

SHIFTING OF HEAT ENERGY VIA TRANSVERSE PARALLEL SHEETS CONFIRMED WITH NATURAL CONVECTION

Abstract

Due to the progression in each and every aspect concern with energy production, utilization and harnessing of the same has been drastically optimized. The energy draws out from heat sources is one of a type among all convectional and non-convectional sources of energies. The pattern of heat energy shift through natural convection has a number of applications found in the market. The uses of this kind of energy can be listed as electrical and electronics equipments, nuclear reactors, domestic convection, dry cooling towers, thermo siphons, installed in ground and many more. On the account of functional continuity and longevity aspect the heat generates due to work execution must be radiated out of the machineries. Considering the influence concern with heat energy shift upon the untie finished pipes of upright means, it is of very trivial magnitude, As a matter of fact, the diverse field of solar and nuclear energy connecting with thermal fluid systems has also been greatly impacted by natural convection heat energy flow. Converging to the current event where heat energy shifts between two parallel flat plates excited electrically on the external region of the units to keeping steady heat flux at the boundary. The magnitude pertaining with parameters/edges like thickness, Breadth and length are 5mm, 150mm and 500 mm correspondingly. Since the exterior region maintained insulation, hence the shifting of heat energy is admitted to regulate from interior region towards the adjoined air molecules. The particular wall heat energy flux denoted by the symbol 'q' maintained at a magnitude of 2188W/m^2 . consequent to this a definite analytical as well as observational values adhering with steady state phenomena

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has been laid out. Keeping the heat flux status same the temp values drawn out analytically on behalf of air and wall units have successfully monitored against the particular experimental values. The outcomes concern with both the approaches match with one another yielding the heat flux magnitude of 2188W/m^2 .

Keywords: Heat energy surge, usual heat convection, heat energy shift, CFD.

I. INTRODUCTION

Heat transfer through natural convection is an economical method of cooling which is widely accepted by electronic industry in order to dissipate heat energy comes out of the electronic gadgets, entrenched upon trivial platform to improve their life span and efficiency. The exact shifting of heat energy implies the heat energy transmission from the heated plate to air due to temperature difference between them. Hot bodies can be cooled faster on behalf of forced heat convection than other means. Natural convection occurs because of the difference in mass compactness of fluid resulting out of the temperature dissimilarity. The ambient air in close proximity of the hot bodies is less than the air away from it. Due to difference in density of fluid, a buoyant force is created which enables the lighter hot air to flow in upward direction. Natural convection observed in diverse domains such as nuclear reactors, power plants adhere pipe carrying steams, the very cooling towers, convectors in domestic means, siphons of thermal means etc. The exact aim behind this investigation implies findings of natural convection via heated upright ducts in both theoretically and experimentally. The test section consists of two vertical plates which are electrically heated on the outer surface, dispelling heat from the inner surface to air. The observational part excited via electrical means keeping fixed heat energy surge on the boundary. The temperature of air in the vicinity of heated plate increases which results in decrease of density of air near the heated plate. Due to variation of density of air a buoyant force is created which causes the hot air through the two plates to move in upward direction. Natural convection heat transfer experiment is conducted with constant wall heat flux conditions for plates of dimensions 500mm length, 150 mm width and 3 mm thickness.

Dynamics of fluid flow and heat transfer through vertical pipes and parallel flat plates have been investigated by many researchers. However, the works on heat transfer with existence of rings are not satisfactory in the literature. Mallik and Sastri [1] experimentally studied the natural convection heat flow over staggered discrete plates and observed that natural convection heat flow over discrete vertical plate is more than that over flat plates. Sparrow and Prakash [2] have examined the natural convection from staggered discrete plates and compared the results of staggered discrete vertical plates with parallel flat plates, considering constant wall temperature. They noticed that with increase in spacing, decrease in height of channels results in improvement of heat transfer. Hung and Shiau [3] studied natural convection flow through vertical parallel plates with rectangular ribs theoretically as well as experimentally. They noticed that heat transfer phenomena in the region beyond the ribs is just like the flow in the turbulent flow conditions. They proposed two correlations for predicting Nusselt number in the downstream vicinity of the rib for calculating heat transfer characteristics. Gortysov *et al.* [4] experimentally investigated the hydrodynamics and heat exchange in vertical channels. They observed that provision of discrete rings in the internal surface results in increase in heat transfer from the vertical walls to air. Experimental investigation by Sparrow and Bahrami [5] imposing three types of boundary conditions on the lateral walls. Dixit *et al.* [6], [7] and [8] used different optimization techniques for example bacterial colony optimization (BCO), cuckoo search (CS), and group search optimization (GSO) to determine the most favorable weight of the structure. Levy *et al.* [9] pointed out the issue concern to most favorable plate gap on behalf of enhancement of streamlined usual heat energy surge among the specimens. Roul and Nayak [10] presented heat flow characteristics through heated vertical tubes. Nayak *et al.* [11-13] investigated the improvement of heat energy shift in upright tube owing to the provision of internal rings.

They have taken obstacles of rectangular cross section as internal rings in their work. Sahoo *et al.* [14-17] applied computational techniques to investigate heat transfer and flow dynamics from pin finned vertical plates. Roul and Sahoo [18] explored the two-phase flow dynamics and pressure drops through pipes with variations in cross sectional area which is essential for assessing power of pump to create forced draft. Churchill and Chu [19] developed general correlation equations applying the experimental and theoretical data for the laminar natural convection flow through vertical channels which is driven by buoyancy. Dey *et al.* [20] observed that the flow dynamics in the close proximity of a fin can significantly improve the heat transfer from a heat sink due to forced convection. The rise in heat transfer is owing to an increase in temperature gradient, as the thickness of the thermal boundary layer decreases because of the presence of fins. Because of the enhancement of temperature gradient, the average Nusselt Number increases on the fin surface. Fluid dynamics and pressure drop predictions through sudden expansions and contractions have been studied using CFD modeling [21-25]. Pradhan *et al.* [26] investigated the heat flow characteristics through an 180° bend pipe having different cross sections using nano-enhanced ionic liquids (NEILs). Buonomo and Manca [27] studied natural convection heat transfer through a vertical micro channel subjected to constant heat flux. They calculated Rayleigh number in terms of wall heat flux in order to evaluate their influence on wall temperature, velocity profile, Nusselt number, and mass flow rate. They perceived that there is an increase in the wall temperature with an enhancement in Knudsen number for higher values of the Rayleigh number. However, they noticed that the wall temperature is minimum for lower values of Rayleigh number for $Kn = 0.05$. They also noticed that Nusselt number declines when Kn increases and mass flow rate upsurges when Kn increases. El-Morshedy *et al.* [28] experimentally explored the heat flow due to natural convection in thin vertical channels having rectangular heated walls. They proposed two correlations for the local Nusselt number for natural convection flows and forced convection flows respectively. The prediction from the correlations was compared with the experimental data which was observed to be fitting well with the experimental data within 5.8% standard deviations.

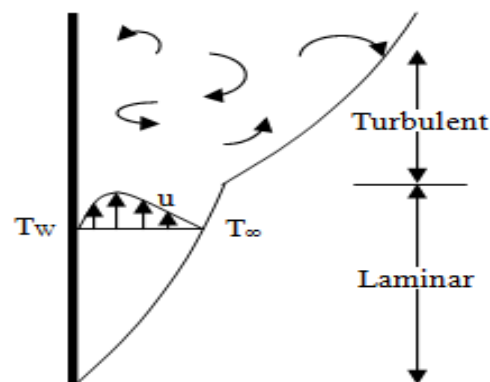


Figure 1: Boundary layer

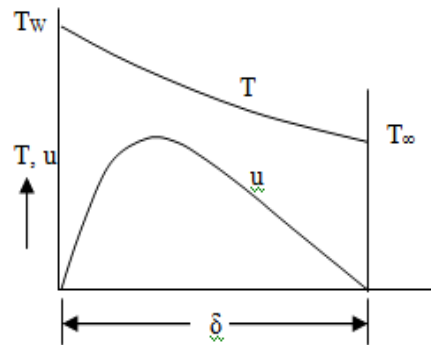


Figure 2: Velocity and Temperature profile on a vertical plate

The average coefficients concern to heat energy shift may expressed in different environments as per the details given below:

$$\overline{Nu}_f = C(Gr_f Pr_f)^m \quad (1)$$

In above notation ‘ f ’ specifies unit-less category of fluid properties where the factor “temperature” is evaluated via:

$$T_f = \frac{T_w + T_\infty}{2} \quad (2)$$

The Rayleigh number is the assessment experienced via the multiplication of Prandtl number and Grashof number:

$$Ra = Gr * Pr \quad (3)$$

The typical length taken inside determination of Grashof as well as Nusselt number depends upon the very physical configuration considered within issue. The specimen laminate height (L) is considered as the characteristic dimension for flat vertical plates and the diameter (D) is taken as the characteristic dimension for horizontal cylinders. The equations used to determine the heat transfer for vertical flat plates can also be considered for vertical cylinders when width concern with outside film not being very big in comparison with cylinder diameter.

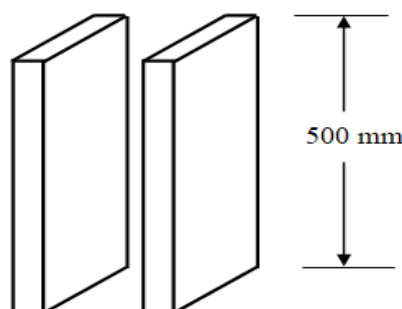


Figure 3: Heated vertical parallel plates

As a matter of fact, an upright cylinder is generally considered as one upright flat plates under the following conditions.

$$\frac{D}{L} \geq \frac{35}{Gr_L^{1/4}} \quad (4)$$

In Eq. 4, D is taken as the cylinder diameter. For determination of Nusselt number the following equation is considered.

$$Nu_f = 0.10(Nu_f Pr_f)^{1/3} \quad (5)$$

Churchill and Chu [19] have provided more relevant equations which are appropriate over extensive ranges of Rayleigh numbers:

$$\overline{Nu}_f = 0.68 + \frac{0.670 Ra^{1/4}}{[1+(0.492/Pr)^{9/16}]^{4/9}} \text{ for } Ra_L < 10^9 \quad (6)$$

$$\overline{Nu}_f^{1/2} = 0.825 + \frac{0.387 Ra^{1/6}}{[1+(0.492/Pr)^{9/16}]^{8/27}} \text{ for } 10^{-1} < Ra_L < 10^{12} \quad (7)$$

Eq. (7) can also be considered for constant heat flux boundary conditions. The fluid properties are taken corresponding to the fluid temperature in the above equations. A large number of experimental works have been reported in the literature for natural convection heat flow from flat upright as well as tilted plates to fluid for fixed heat surge boundary values. In these experimentations, outcomes have been expressed through modified Grashof number, Gr^* .

$$Nu_x^* = Gr_x Nu_x = \frac{g \beta q_w x^4}{k v^2} \quad (8)$$

Here, the unit of wall heat flux, q_w is taken as watt/m². For laminar flow conditions, the local Nusselt number is estimated out of the function.

$$Nu_{xf} = \frac{hx}{k_f} = 0.60(Gr_x^* Pr_f)^{1/5} \text{ for } 10^5 < Gr_x^* < 10^{11}; \quad (9)$$

Similarly, for turbulent flow conditions, the regional Nusselt number is calculated using the relation given below.

$$Nu_x = 0.17(Gr_x^* Pr)^{1/4} \text{ for } 2 \times 10^3 < Gr_x^* Pr < 10^{16}; \quad (10)$$

The local Nusselt number is calculated using the relation given below.

$$Nu_x = C(Gr_x Pr)^m \quad (11)$$

Substituting the value of Grashof number in Eq. (11), we have,

$$\text{Nu}_x = C^{1/(1+m)}(\text{Gr}_x^* \text{Pr})^{1/(1+m)} \quad (12)$$

For the case of laminar and turbulent surge conditions, m is considered as $1/4$ and $1/3$ correspondingly.

Churchill and Chu [19] have proposed that Eq. (12) can be customized as per the details given below so that it can be applied for the case of constant heat flux conditions.

$$\overline{\text{Nu}}_L^{1/4} [\overline{\text{Nu}}_L - 0.68] = \frac{0.67(\text{Gr}_L^*)}{[1 + (0.492/\text{Pr})^{9/16}]^{4/9}} \quad (13)$$

Where,

$$\overline{\text{Nu}}_L = q_w L / (k \overline{\Delta T}) \text{ and } \overline{\Delta T} = T_w (\text{at } L/2) - T_\infty \quad (14)$$

In Eq. (14), the average Nusselt number is expressed in terms of boundary heat energy surge conditions. Temperature distinction is taken at the mid-point of the plate *i.e.* at $x = L/2$.

II. THEORETICAL ASPECTS

A Theoretical study was carried out by using ANSYS Fluent 15.0 software. Following steps has been followed for theoretical analysis

- Create the geometry by using ANSYS Fluent 15.0 Design Modeler.
- After Domain generation, mesh is created, where number of nodes and number of elements are fixed. Finer meshes are provided in the region where the variations of fluid properties are expected to be more.
- Then under set up solution set up is done, where General set up, Models, Materials, Boundary conditions are developed.
- Then solution is initialization and solution of the governing equations is done.
- In post processing, the contours of temperature, velocity and pressures are developed.
- Various plots such as temperature plots and velocity plots are obtained.

Natural convection heat flow happens owing to the movement of fluid because of density differences produced by the heating processes. The force of buoyancy causes the movement of fluid in the upward direction as the density of fluid in the proximity of the heated plate is decreased due to heat transfer from the hot surface to the air. The buoyant force has an important role to play in natural convection flow. Fig. 1 shows the boundary layer and profiles of velocity and temperature over a heated vertical flat plate. In this boundary layer, the inertia force as well as the buoyant and viscous forces is predominant. It is observed from fig. 1 that the velocity happens to be zero at the walling to no slip boundary condition and it increases to attain maximum value and there after it declines reaches null value towards part of the edging film as the air is at rest away from the wall. Conversion from laminar to turbulent boundary layer commences after certain distance away from the leading edge which depends on the temperature of the wall and fluid properties. In

the transitional region turbulent eddies are formed. Farther up the vertical plate, fully turbulent boundary layer may be formed.

The velocity of flow in natural convection is very less in comparison to the velocity of flow in forced convection. Consequently, the value of the convection heat transfer coefficient is lower, generally by one order of magnitude. Grashof number, which is defined as the ratio of the buoyancy force to the viscous force in the free convection flow conditions has the same role which is played by the Reynolds number in forced convection. The factor that decides the conversion from laminar to turbulent flow is Grashof number. To analyze the heat-transfer problem, we must first obtain the differential equations of motion for the boundary layer. Uniform wall heat flux produces a natural convective flow with the fluid entering at the bottom and leaving at the top of the heated flat plate. The difference in the density of fluid which is caused because of the temperature gradients produces the buoyant forces which are responsible for causing the motion of fluid. The following assumptions have been considered for the computational analysis of natural convection flow of heat from the heated vertical flat plate.

- Fluid properties except density are considered to be constant.
- Density variations are significant only in the gravity term which is responsible for producing the buoyant force.
- The flow is considered to be steady, laminar, incompressible and inviscid.

The governing equations for 2-D, steady, in viscous flow is given below.

Continuity:

$$\left(u \frac{d\rho}{dx} + v \frac{d\rho}{dy}\right) + \left(\rho \frac{du}{dx} + \rho \frac{dv}{dy}\right) = 0 \quad (15)$$

Momentum:

$$\frac{\partial p}{\partial x} = -\left(\rho u \frac{\partial u}{\partial x} + \rho v \frac{\partial u}{\partial y}\right) \quad (16)$$

$$\frac{\partial p}{\partial y} = -\left(\rho u \frac{\partial v}{\partial x} + \rho v \frac{\partial v}{\partial y}\right) \quad (17)$$

Energy:

$$\rho c_v \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y}\right) = -\left(\frac{\partial q_x}{\partial x} + \frac{\partial q_y}{\partial y}\right) - T \left(\frac{\partial \rho}{\partial T}\right)_p \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y}\right) \quad (18)$$

III. RESULTS AND DISCUSSION

Utilized theoretically to explore via excited plates maintained upright and parallel, In exact two upright parallel plates, excited by means of heating coils using electric energy accomplished at the outer region to ensure steady heat surge status at the boundary. The dimensions considered for the plates are 500 mm, 150 mm, 5mm for the designation of length, breadth and thickness respectively. Again here the outside boundary is kept conduction restrict and consequently permitting the surge of heat energy commencing at inside region is the ambient and heat flux equal heat energy surge The heat flux at the wall is

maintained at $q'' = 2188 \text{ W/m}^2$. The temperature of the wall is measured experimentally using thermocouples at different locations on the inner face of the vertical plate. The hypothetical consequence intended for partition temperature in addition to investigational data for heat flux of 2188 W/m^2 are compared with each other as demonstrated in the Figure 3. It is observed from fig. 3 that the wall temperature goes on increasing along the length of the plate from bottom to top for uniform heat flux conditions at the wall. It is also evident from fig. 3 so as to hypothetical consequence goes with differ methodically through the investigational consequence. The divergence concern to hypothetical consequence out of the investigational consequence found practical within $\pm 10 \%$.

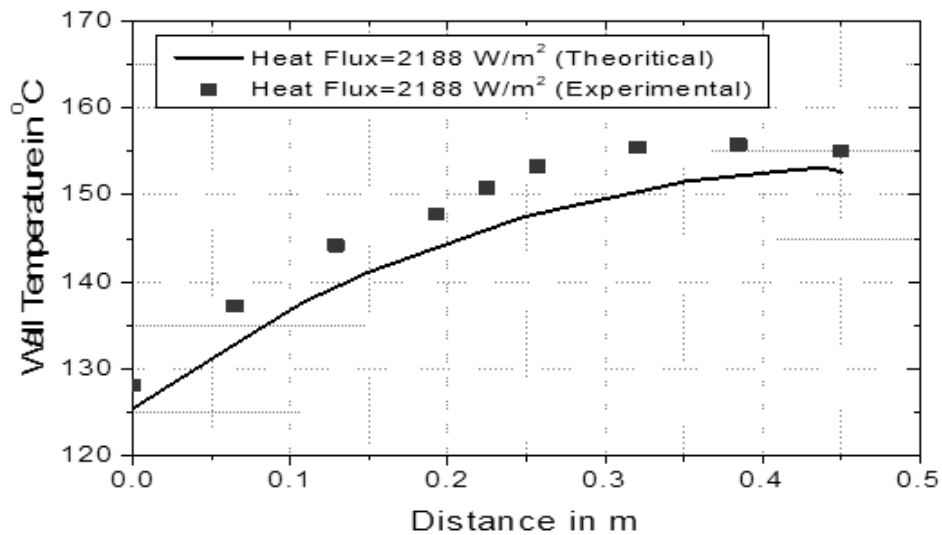


Figure 4: Theoretical and experimental wall temperature (heat flux 2188 W/m^2)

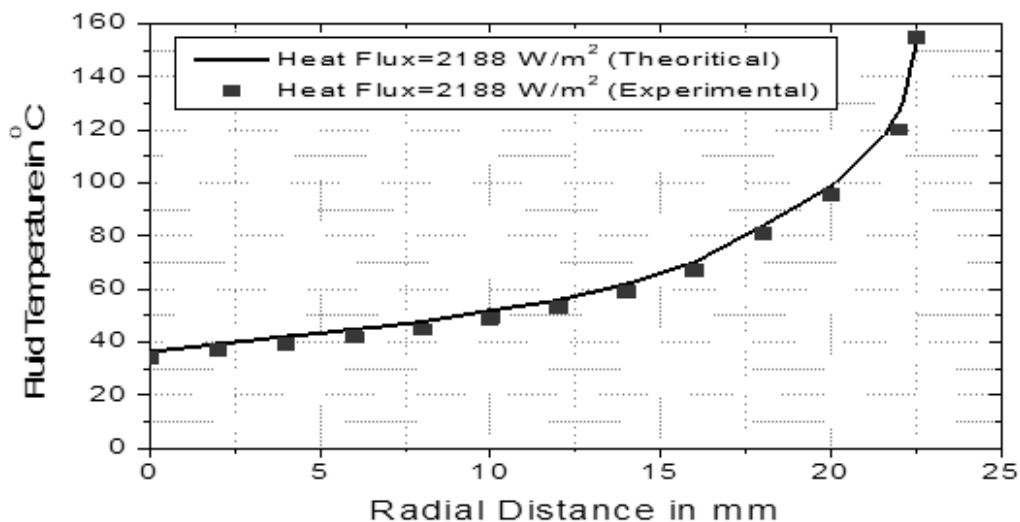


Figure 5: Theoretical and experimental fluid temperature (heat flux 2188 W/m^2)

Similarly, the temperature of air at the top of the vertical plates is measured experimentally and theoretically. The temperature of air at the middle of the two parallel plates is found to be minimum and it increases towards the wall in the lateral direction. The fluid temperature is observed to be maximum near the wall. The hypothetical consequence

intended for flowing hotness calculated fundamentally near out of plate is assessed by means of experimental consequence for the heat energy surge of value 2188 W/m^2 as illustrated in the Fig. 4. The hypothetical consequence nearly confirms to the observational fact which has established by the graphical presentation. Ultimately the divergence of both the magnitudes i.e. hypothetical as well as practical remains within $\pm 10\%$.

Air temperature at the middle of the two parallel plates was observed and it found minimum at the top and it increases gradually towards the walls it also found the air temperature near the plates is vary and it indicate maximum.

IV. CONCLUSIONS

The natural convection heat transfer through two heated vertical flat plates has been investigated theoretically as well as experimentally. The major conclusions drawn from the investigation can be enumerated as follows.

- The wall temperature rise along the length of the plates from bottom to top to maintain a uniform heat flux conditions at the wall.
- The temperature of air at the middle of the two parallel plates is found to be minimum and it increases radially towards the wall. The fluid temperature is found to be maximum near the wall.
- The theoretical results are found to be very close to the experimental result with $\pm 10\%$ error.

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