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Natural convection from a variable tilt angle duct with a built in electric module

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Abstract- The present work represents a parametric study with experimental and numerical work to study the natural convection heat transfer from a cube which represent an electric module fixed in a relatively long duct tilted at different angles of $\theta = 0°$ (horizontal), $\theta = 30°$, $\theta = 45°$ and $\theta = 90°$. Module side length (L=30mm) fixed at different locations of (X/L=5,10 and 15) with different input power of (0.147, 0.51, 0.96, 1.65, 2.34 and 3.075) Watt, which represent a heat flux of (32.67, 113.33, 213.33, 366.67, 520 and 683.33) W/m^2 respectively.

The CFD numerical method is used to verify the flow and thermal field around the cube during operation in natural mode. A commercial code Fluent is used for this purpose.

Results show that the most effecting parameter during natural convection modes is the power input which leads to enhance the heat transfer due to buoyancy effect which increase the induced velocity giving a chance for the flow plums to absorb more heat from the cube. The study shows that a little effect is recognized for the module position and the tilt angle.

Nemclature

A_s: Surface area of cube (m^2), CP: Air specific heat capacity (J/Kg.°C), D_h: Hydraulic diameter (m), G_r: Grashof number, g: Gravitational acceleration (m/s^2), h: Convection heat transfer coefficient (W/m^2.K), I: Electric current (Amp), K_a: Thermal conductivity of air (W/m.K), L: Length of cube (m), Nu: Nusselt number, Pr: Prandtl number ν/α, q: Input power (W), Ra: Rayleigh number $\frac{g\beta\Delta T L^3}{\nu\alpha}$, T_s: surface temperature (K). T_∞: Air temperature (K), t: Time (s), V: Electric voltage (V), β : Thermal expansion coefficient ($1/K$), ϑ : Kinematics viscosity of air (m^2/s).

Index Terms- Convection from a cube, Duct heat transfer with tilt angle, Numerical solution

I. Introduction

Advanced very large-scale integration (VLSI) technology has resulted in significant improvements in the performance of electronic systems in the past decades. With the trend toward higher circuit density and faster operation speed, however, there is a steady increase in the dissipative heat flux at the components, modules, and system levels. It has been shown that most operation parameters of an electronic components are strongly affected by its temperature as well as its immediate thermal environment. This leads to an increasing demand for highly efficient electronic cooling technologies to meet this demand, various electronic cooling schemes have been developed [1].

Convection can be divided broadly into two types forced convection and free (or natural) convection. When flow is generated by buoyancy force during heating or cooling a fluid, it is called free or natural convection [2]. Natural convection is of a great importance in many industrial applications. It is the foundation in modern electronics industry, materials processing, flow and heat transfer in solar ponds, float glass production, metal casting and food processing, etc.

In common use it is metal object brought in to contact with an electronic component's hot surface- though in most cases, a thin thermal interface material mediates between the two surfaces. Microprocessors and power handling semiconductors are examples of electronics that need a heat sink to reduce their temperature through increased thermal mass and heat dissipations (primarily by conduction and convection and to a lesser extent by radiation)[3].

II. Literature review

Aung et al. (1972),(1973) [4,5], performed extensive measurements for in-line and staggered card arrays mounted in an electronic cabinet with relatively smooth wire wrapped cards and with discrete, protruding components. Their model is an array of six vertically stacked cards in the interior of the cabinet. They found that, a reasonable agreement was apparent for the smooth channel results, despite the presence of flow obstruction from the card carries and surface components. Also in particular, the maximum wall temperatures were well predicted for closely spaced boards.

Ortega and Moffat (1985),(1986),(1986),(1986) [6-9], performed a series of experiments on sparse heated arrays of cubical elements located on an insulated vertical plate, with and without an opposing shrouding plate. The length is L=35 cm with sealed lateral. They measured heat transfer coefficient based on ambient temperature difference, and averaged across a horizontal row of elements, at each vertical position in the array. They found that in the first six rows of element, the heat transfer was enhanced by the induced forced flow, and the enhancement increase for the narrowest channel and the row of elements, at the lowest position. This findings are agree with that of Sparrow et at.(1982) [10]. Also they found that the heat transfer was degraded beyond row six because the beneficial effects of the chimney flow did not offset the increased temperature of the fluid. For large channel spacing, the heat transfer in the array was characterized by a complex plume-boundary-layer flow that resulted in a uniform temperature throughout the array. Also the array temperature distribution depended on element spacing. For the narrowest channel spacing, it was found that fully developed conditions were achieved within a few rows and that the local heat transfer from an element was dictated by the buoyancy-induced channel forced convection.

O.Zeitoun and Mohamed Ali (2006) [11], numerically investigated the natural convection around isothermal horizontal rectangular ducts, with two-dimensional laminar natural-convection heat transfer in air around horizontal ducts with rectangular and square cross sections. Different aspect ratios are used for wide range of Rayleigh numbers. Results are presented in the form of streamlines and isothermal plots around the circumference of the ducts. They used the finite-element technique. Temperature and velocity profiles are obtained near each surface of the ducts. Reverse flow and circulations are observed at high aspect ratios. Heat transfer data are generated and presented in terms of Nusselt number versus Rayleigh number for different aspect ratios. They used a correlation covering the aspect ratios in dimensionless form of Nusselt number, Rayleigh number, and aspect ratio.

III. EXPERIMENTAL APPARATUS

The test rig and the test procedure which are carried out in order to investigate the thermal behavior of a single module (30mm) side length to be fixed in different positions of a rectangular duct made of transparent plexiglass. The module is fixed at the base of the duct (lower side of the duct). The duct is designed so as it can be tilted at different angles of $\theta = 0°$ (horizontal), $\theta = 30°$, $\theta = 45°$ and $\theta = 90°$ (vertical). A heater capsule is inserted inside the module (electric resistance=201.67Ω) so that a constant temperature, constant heat flux is the mode at which the thermal behavior investigated for the system.

A. THE AIR DUCT

The air duct is made of plexiglass, constructed of two parts. The upper part is designed with a curved entrance, its cross section of concave downward rectangular shape. The curved entrance is shaped so as to ensure a smooth entrance of air through the duct when assembled with the lower portion which is constructed in the same way, but with a flat shape with edges of (4cm) around its circumference. The upper and lower portions of the duct are of the same length, which is designed to be equal to 25 times the cube side length (i.e. 25L) show in figure 1.

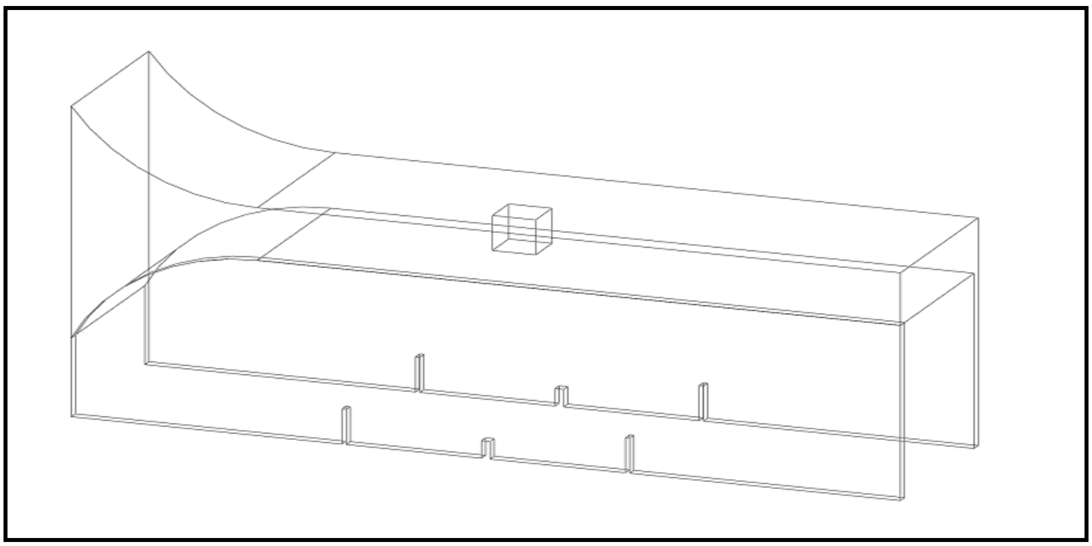


Fig 1. Test section

B. The Electric Module

The cubic shape electric module is made of Duralamine alloy (Al:90%, K=177W/m.K). Its side length is of (30mm) and its wall is thick enough so that the heat is transferred in constant temperature, constant heat flux due to high thermal conductivity of its material. The module is painted with a black color to get the best thermal image using infrared camera used to measure its temperature to ensure uniform temperature of its surface. Three holes are made at the lower of the module. At the lower surface center a (8mm) screwed hole is made to ensure module fixation to the lower duct side and another cavity (10mm) diameter flushed with an epoxy resin to ensure preventing electric conductivity to the module inner surface. A special capsule heater of (201.67Ω) resistance is inserted to the hole and the rest of the hole is filled with Zinc, epoxy to ensure good conductance to the module walls so that all heat input is to be transferred to the module surface then to the surrounding air during experimental operation of the apparatus. Other small hole (1mm) in diameter is used to measure the module temperature using a calibrated thermocouple of K (Komel-Alumel) type.

The lower surface of the module is insulated using a Teflon layer of (1mm) thickness (K=0.25W/m.K). This is done by using the layer of (30mm) Teflon square size glued with a special epoxy resin. This layer is machined to retain the thermocouple, the screw and the electric terminal of the heating module.

C. Measurement Devices

The measurement devices are used in this study as DC power supply to supply the input voltage and the current to the heater, Power Controller to control the input voltage of the heater, Temperature Recorder to measure the temperature and Thermal Imager.

IV. DATA REDUCTION

The total heat that is generated by the heater can be calculated as:

$$Q_{gen} = V * I \quad (1)$$

Then to calculate the heat transfer coefficient:-

$$q = h.A_s.(T_s - T_\infty) \quad (2)$$

The Rayleigh number is defined as the ratio between the buoyancy force and the viscous force:

$$Ra = \frac{\beta.g.(T_s - T_\infty).L^3}{\vartheta_{air}^2}.Pr \quad (3)$$

To calculate the Nusselt number:-

$$Nu = \frac{h.L}{K_{air}} \quad (4)$$

V. RESULTS AND DISCUSIONS

A. EXPERIMENTAL RESULTS

The present work is to study the behavior of cooling rate of an electric module in the shape of a cube shape put in a long duct to be tilted at different angles. The system is operate in the natural convection mode. The Nu number calculated on the basis of side length of the cube (L).

Figure (2) represents the variation of Nusselt number with Ra number in the mode of natural convection. Different variable are tested in addition to the Ra number, the tilt angle and module position. Results show that Nusselt number always increases with Rayleigh number (due to higher temperature difference which give bigger density change) leading to higher velocities of convection plums which transfers higher heat rates. Figure (2) also show that an asymptotic level is reached as Rayleigh number is increased. For figure (2 a, c) it can be concluded that the tilt angle does not change Nussult number values effectively specially for (X/L=10).

Figure (3) represent Nusselt number variation with different position of the module during natural convection operation. The figures declares that Nusselt number increases as the power supplied increases due to fact that the temperature rise of the module increases relative to ambient temperature causing higher induced velocity of the air leading to higher heat transfer coefficient. It can be concluded that module position (X/L) is with little effect on Nusselt number values.

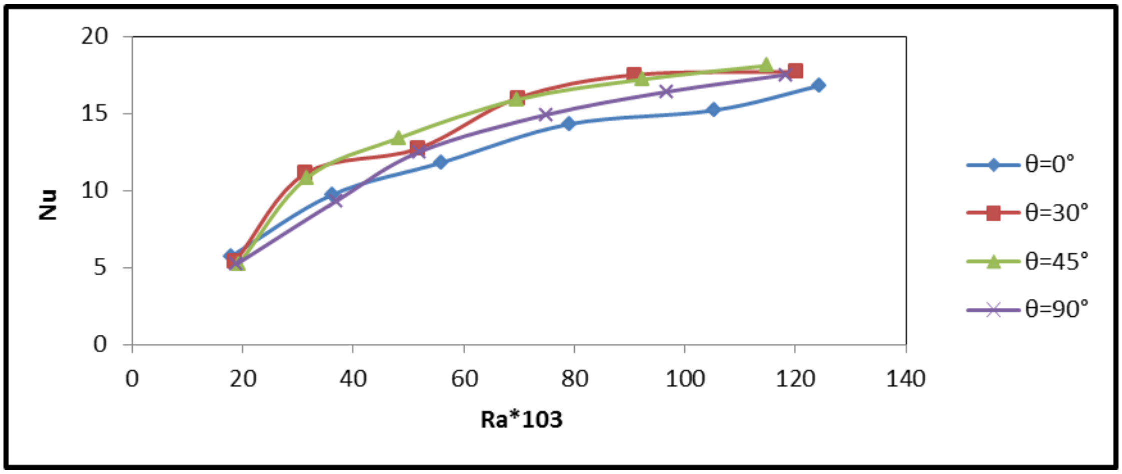
B. NUMERICAL RESULTS

The numerical part of this work uses a finite volume 3D, CFD solution method. The mesh used of size 0.003. the laminar method is used to estimate energy dissipation through the induced upward moving plumes which is generated due to natural convection of heat dissipation of cube. The CFD solution used to find the flow and thermal filed generated due to heat transfer by natural convection. The results of this part of the study is used to examine the effect of different variables as the input power and cube position for $\theta = 0° \; and \; \theta = 90°$ (horizontal and vertical cases respectively) on both the velocity induced in the duct and the temperature field generated within the duct.

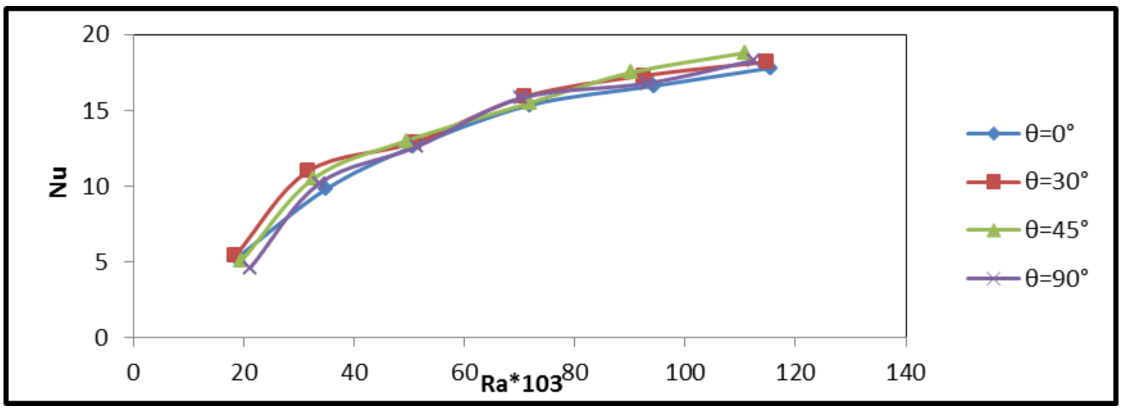
Figures (4) show the numerical results for horizontal ($\theta = 0°$) with the cube being fixed at positions (X/L=5, 10 and 15) respectively. A and b cases shows the thermal field for relatively low input power (power=0.147W) and other higher power (power=3.075W) in each of mentioned figures, while c and d in the figures represent the velocity field through the duct for the same given input power.

The figures declares that higher temperature field results with higher power being input to the module with air plums being directed towared the duct exit in each of the three mentioned positions of the cube, with induced velocities through the duct is relatively higher with relatively higher input power relative to the lower one.

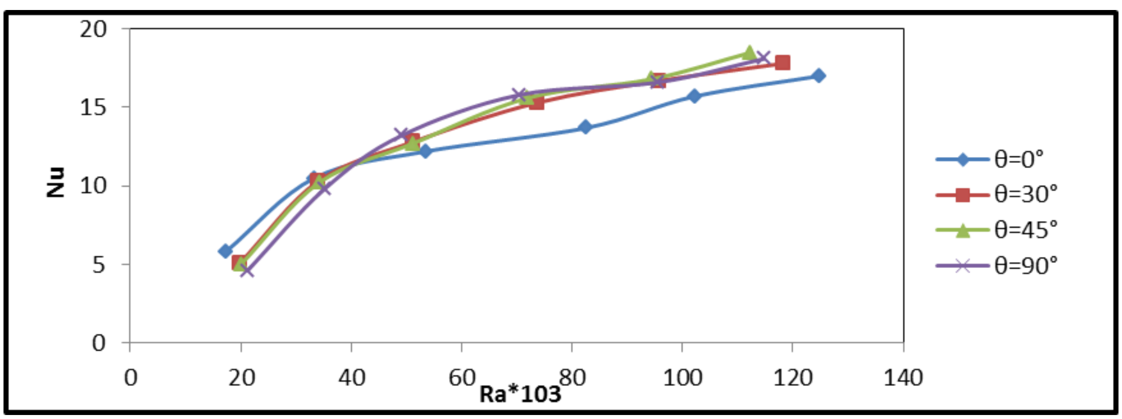
Figures (5) give the numerical results as thermal field (given by a and b) and velocity field (given by c and d) for the case of vertical position ($\theta = 90°$) for tilt angle of the duct for two input powers for the module of (power=0.147W and 3.075W) in each case and for three position of the location for the cube of (X/L=5, 10 and 15) respectively. The results declares higher temperature through the duct and around the cube with relatively higher input power. Unexpected results are noticed in figure (5) for the case of (X/L=5) (i.e at duct entrance) the thermal field show that the air plumes are restricted in motion and higher temperature field are limited in the lower zone below the module in inverse to the expected direction of flow, this can be attributed to the relatively long length of the used duct which gives a restriction to air flow in direction of induced flow casing a vortex near the module.



(a) X/L=5

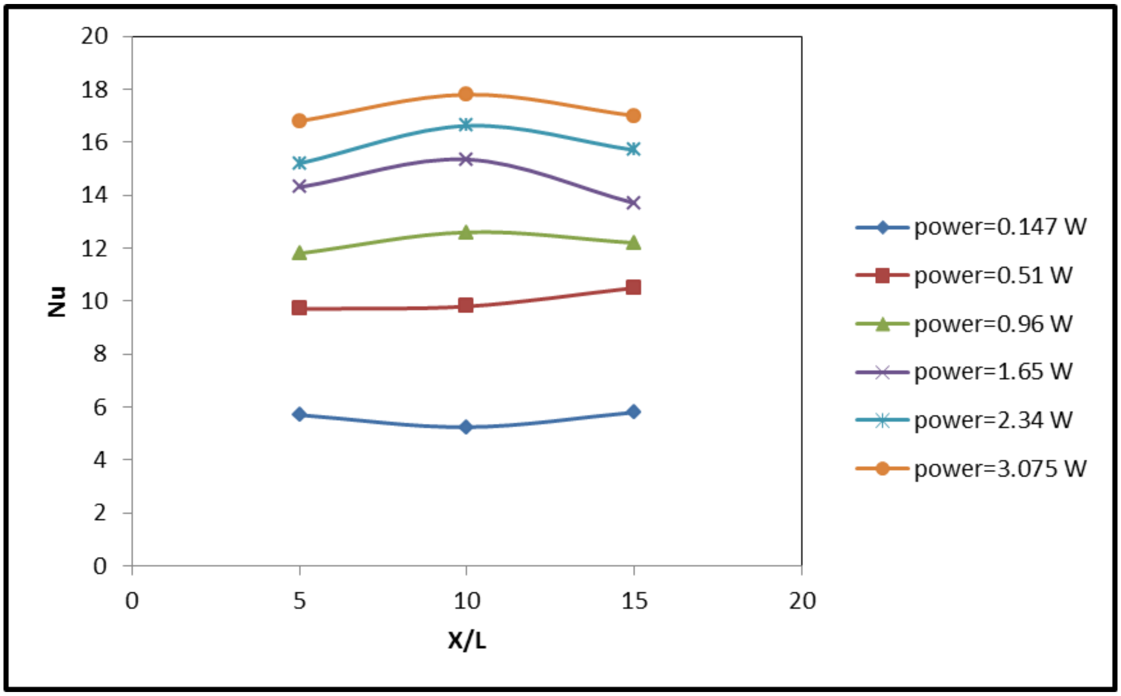


(b) X/L=10

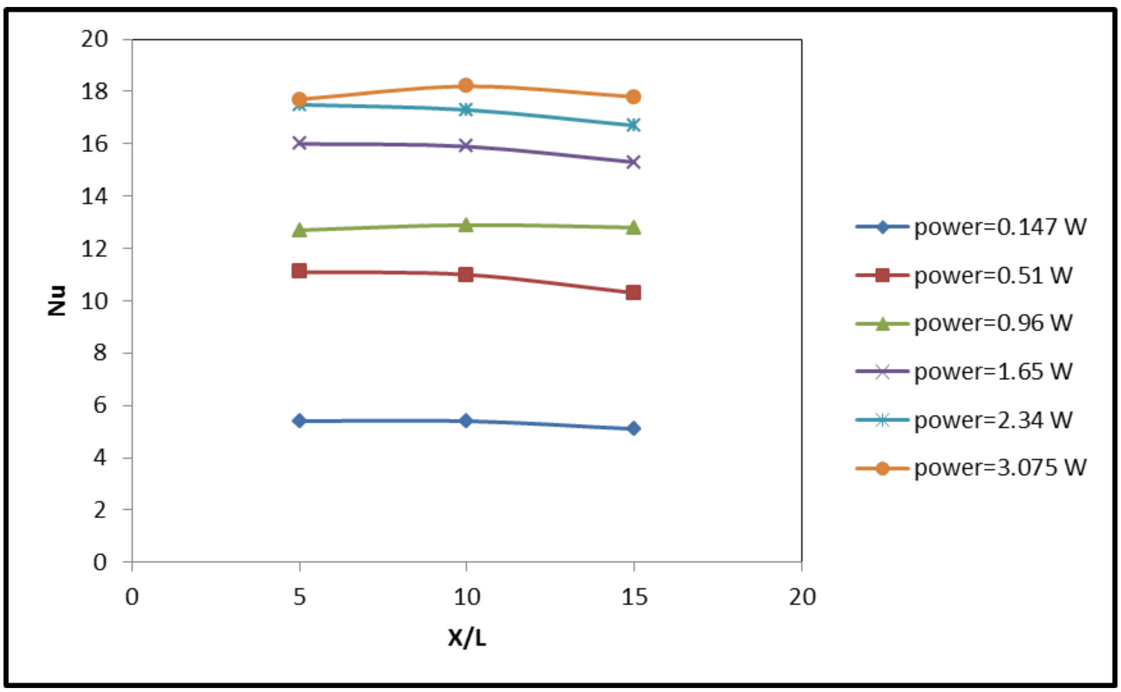


(c) X/L=15

Figure (2): Variation of Nusselt number with Rayleigh number (Natural convection)



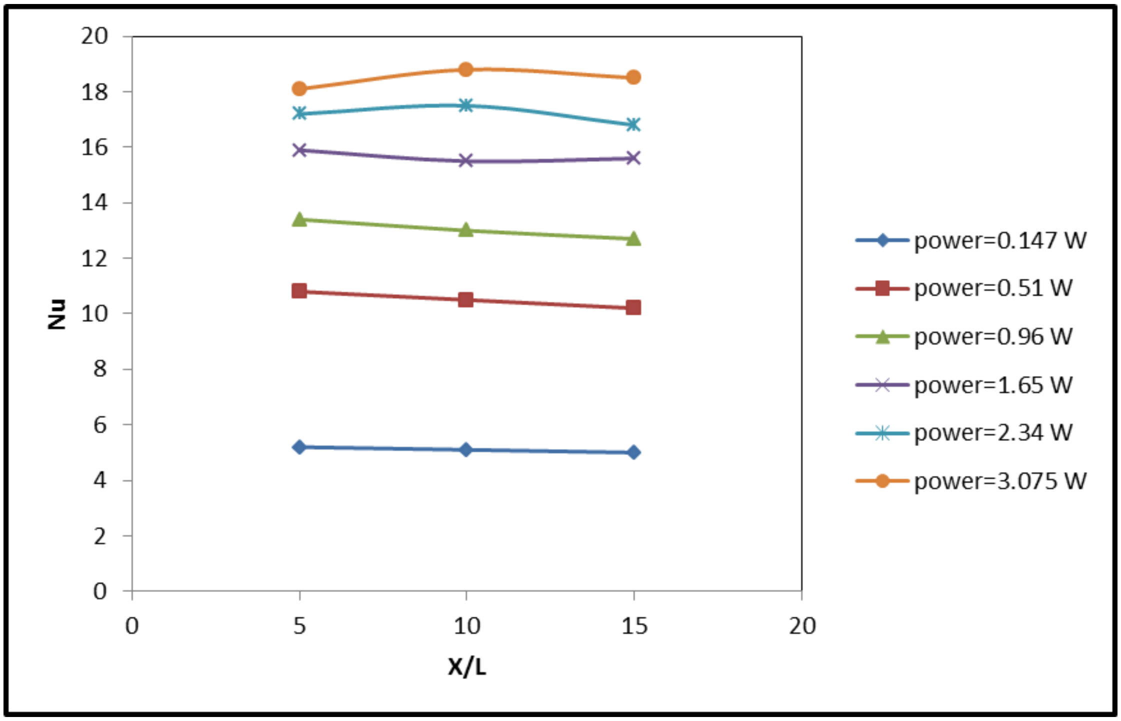
(a)



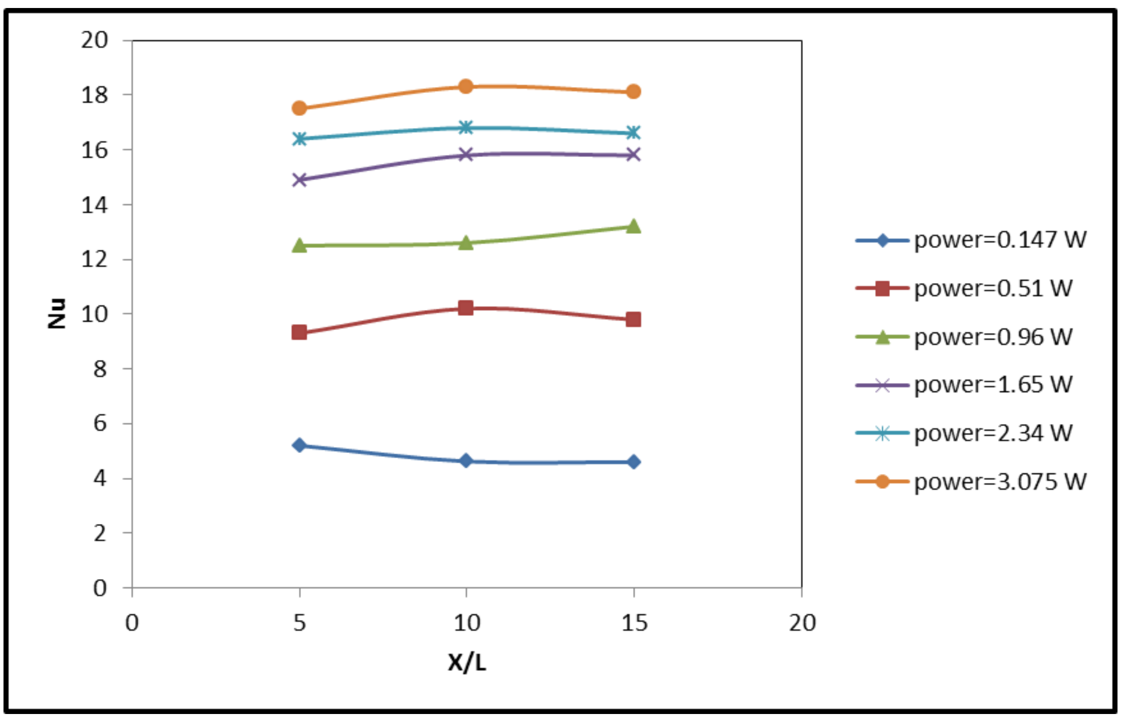
(b)

Figure (3): Variation Nusselt number with X/L (Natural convection)

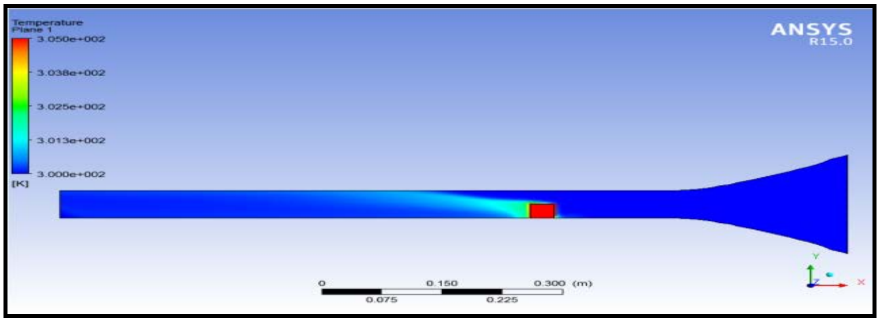
a) $\theta = 0°$ b) $\theta = 30°$ c) $\theta = 45°$ d) $\theta = 90°$


Nu
X/L
0
5
10
15
20
2
4
6
8
12
14
16
18
power=0.147 W
power=0.51 W
power=0.96 W
power=1.65 W
power=2.34 W
power=3.075 W

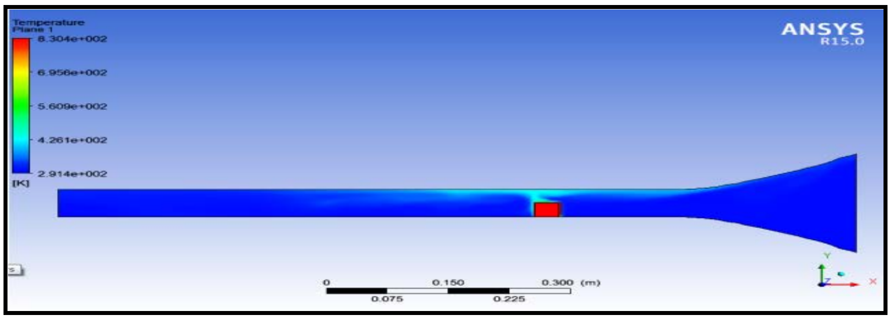
(c)


Nu
X/L
0
5
10
15
20
2
4
6
8
12
14
16
18
power=0.147 W
power=0.51 W
power=0.96 W
power=1.65 W
power=2.34 W
power=3.075 W

(d)

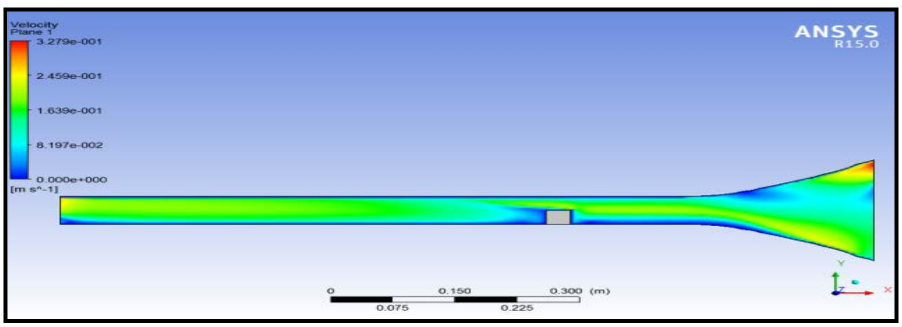


(a) power=0.146W

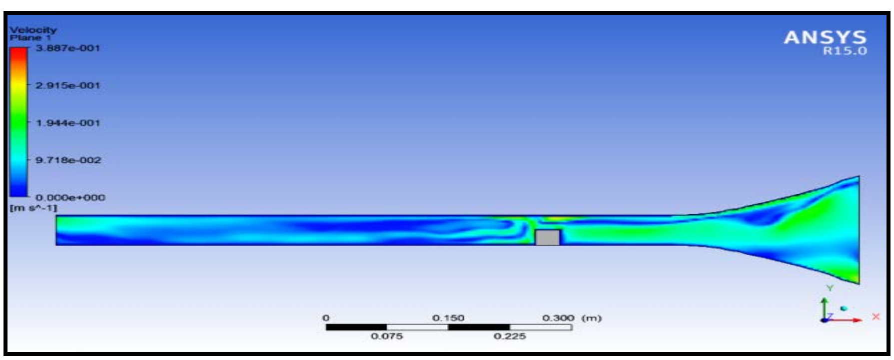


(b) power=3..075W

Figure (4): Thermal field for natural convection, horizontal ($\theta = 0°$), X/L=5

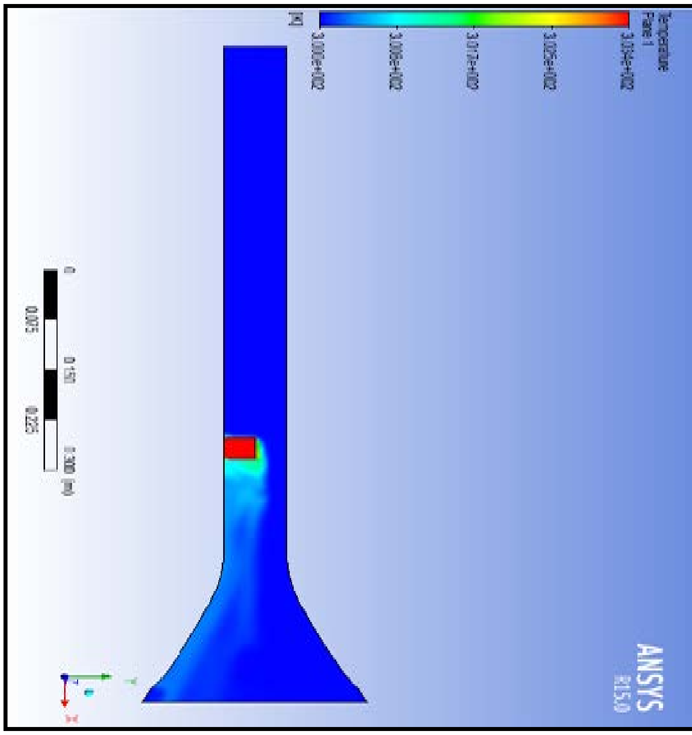


(c) power=0.147W

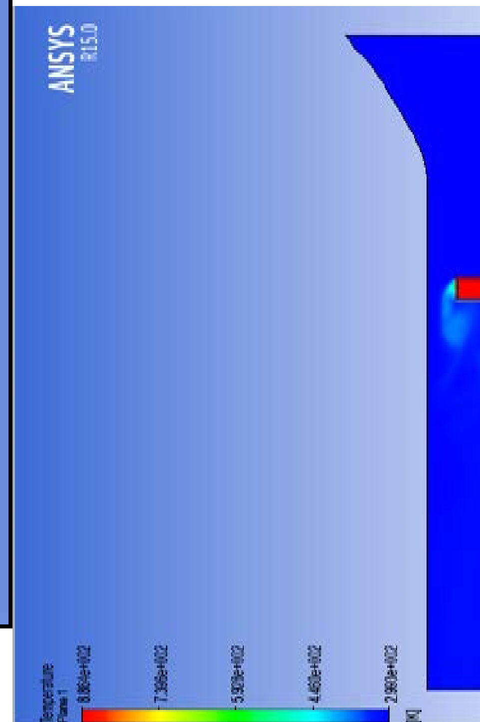


(d) power=3.075W

Figure (4): velocity field at horizontal and natural convection, X/L=5



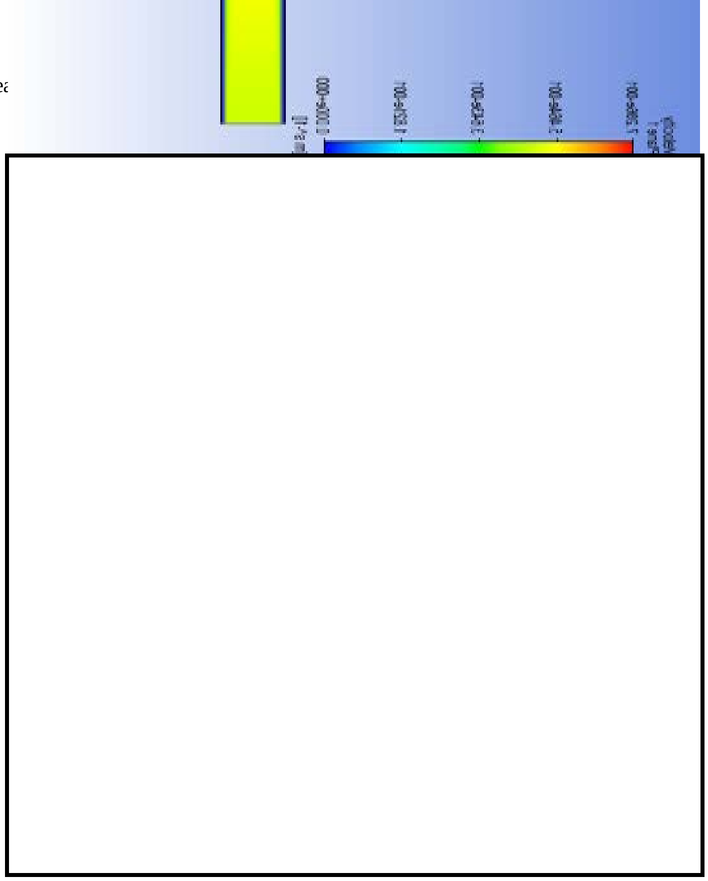
(a)



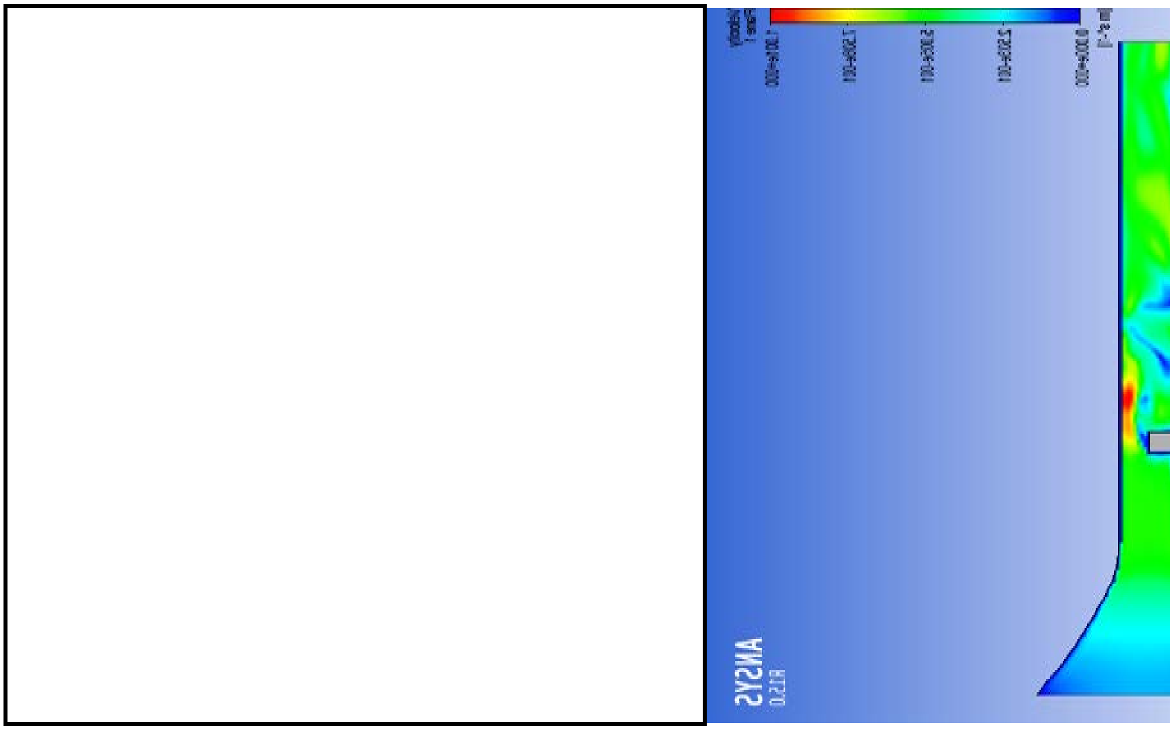
(b)

Figure (5): Thermal field for natural convection, vertical ($\theta = 90°$), X/L=5

a)power=0.147W **b)power=3.075W**



(c)



(d)

Figure (5): velocity field at vertical and natural convection, X/L=5

c)power=0.147W **d)power=3.075W**

VI. CORRELATSIONS

Nusselt number correlations have been developed with the help of relevant dimensionless groups involving parameters like tilt angles of the duct θ, X/L, Reynolds number and Grashof number.

In analyzing the experimental data for the effect of the individual dimensionless group, the values of constant (K) and the exponent (a-d) have been obtained by using LAB fit curve fitting software-version 7.2. The LAB fit is software for windows developed aiming the treatment and the analysis of experimental data and determine propagated error (error propagation up to eight independent variables):

The developed correlation of (AD) approach for Nusselt number (at horizontal case) with error of +7% and -7% is as follows:

$$Nu = K.\left(\frac{X}{L}\right)^{a}.(Gr)^{b} \quad (5)$$

$$Nu = 0.042.\left(\frac{X}{L}\right)^{0.0314}.(Gr)^{0.495} \quad (6)$$

The developed correlation of (AD) approach for Nusselt number (at inclined case) with error of +10% and -10% is as follows:

$$Nu = K.(\theta)^{a}.\left(\frac{x}{L}\right)^{b}.(Gr)^{c} \quad (7)$$

$$Nu = 0.0225.(\theta)^{-0.0377}.\left(\frac{X}{L}\right)^{-0.0123}.(Gr)^{0.564} \quad (8)$$

VII. CONCLUTION

- Heat dissipation rates by natural convection from electric module represented by the Nusselt number increase with increasing the Rayleigh number.
- In natural convection, the value of Nusselt number is higher when the electric module position (X/L=10).
- A general formula for correlating the data of the cube inside the duct is presented in the form

A) At Horizontal

$$Nu = 0.042.\left(\frac{X}{L}\right)^{0.0314}.(Gr)^{0.495}$$

At Inclined

$$Nu = 0.0225.(\theta)^{-0.0377}.\left(\frac{X}{L}\right)^{-0.0123}.(Gr)^{0.564}$$

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