Shifting Of Heat Energy Via Transverse Parallel Sheets Confirmed With Natural Convection

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**Abstract**

Due to the advancement in the field of technology the aspect of 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 of natural convection heat transfer upon the open ended 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 by means of electrical heating coils on the outer surface 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/m2.consequent to this a definite analytical as well as observational values adhering with steady state phenomena 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/m2.

*Keywords - Heat flux, natural convection, heat transfer, temperature, CFD.*

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**Introduction:**

Heat transfer through natural convection is an economical method of cooling which is widely accepted by electronic industry in order to dissipate heat from the electronic devices which are entrenched on a small base in order to improve their life span and efficiency. Heat transfer denotes the transmission of heat energy happening from the heated plate to air due to temperature difference between them. Hot bodies can be cooled faster by forced convection than by natural convection. Natural convection occurs when heat transfer takes place because of the variation of density of fluid resulting from the temperature difference. The 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 finds application in number of industries such as nuclear reactors, steam pipe lines in power plants, cooling towers, domestic convectors, and thermo siphons etc. The objective of this investigation is to explore both theoretically and experimentally the natural convection through two heated vertical plates. 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 test section is heated electrically maintaining constant heat flux on the wall. 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. , and 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] addressed the issue of optimum plate spacing for enhancement of laminar natural convection heat transfer between two plates. Roul and Nayak [10] presented heat flow characteristics through heated vertical tubes. Nayak *et al.* [11-13] investigated the improvement of heat transfer in vertical 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 toconstant 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.

Turbulent

Laminar

T∞

TW

u

(a)

TW

u

T

T∞

T, u

 δ

(b)

Fig. 1.(a) Boundary layer and (b) Velocity and Temperature profileon a vertical plate

The average heat transfer coefficients can be presented for different environments as per the details given below:

$\overbar{Nu\_{f}}=C\left(Gr\_{f}Pr\_{f}\right)^{m}$ (1)

The subscript f specifies the fluid properties in the dimensionless groups. The fluid temperature is evaluated by:

$T\_{f}=\frac{T\_{w}+T\_{\infty }}{2}$ (2)

The Rayleigh number is defined as the product of Grashof number and Prandtl number:

Ra = Gr.Pr (3)

The characteristic length taken in the determination of Grashof and Nusselt number depends on the geometry considered in the problem. The plate height(L) is considered as the characteristic dimension for flat vertical plates and the diameter (D) is taken as the characteristic dimensionfor horizontal cylinders. The equations used to determine the heat transfer for vertical flat plates can also be considered for vertical cylinders when the boundary layer thickness is not very large in comparison to the diameter of cylinder.

500 mm

Fig. 2 Heated vertical parallel plates

As a matter of fact, a vertical cylinder is generally treated as vertical flat plates under the following conditions.

$\frac{D}{L}\geq \frac{35}{GR\_{L}^{^{1}/\_{4}}}$ (4)

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

$Nu\_{f}=0.10\left(Nu\_{f}Pr\_{f}\right)^{^{1}/\_{3}}$ (5)

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

$\overbar{Nu\_{f}}=0.68+\frac{0.670Ra^{^{1}/\_{4}}}{\left[1+\left(0.492/Pr\right)^{^{9}/\_{16}}\right]^{^{4}/\_{9}}} for Ra\_{L}<10^{9} $(6)

$\overbar{Nu\_{f}}^{^{1}/\_{2}}=0.825+\frac{0.387Ra^{^{1}/\_{6}}}{\left[1+\left(0.492/Pr\right)^{^{9}/\_{16}}\right]^{^{8}/\_{27}}} for 10^{-1}<Ra\_{L}<10^{12}$ (7)

Eq. (7) can also be considered for constant heat flux boundary conditions. The fluid properties is 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 vertical and inclined plates to fluid for constant heat flux boundary conditions. In these experimentations, the results have been presented in terms of modified Grashof number, Gr\*.

$Nu\_{x}^{\*}=Gr\_{x}Nu\_{x}=\frac{gβq\_{w}x^{4}}{kv^{2}}$ (8)

Here, the unit of wall heat flux, qw is taken as watt/m2. For laminar flow conditions, the local Nusselt number is calculated from the relation.

$Nu\_{xf}=\frac{hx}{k\_{f}}=0.60\left(Gr\_{x}^{\*}Pr\_{f}\right)^{^{1}/\_{5}} for 10^{5}<G\_{x}^{\*}<10^{11};$ (9)

Similarly, for turbulent flow conditions, the local Nusselt numberiscalculated using the relation given below.

$Nu\_{x}=0.17\left(Gr\_{x}^{\*}Pr\right)^{^{1}/\_{4}} for 2×10^{3}<Gr\_{x}^{\*}Pr<10^{16};$ (10)

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

$Nu\_{x}=∁\left(Gr\_{x}Pr\right)^{m}$ (11)

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

$Nu\_{x}=∁^{^{1}/\_{(1+m)}}\left(Gr\_{x}^{\*}Pr\right)^{^{1}/\_{(1+m)}}$ (12)

For the case of laminar and turbulent flow conditions, m is taken as 1/4 and 1/3 respectively.

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

$\overbar{Nu\_{L}}^{^{1}/\_{4}}\left[\overbar{Nu\_{L}} -0.68\right]=\frac{0.67\left(Gr\_{L}^{\*}\right)}{\left[1+\left(0.492/Pr\right)^{^{9}/\_{16}}\right]^{^{4}/\_{9}}}$ (13)

Where,

$\overbar{Nu\_{L}}=q\_{w}L/\left(k\overbar{∆T}\right) and \overbar{∆T} =T\_{w}\left(at L/2\right)-T\_{\infty }$ (14)

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

**Theoretical Aspects:**

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 are predominant. It is observed from fig. 1 that the velocity happens to be zero at the wallowing to no slip boundary condition and it increases to attain maximum value and there after it declines to zero towards the side of the boundary layer 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 is 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.

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

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

Continuity:

$\left(u\frac{dρ}{dx}+v\frac{dρ}{dy}\right)+\left(ρ\frac{du}{dx}+ρ\frac{dv}{dy}\right)=0$ (15)

Momentum:

$\frac{∂p}{∂x}=-\left(ρu\frac{∂u}{∂x}+ρv\frac{∂u}{∂y}\right)$ (16)

$\frac{∂p}{∂y}=-\left(ρu\frac{∂v}{∂x}+ρv\frac{∂v}{∂y}\right)$ (17)

Energy:

$ρc\_{v}\left(u\frac{∂T}{∂x}+v\frac{∂T}{∂y}\right)=-\left(\frac{∂q\_{x}}{∂x}+\frac{∂q\_{y}}{∂y}\right)-T\left(\frac{∂ρ}{∂T}\right)\_{p}\left(\frac{∂u}{∂x}+\frac{∂v}{∂y}\right)$ (18)

**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 500mm,150mm,5mm for the designationof 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 W/m2. The temperature of the wall is measured experimentally using thermocouples at different locations on the inner face of the vertical plate. The theoretical result for wall temperature and experimental data for heat flux of 2188 W/m2are 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 that the theoretical result matches vary thoroughly with the experimental result. The deviation of the theoretical result from the experimental result is observed to be within $\pm $10 %.



Fig. 3 Theoretical and experimental wall temperature (heat flux 2188 W/m2)

Fig. 4Theoretical and experimental fluid temperature (heat flux 2188 W/m2)

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 theoretical result for fluid temperature measured radically at the exit of plate is compared with experimental result for the heat flux 2188 W/m2 as illustrated in the Fig. 4. It is evident from figure that the theoretical result closely matches with the experimental data. The deviation of the theoretical result from the experimental result is observed to be within $\pm $10%.

Air temperature at the middle of the two parallel plates was observed to be minimum at the top and it increases gradually towards the walls. The temperature of air very near the plates is found to be maximum.

**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.

1. The wall temperature rise along the length of the plates from bottom to top to maintain a uniform heat flux conditions at the wall.
2. 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.
3. The theoretical results are found to be very close to the experimental result with $\pm 10\%$ error.

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