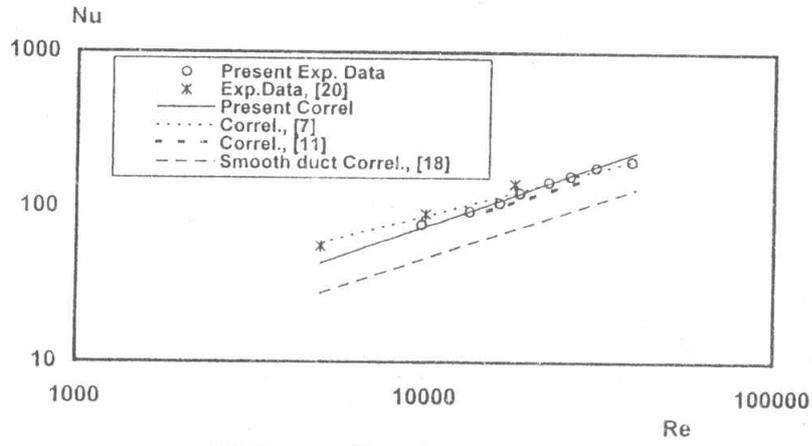
(a): Average Nusselt number
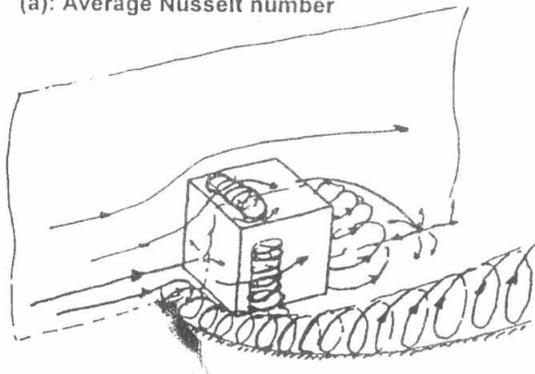
(b): Flow structure around a module in a turbulent channel flow:
Sketched based on the visualization of [19].
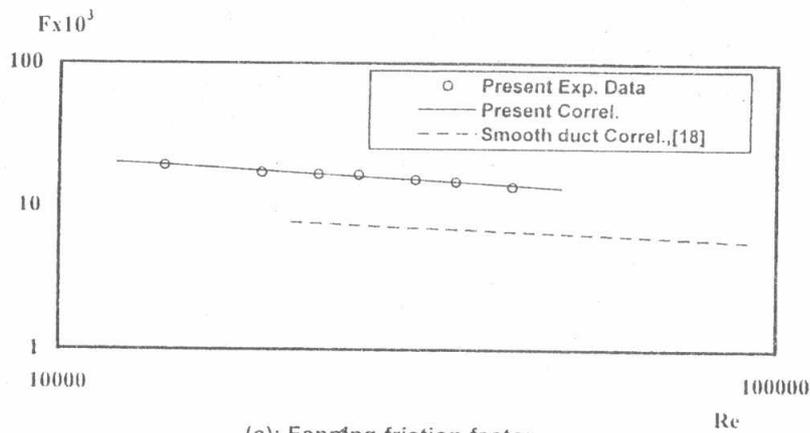
(c): Fanning friction factor

Fig.(2): Validation of the present experimental data for the case of No-Fence.
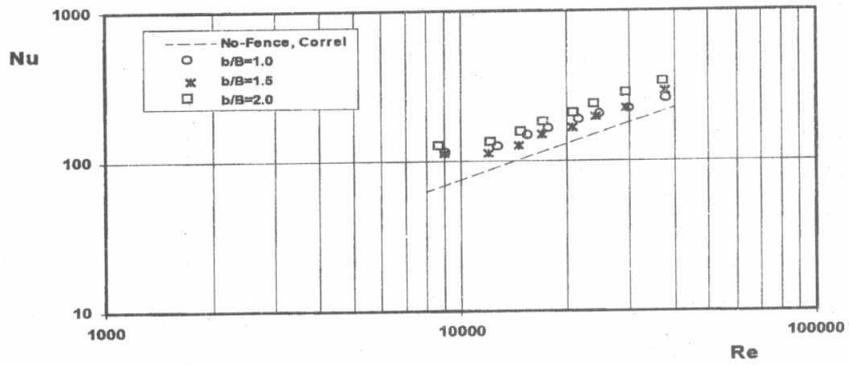
(a): S/B = 1.0
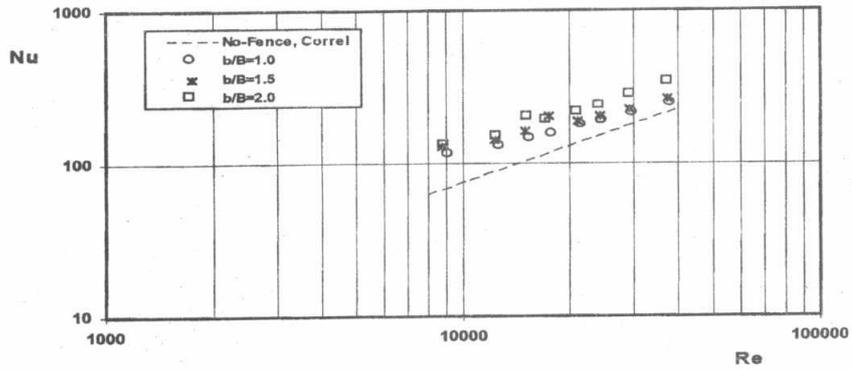
(b): S/B = 2.0
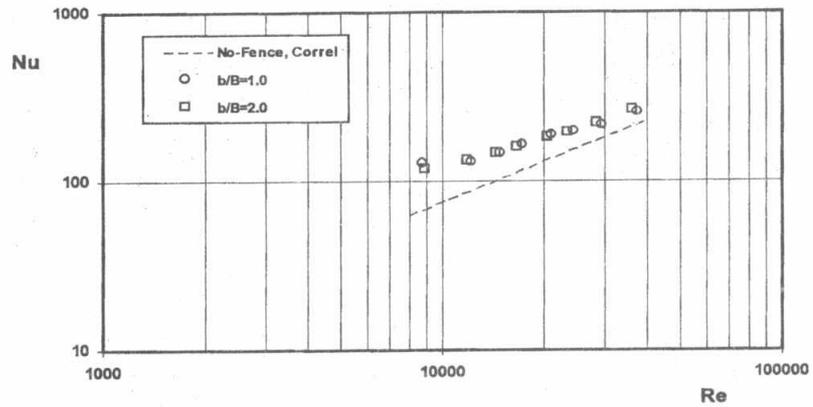
(c): S/B = 4.0

Fig.(3): The average Nusselt number versus Reynolds number for different configurations

(a): S/B = 1.0

(b): S/B = 2.0

(c): S/B = 4.0
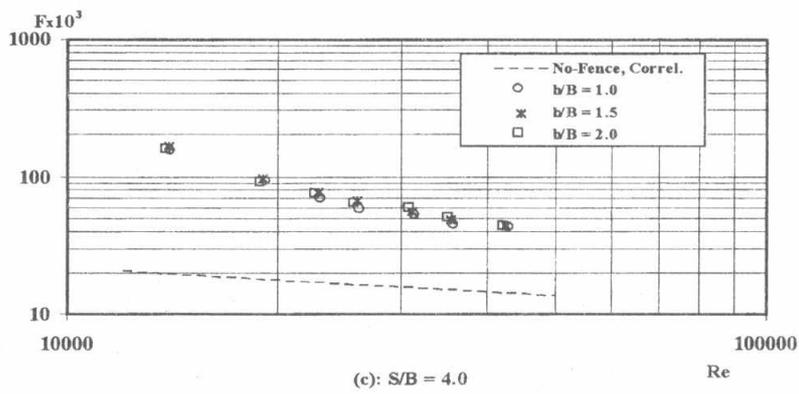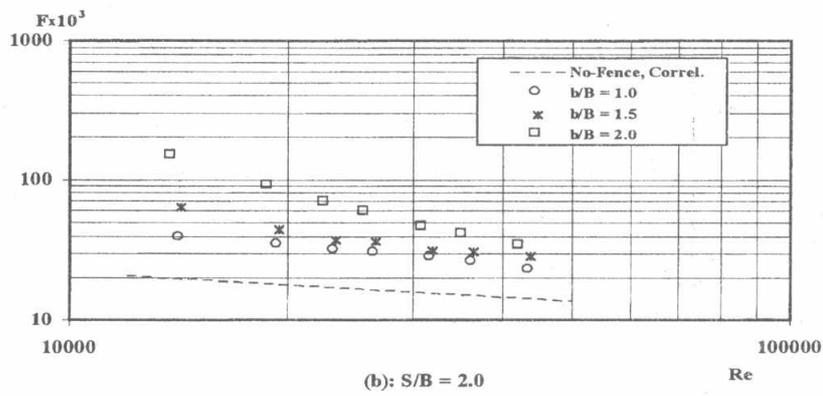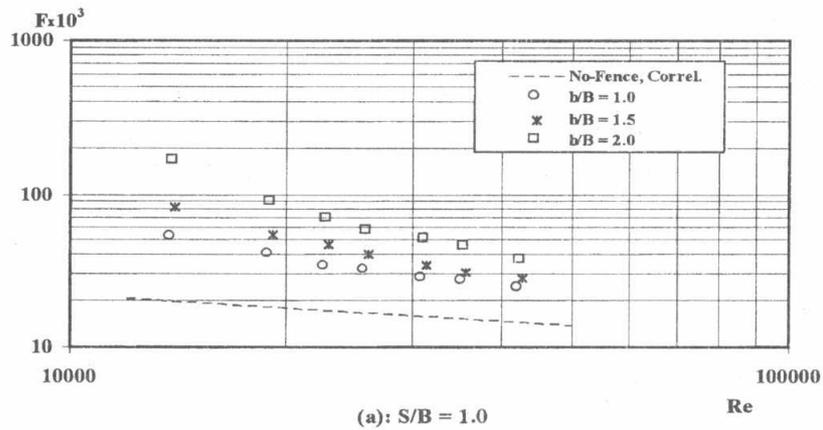
**Fig.(4): Fanning friction factor versus Reynolds number
for different configurations**
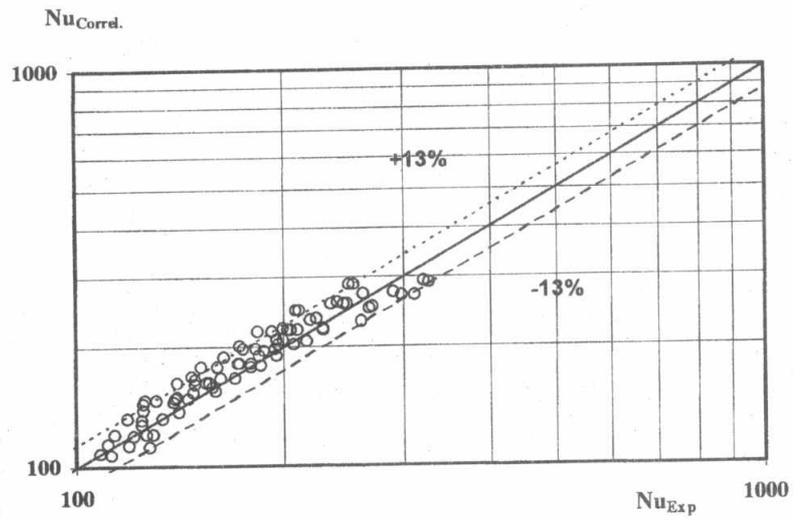
Nu$_{Correl.}$

(a): Nusselt number Correlation

F$_{Correl.}$ x10^3
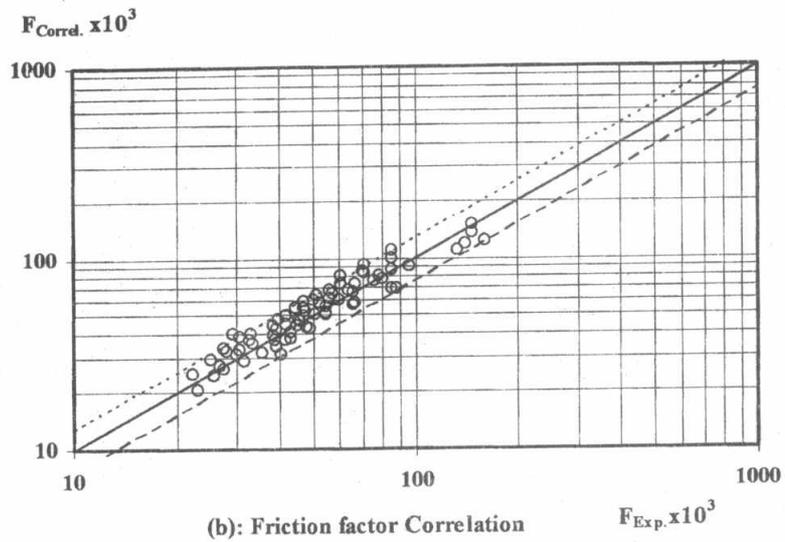
(b): Friction factor Correlation   F$_{Exp.}$x10^3

**Fig.(5): Experimental data correlations:
Correlated versus experimental data**

(a): S/B = 1.0

(b): S/B = 2.0

(c): S/B = 4.0
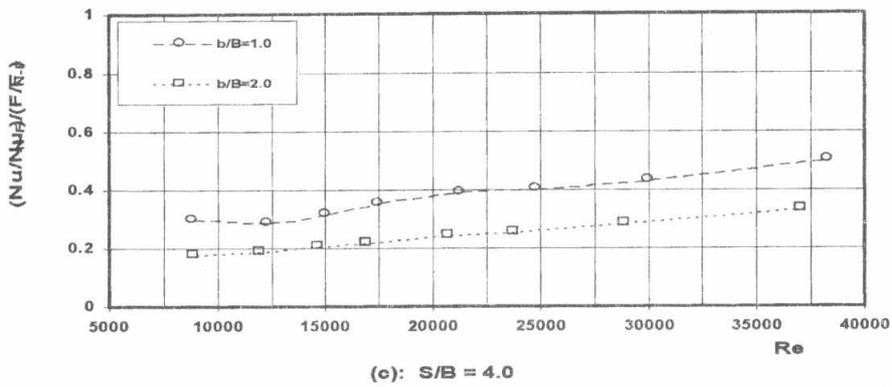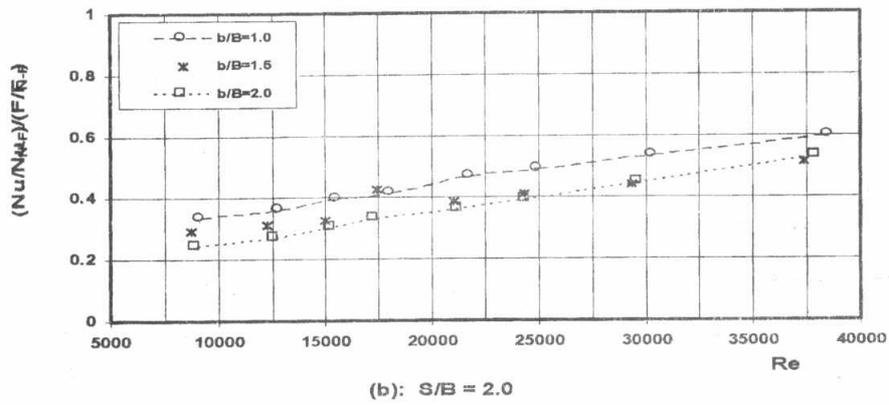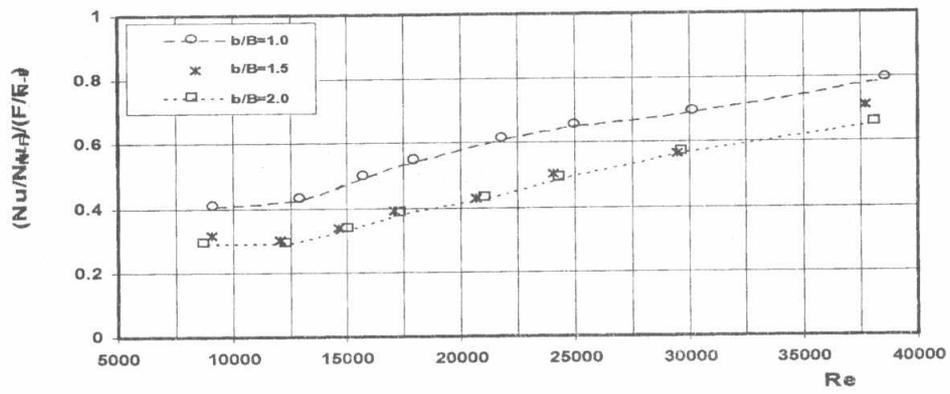
**Fig.(6): Thermal performance of a module downstream of a fence
at the condition of equal mass flow rate of the coolant**