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NUMERICAL SIMULATION OF CONJUGATE HEAT TRANSFER FROM MULTIPLE ELECTRONIC MODULE PACKAGES COOLED BY AIR

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ABSTRACT

This paper reports on the numerical simulation of conjugate heat transfer from multiple electronic module packages (45 x 45 x 2.4 mm) on a printed circuit board placed in a duct. The dimensions of the modules are the same as a single module package previously studied. In the series arrangement, two module packages are installed on the center of the printed circuit board along the airflow direction. In the parallel arrangement, two and/or four module packages are installed normal to the airflow direction. In the numerical simulations, the interval between the module packages was varied and three values were considered (45, 22.5 and 9 mm). The variation of the printed circuit board thermal conductivity was also considered and 0.3, 3 and 20 W/m/K were used with the mean velocity in the duct also at three different values (0.33, 0.67 and 1 m/s).

In order to derive a non-dimensional correlation from the numerical results, the concept of the effective heat transfer area previously used for a single module package was used for the multiple module packages. For the series arrangement, the effects of the interval on the effective heat transfer area are relatively low, and the numerical results can be summarized with the same correlation obtained from the single module package.

On the other hand, the effective heat transfer area for the parallel arrangement is strongly affected by the parallel interval and the thermal conductivity of printed circuit board. When the interval increases, the temperature of the module packages greatly reduces as the thermal conductivity of the printed circuit board increases.

Keywords: *Conjugate Heat Transfer, Effective Heat Transfer Area, Multiple Chips, CFD*

INTRODUCTION

As the electronic products are rapidly developing with denser and more powerful circuits, a higher level of cooling performance is required. Many cooling methods have been proposed, including conjugate heat transfer methods. The conjugate heat transfer problem has recently received considerable attention from heat transfer researchers, especially from those who are performing CFD computations. Although numerous CFD codes are extensively used in the field of electronic packaging, validation of CFD simulations is very important for both designers and researchers. This requires accurate experimental data to be used for benchmarking. Some attempts have been made as evidenced in Chang et al. [1987] who studied experimentally the heat transfer from surface mounted components in an air channel. Nakayama and Park [1996] studied the conjugate heat transfer from a single surface-mounted block cooled by air flow. Hong et al. [2000] further reported their new measurement results in a similar experiment. However, a comparison of their data with those of Nakayama shows a large scatter. Such differences need to be understood before researchers and designers working in the field of thermal design and management for electronic equipment can use the data for design purposes.

In our previous papers (Yoshino et al, [2002], Fujii et al, [2002]), a unique correlation between Nusselt and Reynolds numbers for the single electronic module package was derived by introducing an effective heat transfer area. However, the heat transfer characteristics of multiple electronic module packages were not considered. Here, the conjugate heat transfer characteristics for the multiple module packages have been numerically studied and compared with the non-dimensional correlation obtained for the single module package.

NOMENCLATURE

A = surface area, m^2
D_H = hydraulic diameter of duct, m
h = heat transfer coefficient, W/m^2K
L_{ps} = side-length of the package substrate, m
L = distance between model packages, m
Nu = Nusselt number

Re = Reynolds number
Q = total heat generation rate, W
U = velocity, m/s
T = temperature, K
t = thickness, m
P = pressure, Pa
λ = thermal conductivity, W/mK
ν = kinematic viscosity, m^2/s

Subscripts
eff = effective
cond= conduction
conv= convection
o = room
a = air
p = printed circuit board, PCB
h = heater
hs = heat spreader
ps = package substrate

SINGLE MODULE PACKAGE

Figure 1 shows the geometry of the experimental setup for single module package. The PCB with electronic module packages is placed in a duct of 235 mm length, 200 mm width, and 10 mm height.

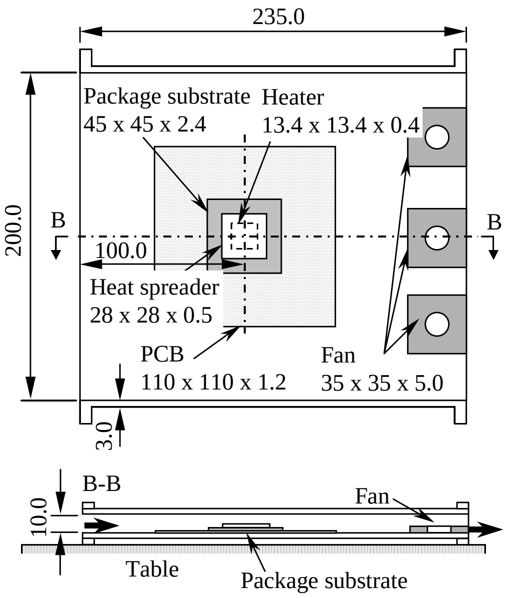


Fig. 1 Experimental setup for single module package

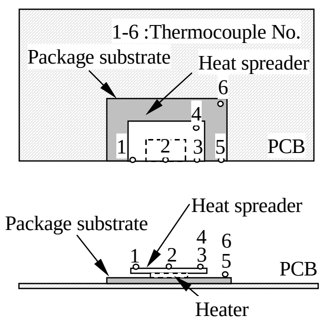
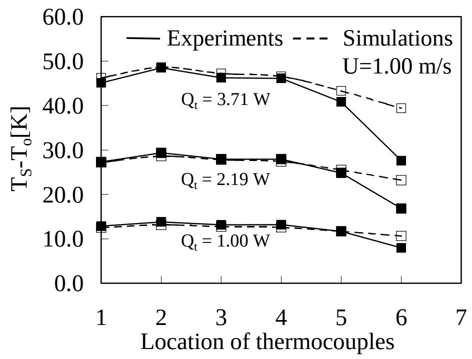


Fig. 2 Location of six thermocouples

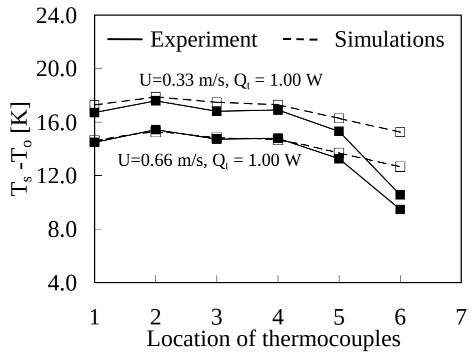
The PCB has a thermal conductivity of λ_p=0.3 W/m/K and is 110 mm square and 1.2mm thick. It is placed at the center of the duct wall about 100 mm away from the inlet. The module package consists of a heat spreader (λ_{hs}=398 W/m/K), a heater (λ_h=45 W/m/K) and a package substrate (λ_{ps}=16 W/m/K). The heat spreader is 28 mm square, and 0.5 mm thick, and is attached to the heater which is 13.4 mm square and 0.4 mm thick. The heat spreader and the heater are tightly adhered to the package substrate which is 45 mm square and 2.4 mm thick. The package substrate is connected with ball grid array to the center of the PCB. Three fans are placed on the downstream of the duct, and their dimension is 35 mm square and 5.0 mm high. The power supply line to each fan can be set ON or OFF, respectively. The measured maximum average air velocity was of the order of 1.0 m/s with all three fans turned on. The heater is energized with a DC power supply in a variable voltage range and the maximum power dissipation of the heater is about 5 watts. The measured average velocities in duct are 0.33 m/s, 0.67 m/s and 1.00 m/s.

Six T-type thermocouples with 50µm diameter are installed on the module package to measure the surface temperature (Figure 2). The thermocouples numbered from 1 to 4 are located on the heat spreader surface, and the other two (5 and 6), are located on the package substrate surface. A built-in diode is installed in the center of the heater to measure the junction temperature.

The computational setup has been developed based on the corresponding experimental setup described above. The dimensions and thermophysical properties of the duct and module package for the present simulations are the same as the experimental ones. On the other hand, the complicated internal



(a) Three fans in operation



(b) Single and two fans in operation

Fig. 3 Temperature profiles

structures of the package substrate and PCB were simplified and they were replaced with two blocks with uniform but different thermal conductivities to save computational time. A volumetric uniform heat generation was assumed in the heater. Natural convection boundary conditions are employed at the outer surfaces of the duct. The heat transfer coefficient was estimated to be 7.5 $W/m^2/K$ by fitting the surface temperatures obtained numerically with those measured for various heating rates. A uniform pressure and uniform velocity are used at the inlet and the outlet of the duct, respectively. The simulations have been performed with the commercial CFD code 'CFdesign'.

To validate the numerical results, the simulated temperature profiles have been compared with the experimental results for a single module package. Figure 3(a) shows a comparison of temperature profiles at U=1 m/s, for three heat dissipation rates. The temperature profiles obtained from simulations agree well with the measured ones. The numerical results for the other two operation modes are also in good agreement with the experimental data as shown in Figure 3(b), where the heat dissipation rate is kept at 1 W. The results are averaged values of those obtained for the different fan operation modes. These results have validated the CFD code it can therefore be used for the conjugate heat transfer problem considered here.

EFFECTIVE HEAT TRANSFER AREA

Figures 4(a) and (b) show the two basic models for evaluation of heat flux. The conventional heat flux q_{ref} shown in Figure 4(a) is defined under the assumption that the volumetric heat generated from the heat source is completely transferred to air only from the heat source surface area A_{ref}. Taking into account the heat conduction leads to the second definition of the heat flux q shown in Figure 4(b). The heat transferred from the area A_{ref} should be the net heat rate, i.e. heat conduction through the wall is subtracted from the total heat generation. Therefore, the net heat flux can be defined as follows.

$$q = (Q - Q_{cond})/A_{ref} = Q_{conv}/A_{ref} \qquad (1)$$

Here, Q is the total heat generated in a heat source, Q_{cond} is the heat conducted through the surrounding wall, and Q_{conv} is the convective heat actually transferred to air from the heat spreader surface, which can only be obtained from simulation. The effective heat transfer area is then defined as follows.

$$A_{eff} = Q/q = A_{ref}\ q_{ref}/q \qquad (2)$$

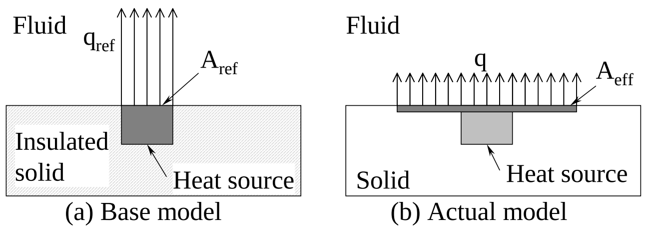


Fig. 4 Concept of effective heat transfer area

NON-DIMENSIONAL CORRELATION

Single module pacakge

Using the effective surface area, the heat transfer coefficient of the single module package is defined as

$$h = Q_{net}/A_{eff}/(T_{max}-T_o) \qquad (3)$$

Where Q_{net} is the net heat transfer rate ($Q-Q_{loss}$) and Q is the total heat dissipated, Q_{loss} is the heat loss convected away from the outer walls of the duct, A_{ref} is the surface area of the heat spreader, and T_{max} represents the maximum temperature at the heat spreader surface.

Therefore, the average Nusselt number can be defined with $A_{eff}^{0.5}$ as the reference length. The definition of the Nusselt and Reynolds numbers are defined as follows.

$$Nu = h\ A_{eff}^{0.5}\ /\lambda_a \qquad (4)$$

$$Re = U\ D_H/\nu_a \qquad (5)$$

Figure 5 shows the relationship between the effective heat transfer area and the thermal conductivity ratio, λ_p/λ_a. For relatively lower thermal conductivity ratios, the non-dimensional heat transfer area increases with Reynolds number. This tendency disappears in the region of higher thermal conductivity ratio. For λ_p/λ_a =780, the effective heat transfer area is almost independent of the Reynolds number.

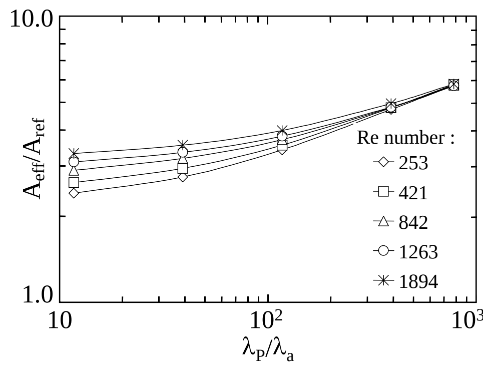


Fig.5 Relationship between A_{eff}/A_{ref} and λ_p/λ_a

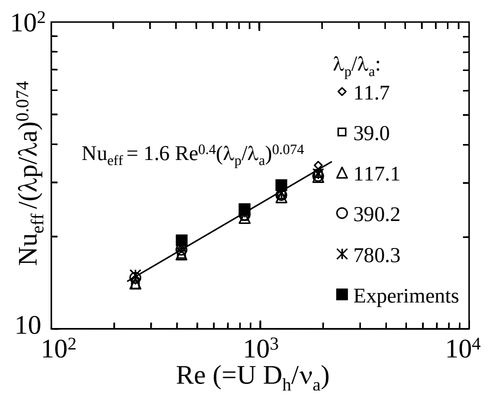


Fig.6 Nu vs Re for single module package

Figure 6 shows the relationship between Nusselt and Reynolds numbers. The numerical results can be expressed by a unique non-dimensional correlation introducing another parameter, the thermal conductivity ratio.

$$Nu = 1.6\ Re^{0.4}\ (\lambda_p/\lambda_a)^{0.074} \tag{6}$$

Multiple module packages

The computational setup for multiple module packages have been considered based on the corresponding single module package. Figure 7 shows the three types of models for the multiple module packages. Figure 7(a) shows the series arrangement, where two module packages are installed on the center of the printed circuit board along the airflow direction. Figures 7(b) and (c) show the parallel arrangement, where two and four module packages are installed normal to the airflow direction, respectively. The side view of the duct is shown in Figure 7(d).

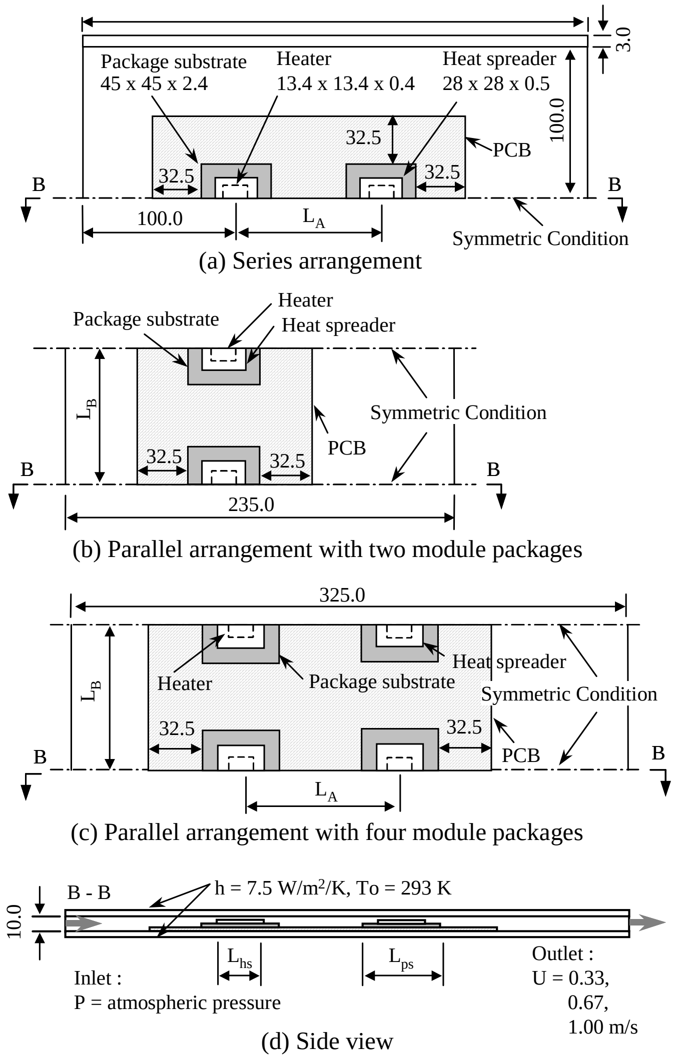


Fig. 7 Computational setup for multiple module packages

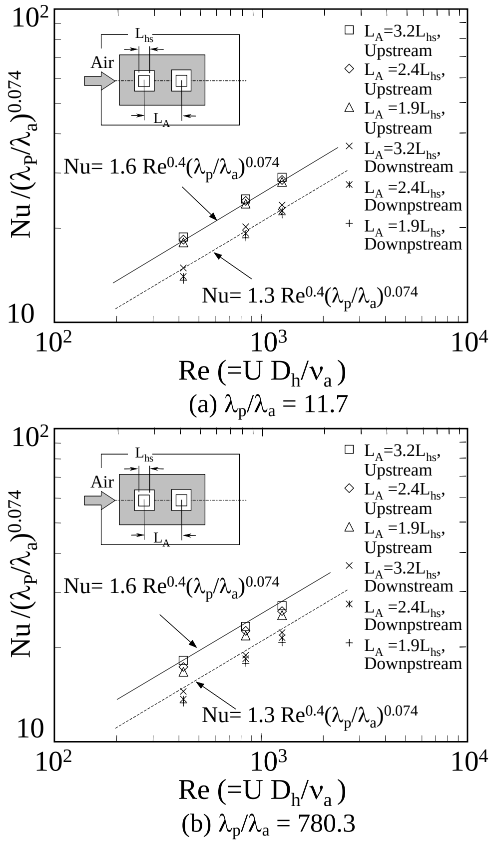


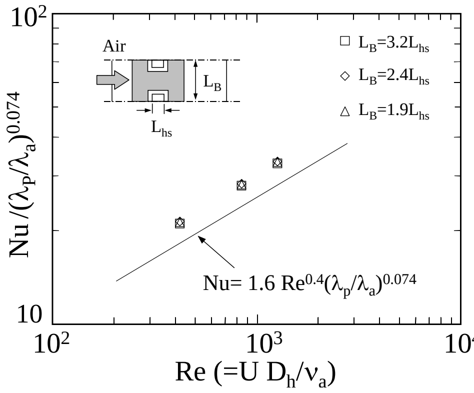
Fig. 8 Nusselt number for various arrangements

In the numerical simulations, the distance L_A and L_B between the module packages was varied and the three values examined were 90.0 ($3.2L_{hs}$), 67.5 ($2.4L_{hs}$) and 54.0 mm ($1.9L_{hs}$). The thermal conductivity of the PCB λ_p, was also varied (0.3, 3 and 20 W/m/K) were simulated. Three different air mean velocities (U=0.33, 0.67 and 1m/s) were considered.

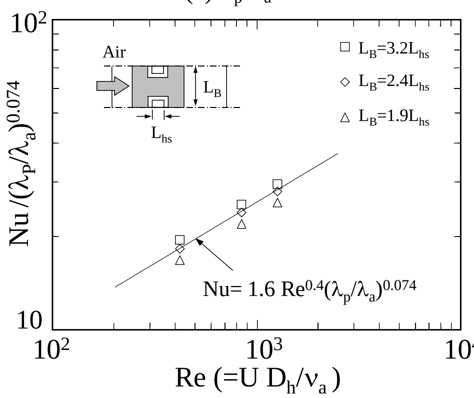
Figures 8(a) and (b) show the relationship between Nu and Re of the series arrangement for λ_p/λ_a =11.7 and 780.3. The heat transfer characteristic of the upstream package shown by the solid line is almost the same as that of a single module package, while that of the second package shown by the dashed line is about 20% lower, due to the thermal wake effect. For the series arrangement, the effect of the interspacing is relatively small and the single module package correlation can be used.

Figures 9(a) and (b) show the relationship between Nu and Re for the parallel arrangement with two module packages and thermal conductivity ratios of λ_p/λ_a =11.7 and 780.3, respectively. For λ_p/λ_a =11.7 the Nusselt number is about 20% higher than that of the single module package. On the other hand, for λ_p/λ_a =780.3 the results cannot be represented by a unique non-dimensional correlation. Here, the Nusselt number decrease as the distance between two modules decreases.

Figures 10(a) and (b) show the relationship between Nu and Re for the parallel arrangement with four module packages and λ_p/λ_a =11.7 and 780.3, respectively. For λ_p/λ_a =11.7 for both the upstream and downstream module packages a unique non-dimensional correlation is observed, which is about 15 % higher than that for the series arrangement.

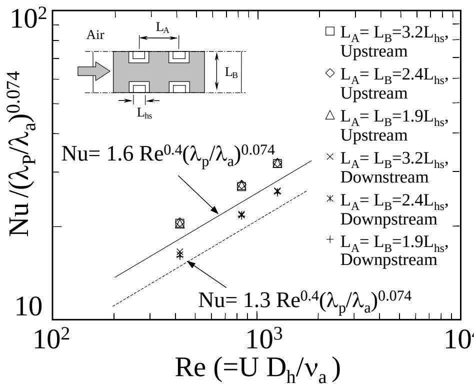


(a) λ_p/λ_a = 11.7

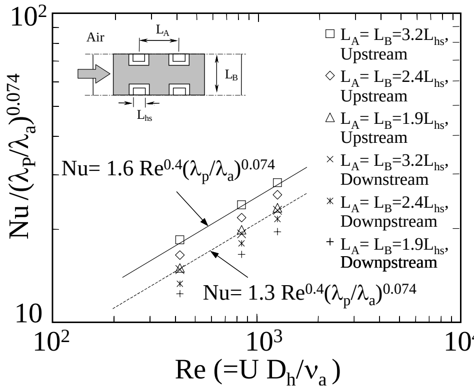


(b) λ_p/λ_a = 780.3

Fig. 9 Nusselt number for various arrangements



(a) λ_p/λ_a = 11.7



(b) λ_p/λ_a = 780.3

Fig. 10 Nu vs Re for various module arrangements

On the other hand, for λ_p/λ_a =780.3 the results can not be represented by a unique non-dimensional correlation. Here, the Nusselt number decreases as the parallel interval decreases.

The heat transfer behavior is further examined by considering the relationship between A_{eff}/A_P and the Biot number which is define as

$$Bi = h\,A_{eff} / t_p / \lambda_p \tag{7}$$

Where t_p is the thickness of the PCB. Figures 12 and 13 show the relationship between A_{eff}/A_P and Bi for the single module package, and the parallel arrangement with two module packages, respectively.

As shown in Fig. 12, the effective heat transfer area for the single module package increases with 1/Bi and reaches up to 40% of the total surface area of the PCB. On the other hand, the effective heat transfer area for the parallel arrangement is relatively larger than that of the single module one as shown in Fig. 13. When 1/Bi is greater than 0.2, the Nusselt number cannot be represented by a unique non-dimensional correlation (similar to Figure 9(b)). The effective heat transfer area in these cases should be considered to be the same as the PCB surface area because of the high values of 1/Bi and A_{eff}/A_P.

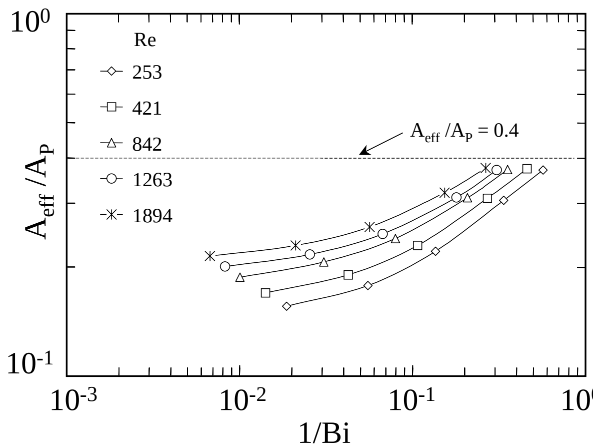


Fig.12 A_{eff}/A_P vs 1/Bi for single module package

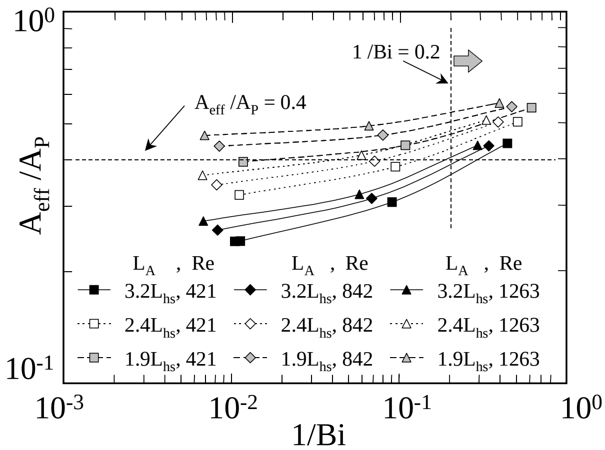


Fig. 13 A_{eff}/A_P vs 1/Bi for parallel arrangement with two module packages

Therefore, the average Nusselt number can be defined with $A_p^{0.5}$ as the reference length by

$$Nu = h^* A_p^{0.5} / \lambda_a \quad (8)$$

Where the heat transfer coefficient h^* is defined as

$$h^* = Q_{net}/A_P/(T_{max}-T_o) \quad (9)$$

Figure 14 shows the relationship between Nusselt number defined by Eq. (8) and Reynolds number defined by Eq. (5). This figure corresponds to the results shown in Fig. 9(b). It is noted that the numerical results can be expressed by a unique non-dimensional correlation by substituting the whole PCB surface area as the effective heat transfer area.

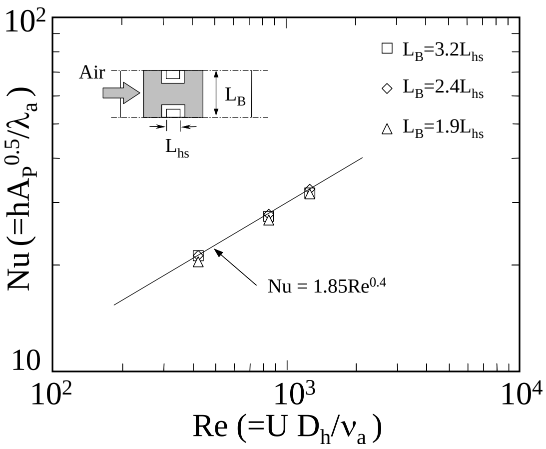


Fig. 14 Nu vs Re (λ_p/λ_a = 780.3)

CONCLUSIONS

Using the concept of effective heat transfer area, a unique non-dimensional correlation is proposed, which can predict the maximum temperature for the single module package, the series arrangement and the parallel arrangement for a low thermal conductivity PCB for various module package distance. However, for the parallel arrangement and high thermal conductivity of PCB when the distance between modules is small the whole PCB surface area should be used as the effective heat transfer area. Further studies should be done for the correlation in the range of 1/Bi from $1.1x10^{-2}$ to 0.2.

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