An Investigation of Natural Transition to Mixed Heat Transfer from an Array of Modules in a Horizontal Duct

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Abstract— the present study is focused on studying different modes of heat transfer on a multi row of cubes fixed in a long horizontal duct simulating electronics modules used in a traditional electric device. The cubes are arranged in inline arrangement with 5x8 arrays. The active or heated module is a cube made of duralumin (95% AL) with 30mm side length in which an electric capsule is fixed. This electric capsule is fed with electric current as a part of an electric circuit. The arrangement (the array) is fixed at the lower surface of a variable height (AR) duct in the range of (3.5, 5, and 6.5). The system is operated in the natural convection, transient and forced convection modes. The Reynolds is tested within the duct and values of (Gr/$Re2$) is verified which gives a values within the forced convection mode. A numerical solution for velocity, temperature and turbulence level within the duct is analyzed using the commercial Ansys Fluent software. A control circuit is designed and manufactured so as to develop a closed circuit controlling the fan operation fixed at duct exit. The range of Grashof number is between (49x103 to 165x103) for five values of power input (0.36 to 5 watt), the range of velocities is (0.5 to 2.3 m/sec) and the range of Reynold number calculated depending on the hydraulic diameter is between (4,000 to 12,500).

Keywords: Transient heat transfer in duct, forced heat transfer, natural convection from cube in horizontal duct.

1 Introduction

A Lot of work has been published on the thermal management of electronic devices over more than a century and many papers have been reviewed in this research which has provided good information on the field. For the past 50 years, the usages of electronic devices have been increased exponentially and have become an essential part in every aspect of our daily life. The growth in power and functions of these devices resulted in many associated problems, including the thermal management of these electronic components and systems. The first electronic digital computers has been developed in 1940s, the effective removal of heat played a key role in ensuring the reliable operation of successive generations of computers .The optimal performance and maximum efficiency of the electronic devices depend on maintaining or reducing the temperature of the components and the surroundings. A typical example of current CPU working temperature ranges between (50-100℃) depending on the manufacturers and the purpose of usage. [1]

O.Zeitoun and Mohamed Ali (2006) [2], numerically investigated the natural convection around isothermal horizontal rectangular ducts, with two-dimensional laminar natural-convection heat transfer in air around horizontal ductwork with rectangular and main market square cross sections. Different aspect ratios are being used for a comprehensive Portfolio of Rayleigh numbers. Results are presented in the form of streamlines and isothermal plots around the area of the ducts. They used the finite-element technique. Temperature and velocity information are obtained near each surface of the ductwork. Reverse flow and animation are observed at high aspect ratios. Heat transfer data are produced and presented in conditions of Nusselt number versus Rayleigh number for different aspect ratios.**W. S. Kim and M. N. Ozisik (1987) [3],** studied the thermal transients in forced convection inside ductwork which have numerous applications in the design of control systems for heat exchangers. In this work inductive solutions are developed for unsteady laminar forced convection inside circular tubes and parallel plate channels ensuing from a step deviation in the wall high temperature flux. The generalized integral transform technique (1974) [4] and the classical Laplace transformation are used to create a simple lowest order solution as well as increased alternatives. **Eiamsa-ard and Promvonge (2008)[5]** numerically investigated turbulent forced convection in a 2-D channel with periodic transverse grooves utilizing four different turbulence models: the standard k−ε, the Renormalized Group (RNG) k−ε, the standard k−ω, and the shear stress transport (SST) k−ω turbulence models. With groove width to channel height ratio (B/H) of 0.5 to 1.75. They predicted that RNG and k−ε turbulence model provide better agreement than others and highest heat transfer is achieved for B/H 0.75, they also conclude that increase in recirculation flow helps in increasing the heat transfer) **L. Martinez-Suastegui and C. Trevino (2008) [6]** studied Transitive laminar mixed convection within an asymmetrically and differentially heated vertical channel of finite length subject to an opposing buoyancy is investigated by solving the unsteady two-dimensional Navier-Stokes and energy equations. Results demonstrate the effects of buoyancy strength or Richardson number $Ri = Gr/Re^2$ and Reynolds number Re also on the overall circulation structure and the nondimensional heat flux (Nusselt number) from the heated surface. Final steady or oscillatory flow response is obtained with regards to the value of the Reynolds and Richardson numbers. The critical value of the buoyancy strength between the two regimes is highly determined by the importance of the Reynolds number.

2. EXPERIMENTAL TEST RIG

2.1 Working duct

The working duct is designed and manufactured in order to carry out different experiment covering the modes of heat transfer to be studied which is natural, transient and forced convection. The manufacturing of test rig and the process of choosing the dimensions are according to previous studies to get accurate results. The duct is made of Plexiglas material as shown in figure (1.1). This type of material is chosen due to its low thermal conductivity (K=0.1 w/m-k) and the ease of deforming and construction of the sheets of such material.

Figure (1.1)

A matrix of cubes is fixed in an inline array on the lower part of the duct. This matrix of cubes represent the test section of the

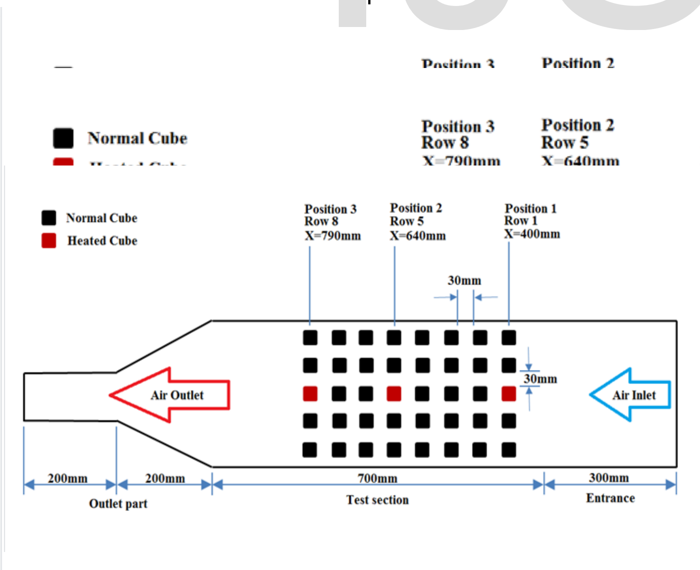

Figure (1.2)

The heated or active module is made in a shape of cube having a dimension of 30 mm side length. It's made of duralumin alloy (90 % AL, 4% Copper, 0.5% Magnesium and 0.5% Manganese). It material is choosen due its high conductivity to reduce temperature variation within it in order to simplify temperature measurement within it by using a single thermal sensor. The heating element is made of electrical resistance of 100 Ω. The cube is drilled with a diameter of 10 mm to sustain the heating element inside it. And there is another small hole (6mm in diameter) used to insert the thermal sensor inside it to measure wall temperature of module as shown figure (1.3)

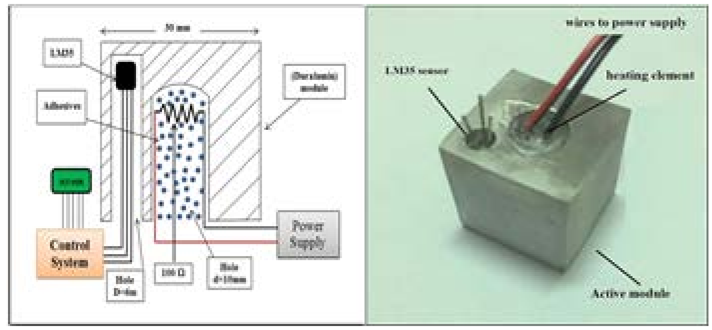

Figure (1.3)

2.2 DC Fan:

A 24 volt DC Fan have Plastic frame (90mmx90mm) with 30 mm thickness placed at the exit of the duct to circulate the air inside duct to get the mass flow rate needed to cooling process.

2.3 Heating system:

Is an electrical circuit consists of three parts, its purpose to generate amount of constant heat flux inside one of the cubes which represent an electrical module. The source of heat generated in the circuit is an electrical resistance linked with voltage power supply to get variable values of heat input that needed to carry out the study and verify the behavior of cooling rate during the variations occurs within the tests.

2.4 Instrumentations:

2.4.1 Velocity Measurement: HOT WIRE ANEMOMETER Model: YK-2005AH

2.4.2) Temperature Measurement:

1: Lm 35 Temperature Sensor

2: Type K Thermocouple

2.5 Automatic Temperature Control System (ATCS)

It has been designed a control system to control the temperature of the system, fan speed and the amount of fluid flowing into the duct by programming the micro controller and specify the required temperature range and display these values in the form of degree Centigrade. The Control system consists of several parts connected with working duct and measurement devices to get an accurate results, good operating temperature and minimum power consumption. The main advantage of control temperature system is to keep the temperature under the manufacturer limits, reduce the vibration and noise because the fan turned off when the system does not need to any cooling process. On the other hand, the power consumption used to run the fan reduced approximately by 50 % because the fan turned off when the temperature is under the allowable limit.

The control system consists of following parts:

1) Micro controller (Arduino) The Arduino UNO is a microcontroller board based on the ATmega328
2) 4 Relay module

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3) LCD Screen 16x2
4) Resistance

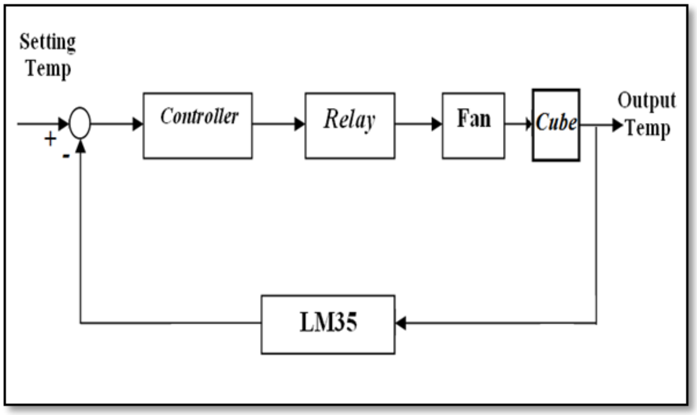

Figure (1.4)

3. Reading collection

The film temperature can be calculated as shown below

$T_f = \frac{T_s + T_\infty}{2}$

The net heat transfer by convection can be calculated by the following balance:

$Q_{conv.} = Q_{in} - Q_{rad.} - Q_{cond.}$

$Q_{in} = VI$ (W)

$Q_{rad} = \epsilon \sigma A_s \, (T_s^4 - T_\infty^4)$ (W)

b. $Q_{conduction}$ = heat transfer by conduction through the base of the module usually neglected due to small conductivity at the base of cube.

Average Convective heat transfer coefficient for each point is calculated as follows:

$q'' = \frac{Q_{conv}}{A}$ (W/m^2)

$h = \frac{q''}{(T_s - T_\infty)}$ (W/m^2.°C)

Now to calculate Nu and Gr number:

$Nu = \frac{h\,L}{K}$

Gr = $\frac{g\,\beta\,(T_s - T_\infty)\,L^3}{\nu^2}$

There are two methods used to calculate Reynolds number depending on the characteristic length, the first one used the hydraulic diameter (D_H) and second used side length of the cube (L).

$Re_D = \frac{\rho_A U_A D_H}{\mu_A}$

$Re_L = \frac{\rho_A U_A L}{\mu_A}$

Using the energy equation representing the heat balance during transient operation:

Heat input – losses - Q_{conv} = m $C_p \frac{dT}{dt}$

4. Results and discussions

The system is operated in different modes, first is by free convection when module temperature reaches its highest temperature. Then the fan is operated in a certain speed. Re_D Is fixed at its lowest value then increased gradually. During operation of the fan till reaching steady state a transient period elapsed. Heat transfer coefficient is calculated during each mode of operation. The heated cube position is changed from one position to another to study different possible cases. (i.e. the cube is put at mid position in each of the eight rows and changed in each case). The test variables are shown in table -1

No.	variable	case 1	case 2	case 3	case 4	case 5
1	Input power (w/m^2)	80	320	500	720	1075
2	Velocity (m/sec)	0.5	0.8	1	1.1	1.7
3	position	Row 1	Row 5	Row 8	----	----
4	Aspect ratio	3.5	5	6.5	----	----

Table (1) calculation variables

Reynolds number is calculated in two ways, either on the basis of hydraulic diameter of the duct (Re_D) which gives an indication for the type of the flow if it is laminar or turbulent, or on the basis of cube side length (L) i.e. (Re_L) which gives an indication for the type of flow or the mode of convection when compared with Gr_L If it is natural, transient or forced convection. (I.e. using Gr/R^2 as verification number indicating the heat transfer mode). Usually if Gr/R^2 ≈ 1 the flow is described as mixed convection while if Gr/R^2 << 1 the flow is denoted as forced convection mode. The free convection mode tests are done by heating the module until reaching the steady state condition with maximum temperature is reached at each test, variable aspect ratios of values of (3.5, 5 and 6.5) are tested with verification of Nu values in each of the taken cases are done.

4.1 Free convection results

Variation of Nusselt number with Grashof number (effect of duct height or aspect ratio):

Figure (4.1.a) indicates that at the position 1 (row number 1) increased value of Gr number which represent higher heating level (greater temperature difference will give a chance to the creation of higher buoyancy forces giving a chance for greater induced velocities specially when the aspect ratio of the duct is increased as indicated.

Figure (4.1.b) indicates the same behavior for Nu-Gr variations. Nu starts to increase with a high rate and ΔT increases Gr number increases. The rate of increase then decreases due to limitations in air circulation and freedom in motion. The bigger the aspect ratio gives higher Nu number values as

ISSN 2229-5518

shown. This position may represent a restriction for air movement. Figure (4.1.c) represents the relationship between Nu and Gr in free convection mode for third position (i.e. row number 8). The last position of heated cube is the nearest position from the outlet. As shown in figure the same trend of the results exist. Smaller areas at duct exit may cause a flow restriction that slows the cooling process.

Figure (4.2.a) shows that slow rate of Nu increase is noted and as the heating power and temperature differences increases the rate of heat removal and slower cooling rates is noted due to the restriction in induced plumes flow from the cube. It can be expected that module position has a noticeable effect on heat transfer coefficients and Nu number values due to flow condition in each of the tested positions.

The figures (4.2 b and c) shows that the behavior of the heat transfer changes according to the module positions. This can be discussed on the basis of freedom of air motion within the duct at different used aspect ratios or duct height.

4.2 Forced Convection Results

According to the previous studies, the value of (Gr/Re^2) give an indicated about the mode of heat transfer if ($Gr/Re^2 >> 1$) the forced convection may be neglected and if ($Gr/Re^2 << 1$) the effect of natural convection may be neglected. In present work, even though the flow can be assumed as forced convection mode, testing the (Gr/Re^2) values is in the range of (0.1 to 0.003) this range means that the effect of natural convection still exist as will be indicated in the results. The system is tested in the forced convection mode using different fan speeds which includes variable velocities through the duct. Different positions of the heated module are tested with different values of H/L = 1.5, 2, 3 which correspond to aspect ratio of (AR=6.5, 5, 3.5) respectively.

Figures show that as air velocities increase the Nusselt number increases almost in linear behavior. The results show that the heating power gives higher temperature levels for the heated modules. As shown in figure (4.3 a,b and c) the natural convection still play a big role in increasing the Nusselt number due to induced air velocities due to buoyancy effect in addition to the forced convection effect. Higher values of Nusselt number are noted relative to the pure natural convection mode.

Figure (4.4 a,b and c) shows the Nu, Re relation for the second position (row no. 5) of the heated module. Linear behavior is noted with same trend of the results which indicates an additional cooling effect due to natural convection which gives higher value of Nusselt number (i.e. higher cooling rates of the module).

The third position results represents the same trend of results with almost same range of Nusselt number for the higher heating power while the low cooling power cooling rates is increased as noticed with higher values of Nusselt number for the lowest heating power in each of figures (4.5 a, b and c).

4.3. Transient Convection Results

Transient convection heat transfer coefficients are measured experimentally at three locations of the module, at the 1^{st}, 5th and 8^{th} row of the array used in the test section. Different three aspect ratios are also tested for the used duct, applying five different heating powers to the active module.

The transient period is that period which takes place when the cooling fan is initially operated and the air starts to flow in the duct. The maximum constant temperature reached during free convection period, Starts to reduce continuously till reaching the steady state condition when the temperature of the module reach a constant value which is the minimum temperature of the module in each test experiment with various conditions mentioned.

Figures (5.18) to (5.26) represent the heat transfer coefficients represented by Nusselt number for the active heated module versus the Reynolds number calculated on the basis of hydraulic diameter of the duct and average velocity through the duct.

It can be noticed from these figures that the Nusselt values are lower than that for the case of forced convection using same module positions, heating powers and duct aspect ratios. Slower rate of Nusselt number increase with increasing Reynolds number is noticed relative to that for the case of forced convection.

Almost linear variation of Nusselt with Reynolds number can be detected with the great heating power is conjugated with higher heat transfer coefficients. This behavior can be attributed to increased natural convection heat transferred in addition to forced convection during these transient periods. Overall cooling effect of the module can be treated as conjugated of the two heat transfer modes related to each other in a complex manner depends on the growth of the thermal boundary layer on the heated cube and natural convection plums from it.

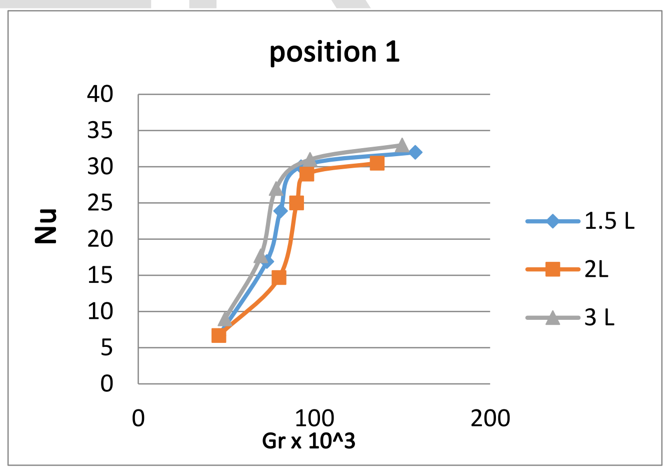

Figure (4.1.a) Relation between Nu & Gr at H/L= (3, 2, 1.5) and at X =400 mm

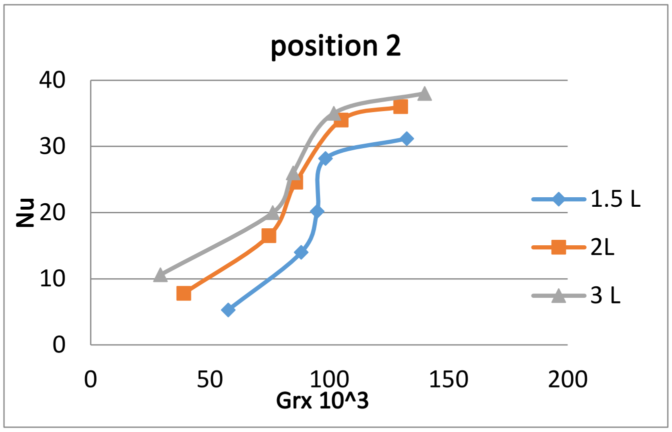

Figure (4.1.b) Relation between Nu & Gr at H/L= (3, 2, 1.5) and at X = 640 mm

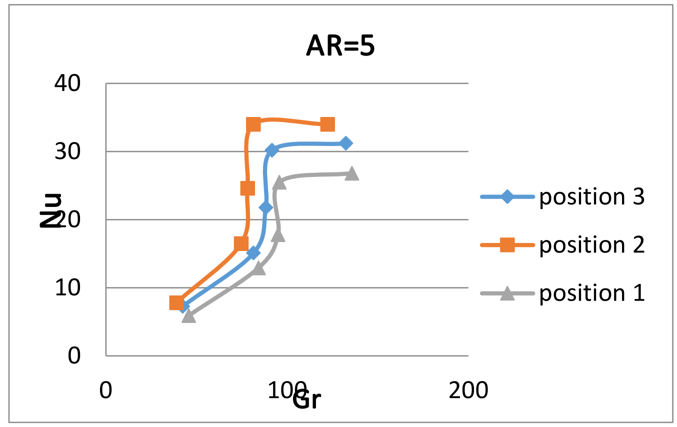

Figure (4.2.b) Relation between Nu & Gr at positions (400, 640,790) mm and H/L = 2

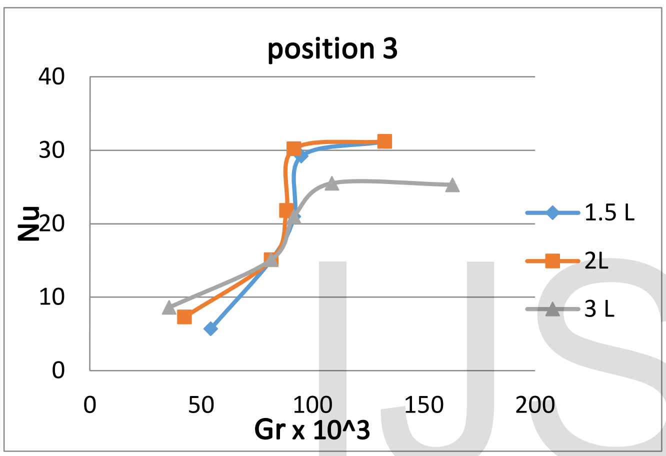

Figure (4.1.c) Relation between Nu & Gr at H/L= (3, 2, 1.5) and at X = 790 mm

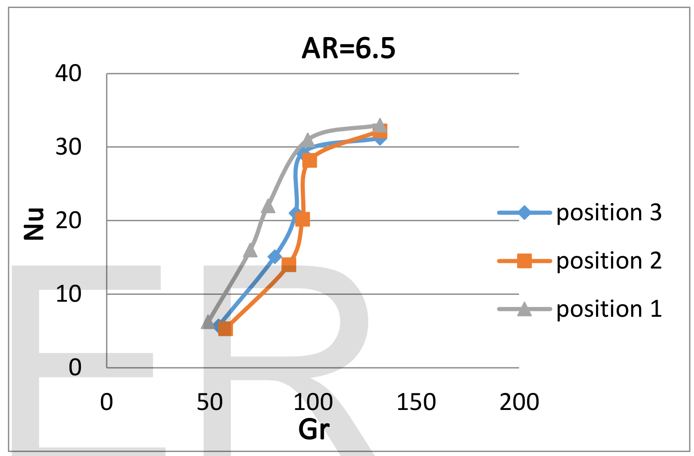

Figure (4.2.c) Relation between Nu & Gr at positions (400, 640,790) mm and H/L = 1.5

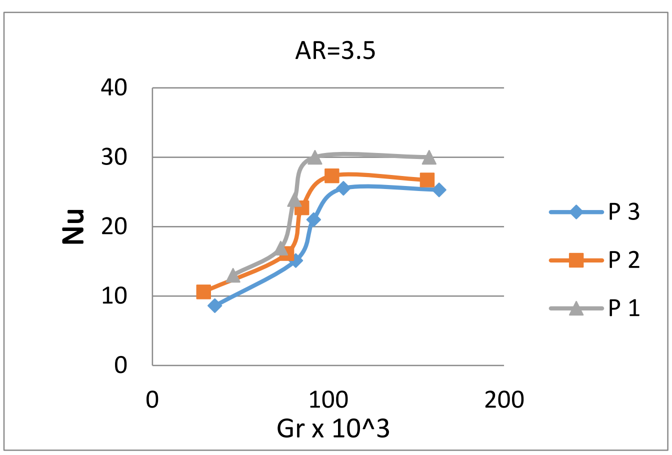

Figure (4.2.a) Relation between Nu & Gr at positions (400, 640,790) mm and H/L = 3

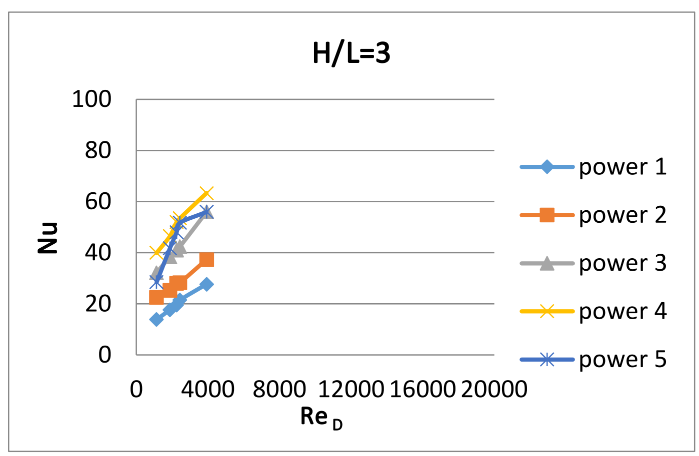

Figure (4.3.a) relation between Nu & Re for X= 400 mm and H/L=3 at different ranges of power

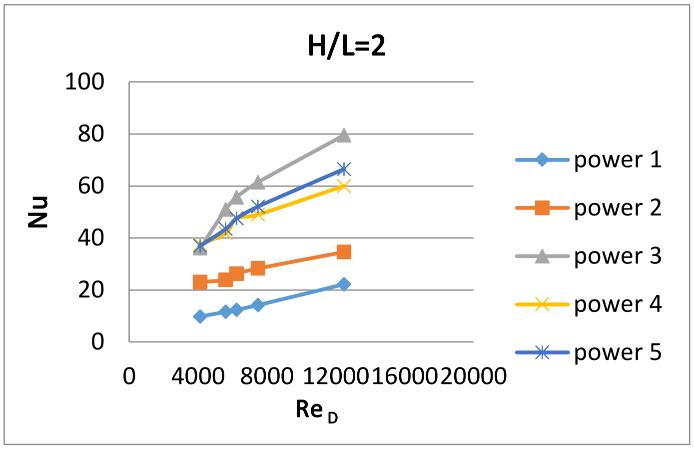

Figure (4.3.b) relation between Nu & Re for X= 400 mm and H/L=2 at different ranges of power

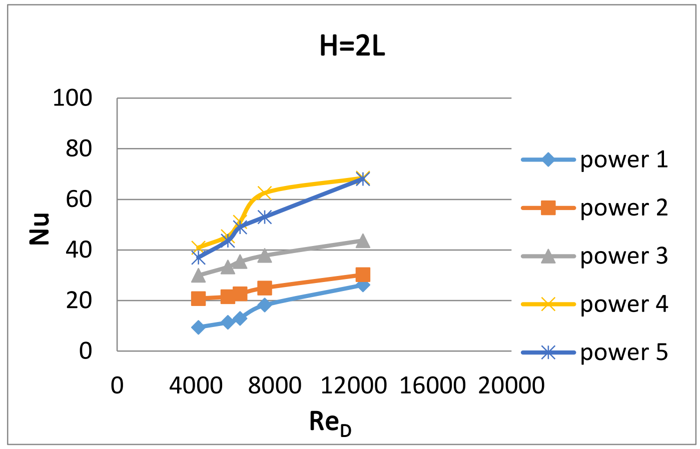

Figure (4.4.b) relation between Nu & Re for X= 640 mm and H/L=2 at different ranges of power

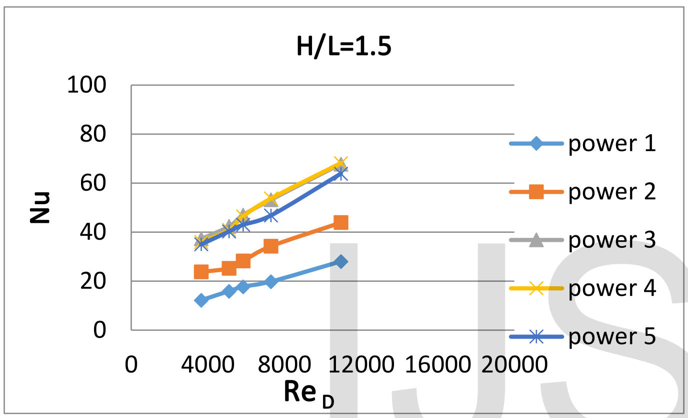

Figure (4.3.c) relation between Nu & Re for X= 400 mm and H/L=1.5 at different ranges of power

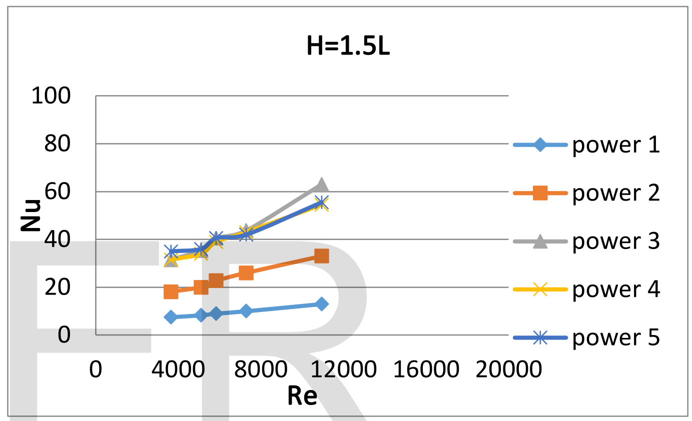

Figure (4.4.c) relation between Nu & Re for X= 640 mm and H/L=1.5 at different ranges of power

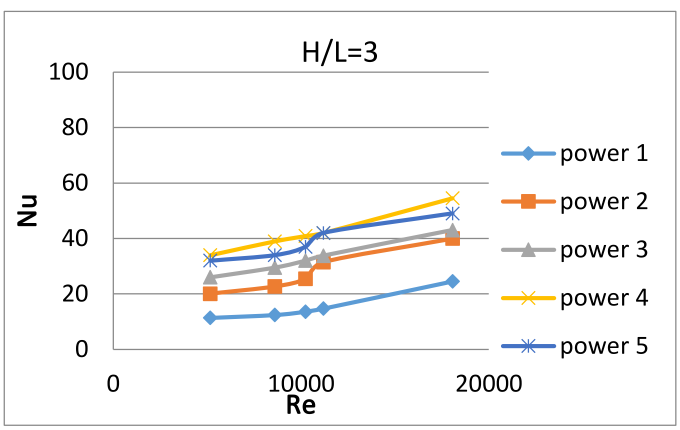

Figure (4.4.a) relation between Nu & Re for X= 640 mm and H/L=3 at different ranges of power

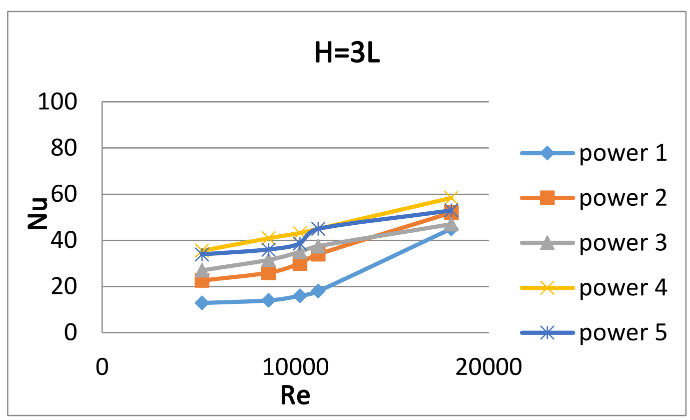

Figure (4.5.a) relation between Nu & Re for X= 790 mm and H/L=3 at different ranges of power

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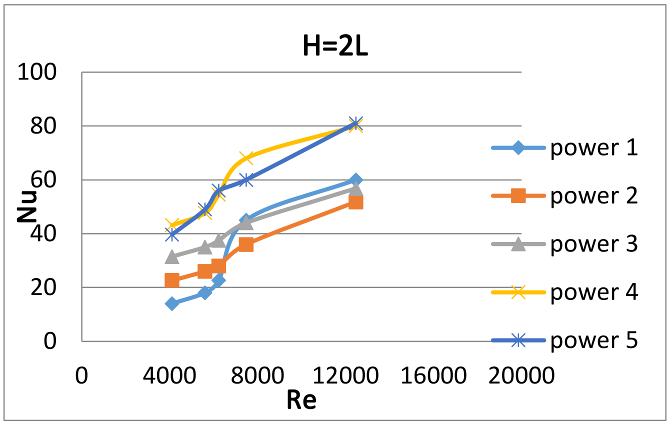

figure (4.5.b) relation between Nu & Re for X= 790 mm and H/L=2 at different ranges of power

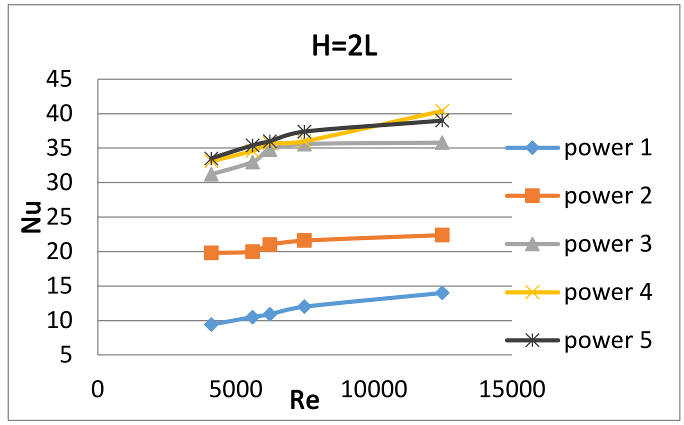

Figure (4.6.b) relation between Nu & Re for X= 400 mm and H/L=2 at different ranges of power

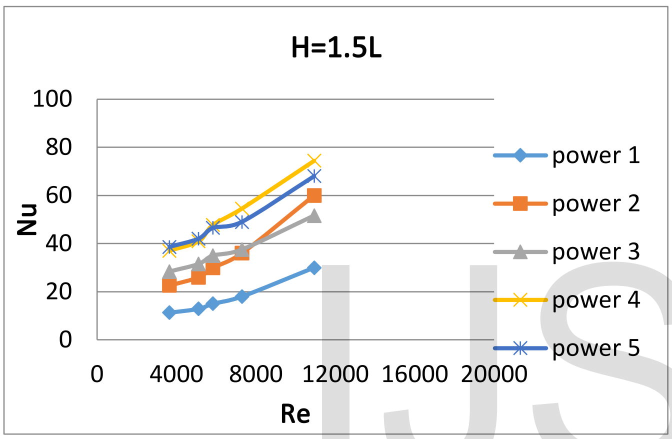

Figure (4.5.c) relation between Nu & Re for X= 790 mm and H/L=1.5 at different ranges of power

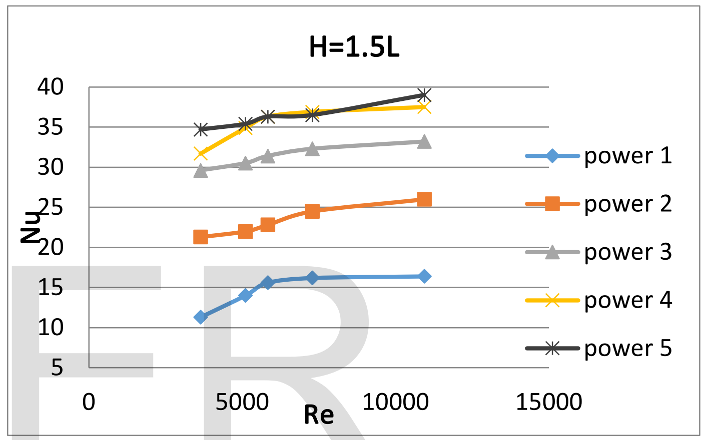

Figure (4.6.c) relation between Nu & Re for X= 400 mm and H/L=1.5 at different ranges of power

Transient Convection Heat Transfer Result

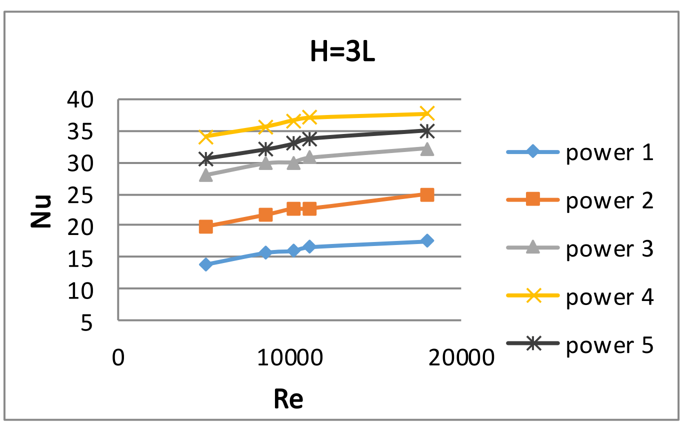

Figure (4.6.a) relation between Nu & Re for X= 400 mm and H/L=3 at different ranges of power

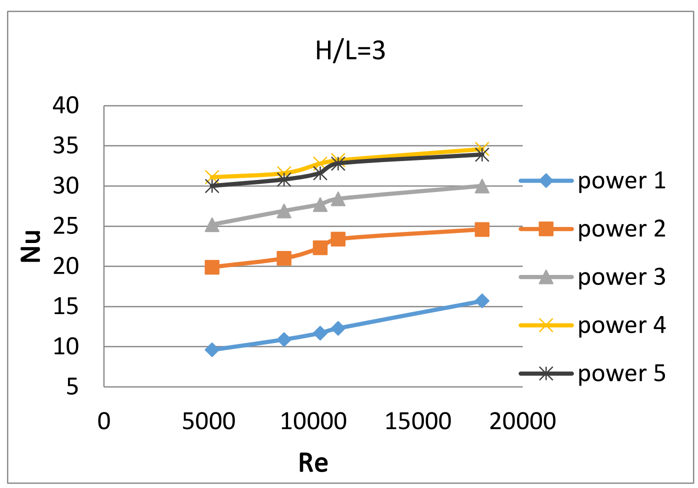

Figure (4.7.a) relation between Nu & Re for X= 640 mm and H/L=3 at different ranges of power

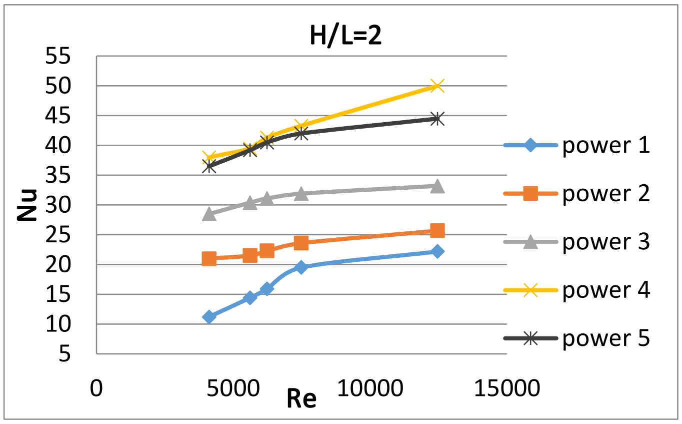

Figure (4.7.b) relation between Nu & Re for X= 640 mm and H/L=2 at different ranges of power

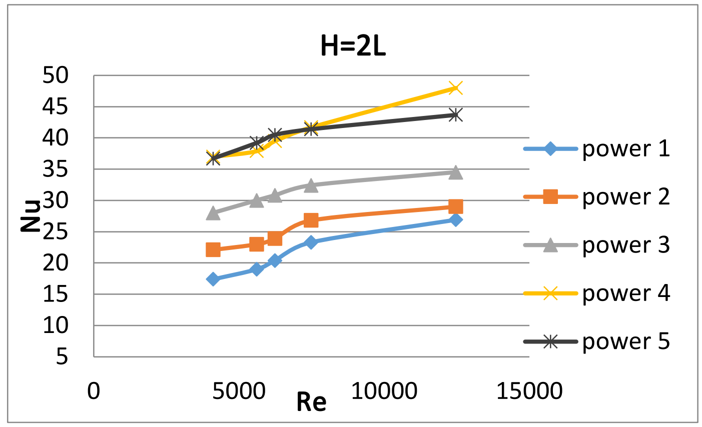

Figure (4.8.b) relation between Nu & Re for X= 790 mm and H/L=2 at different ranges of power

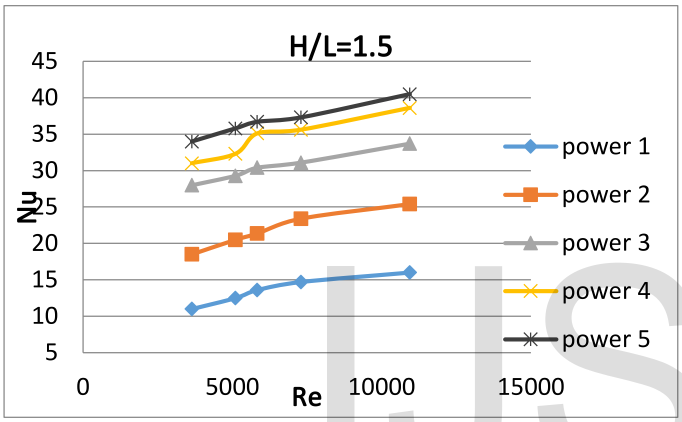

Figure (4.7.c) relation between Nu & Re for X= 640 mm and H/L=1.5 at different ranges of power

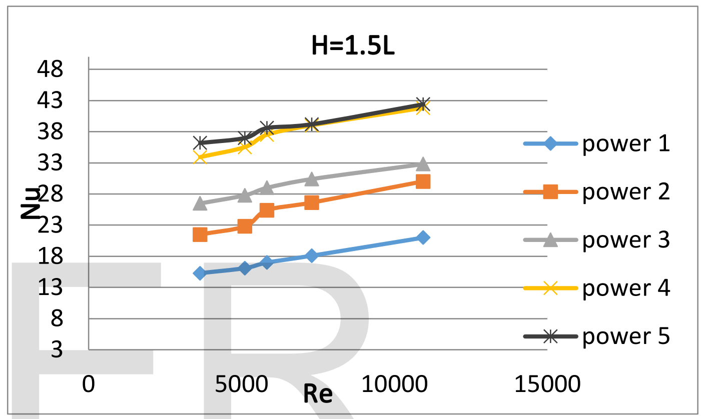

Figure (4.8.c) relation between Nu & Re for X= 790 mm and H/L=1.5 at different ranges of power

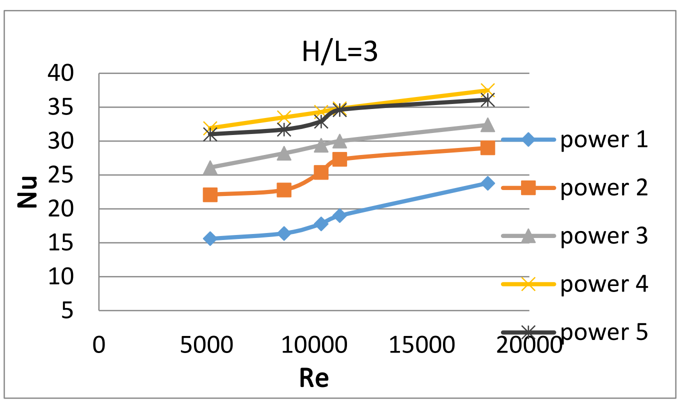

Figure (4.8.a) relation between Nu & Re for X= 790 mm and H/L=3 at different ranges of power

5. Conclusions

1. The first position of module (X=400, Row No. 1) has the maximum heat transfer rate due to high turbulence.
2. The height of the duct to side length (H/L=3) has significant effect on heat transfer rate.
3. The heat transfer rate increases with Reynolds number.
4. The heat transfer coefficient increases in transient period more than the steady state.
5. The heat transfer coefficient increases with power input increases.
6. The increase of ratio of (H/L) increase the heat transfer and velocity.
7. The position number 2 (X=640mm, Row No. 5) shows the lowest heat transfer rate.
8. The large area of entrance gives good heat transfer rate especially in the first position of the heated cube.
9. The power consumption decrease when using the closed loop system approximately by 50 %.
10. The closed loop system gives a good operating tempera-ture, less vibration and less noise of fan.
11. From numerical simulations, it was found that the heat generated from the electrical resistance less than the pow-

er input to the electrical circuit due to heat losses by conduction from heated module to the bottom base of the duct.

12. The numerical prediction shows a good agreement with experimental results.

6. Recommendations

1. Use multi heated cube and study the effect of the central cube and the properties of air around it.
2. Programing the micro controller to change the velocity of fan with temperature variation.
3. Testing the thermal conductivity of duralumin for more accurate calculations.

7. References

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