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# NUMERICAL STUDY ON HEAT TRANSFER CHARACTERISTICS WITH SQUARE HEATED OBJECT IN ANENCLOSURE

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***Abstract-*** *This paper deals with the numerical study on forced convection in a vented enclosure by finite difference method. An external fluid flow enters into the enclosure through an opening in the left bottom wall and exit from opening in right top wall. The significant parameters are Reynolds number, Prandtl number,outlet dimension andclearance of heated object from bottom wall. The two-dimensional mathematical model includes the system of four partial differential equations of continuity, momentum equation, Navier stokes and energy equation, solved by finite difference method. Flow fields are investigated by numerical simulation for air flowing with different Reynolds number and Prandtl number. Successive under relaxation (SUR) method is employed. There are ten variations in clearance C from 0.1 to 1. Reynolds number varied from 10 to 300, Prandtl number varied from 0.1 to 10. The results represented that circulation zone increases with higher.Localnusselt number increase with the increase of Re and increase of Pr but decrease along with position point forwarded. Heating efficiency also increase with increase of Re and Pr. Change in outlet dimension affects Nusselt number, heating efficiency and strength of circulation zone. The influence of clearance between the blunt body and bottom wall has also been demonstrated.*

**Keywords:** Forced convection, Finite difference method, Effects of Re and Pr number, Outlet dimension, Clearance.

## 1. INTRODUCTION

Forced convection flow phenomena occur in a wide range of engineering application like solar collectors and similar industrial applications. Especially, heat transfer from discrete heat sources in relation to cooling of electronic equipment’s has become a subject of interest in the last decade because of advances in the electronic industry. It is a challenging problem in heat exchangers, cooling of electronic devices, nuclear reactors and other thermal devices. The flow of these devices is often laminar because the passages are frequently narrow and the velocity of flow is typically low.

We examined the change of the heat transfer characteristics and its mechanisms with the clearance length and outlet length.The problem is solved using Navier-Stokes equations as a governing equation with the help of numerical method. A uniform grid system of 100X60 points has been used in the computation domainwhich provides a mesh of 6000 nodes.

The pressure term is eliminated from the Navier-Stokes equations to simplify the problem. The mathematical problem that represents the system is reduced to a simple from where the stream function and the velocity formulation are the base.

An external fluid flow enters into the enclosure through an opening at the left bottom side wall and exit from another fixed opening in the top of the right side wall. There is a clearance between bottom wall and heat source. The inlet and outlet are same for this problem.

For a real fluid, experimental evidence shows that the velocity of the layer adjacent to the surface is zero. This means that a velocity profile must show a velocity of zero at the boundary. When fluid flows inside an enclosure, temperatures of the fluid and the solid surfaces are different, then heat transfer between the fluid and the solid surface takes place by convection heat transfer mechanism.

In the present study, the heat transfer characteristics is discussed corresponding to the flow behavior varying the clearance length between heated body and wall surface, for obtaining information on how the laminar promoter may be close to the wall surface from a heat transfer enhancement perspective.

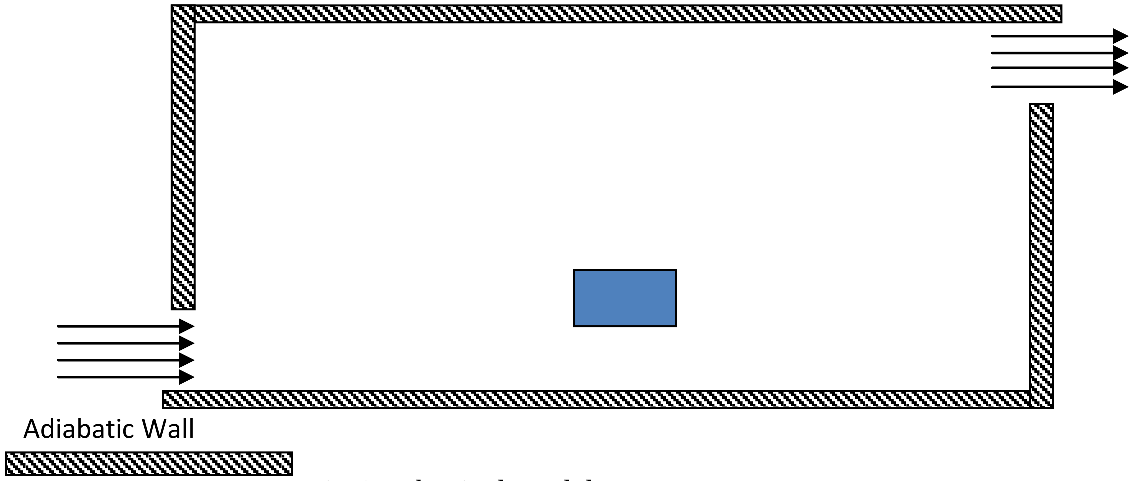


Fig 1: Physical model

KenyuOyakawa, YasuakiKourogi, Minoru Yaga [1] observed the thermal performance in a duct with a blunt body close to its wall. They have shown the effects of clearance length between a body and a duct wall, and duct height on heat transfer characteristics. N.M. Brown, F.C.Lai [2] numerically examined combined heat and mass transfer from a horizontal channel with an open cavity heated from below. Air was the fluid considered (Pr = 0.7). The main focus of their study was mass- transfer driven flows. The governing parameters were the buyouncy ratio N, Lewis number Le, Reynolds number Re and Grashoff's number Gr. Based on the scale analysis correlation for the entire convection regime, from natural, mixed to forced convection was proposed.

Bilgen and Yamane[3] examined the effect of conjugate heat transfer laminar natural convection and conduction in two dimensional rectangular enclosures with openings. They investigated the effects of various geometrical parameters and the thickness of the insulation layer on the fluid flow and heat transfer characteristics.Zhao et al. [4] numerically investigated conjugated natural convection in enclosures with external and internal heat sources and Xu et al. [3] experimentally observed the thermal flow around a square obstruction on a vertical wall in a differentially heated cavity. Type author's affiliation information 10-pt., in upper and lowercase letters with a single line spacing. Skip single line-space between the authors' line and first affiliation line. FaridBabadi, BijanFarhanieh [5] developed a physical model and corresponding calculation procedure have been for the combined heat and mass transfer velocity field have been assumed on the basis of Nusselt and continuity equations. The energy and diffusion equations have been solved simultaneously with an equilibrium boundary condition at the vapor–liquid interface.The results can be used as a reference in designing actual absorption chiller, heat pump and heat transfer apparatuses.

MatsuhaSoichi, SenahaIzuru [6] measured forced convection heat transfer, relationships between heat transfer and flow behaviour were experimentally investigated when the longitudinal vortex was artificially generated by single vortex generator in a pipe. They found that the local heat transfer co-efficients was mainly increases depending on the circumferential velocity component.

Yagi Minoru, OyakawaKen'yu [7] studied the effects of the longitudinal vortex which was generated by a triangular plate inserted in a pipe on the heat transfer enhancement. The longitudinal vortex moved spirally downstream. The local heat transfer distribution was depended on the longitudinal vortex.

## 2. MATHEMATICAL FORMULATION

### 2.1 Approximations

The flow is considered two dimensional, steady state, incompressible, laminar and all physical properties are assumed constant through the cavity with a heated rib at the bottom wall. The body force and viscous dissipation are neglected.

### 2.2 Numerical Modeling

Continuity equation -

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial x} = 0 \ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

Navier- Stokes equation –

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial x} = -\frac{1}{\rho}\frac{\partial p}{\partial x} + \nu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right)\ldots\ldots(2)$$

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial x} = -\frac{1}{\rho}\frac{\partial p}{\partial y} + \nu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right)\ldots\ldots(3)$$

Energy equation –

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial x} = \frac{K}{\rho C_p}\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right)\ldots\ldots\ldots(4)$$

Now to make this equation in non dimensional form, the width of inlet opening is considered as the characteristics length and initial pressure as the characteristic pressure, maximum velocity at the inlet is the characteristics velocity.

$$X = \frac{x}{H};\ Y = \frac{y}{H};\ U^{*} = \frac{u}{u_{max}};\ V^{*} = \frac{v}{u_{max}};\ P = \frac{p}{\frac{1}{2}\rho u_{max}^{2}};$$

$$\theta = \frac{T - T_{\infty}}{T_{0} - T_{\infty}};$$

Normal forms of these equations are as follows:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \ (5)$$

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{1}{2}\frac{\partial p}{\partial x^{2}} + \frac{1}{Re}\left(\frac{\partial^{2} u}{\partial x^{2}} + \frac{\partial^{2} u}{\partial y^{2}}\right) \ldots\ldots (6)$$

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{1}{2}\frac{\partial p}{\partial x^{2}} + \frac{1}{Re}\left(\frac{\partial^{2} v}{\partial x^{2}} + \frac{\partial^{2} v}{\partial y^{2}}\right) \ldots\ldots \ (7)$$

$$u\frac{\partial \theta}{\partial X} + v\frac{\partial \theta}{\partial Y} = \frac{1}{Re.Pr}\left(\frac{\partial^{2} \theta}{\partial x^{2}} + \frac{\partial^{2} \theta}{\partial y^{2}}\right) \ldots\ldots\ldots\ldots\ldots \ (8)$$

## 3. RESULTS AND DISCUSSION

A uniform grid system of 100×60 nodes have been chosen in the computational domain which provides a mesh of 6000 nodes.More fine grid than specified will not calculate accurate result.

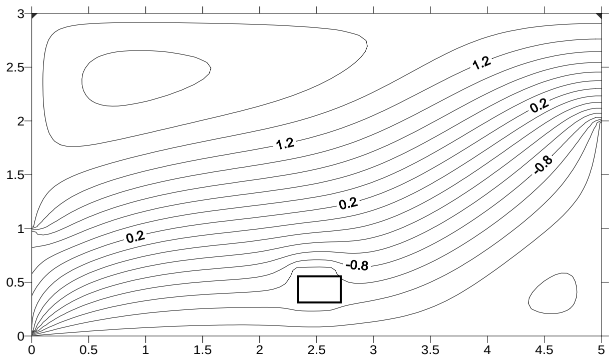

Fig 2: stream function at Re=160, Pr=0.71

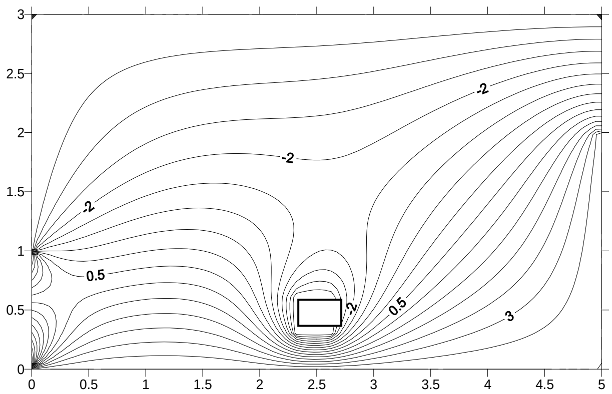

Fig 3: Vortices at Re=160, Pr=0.71

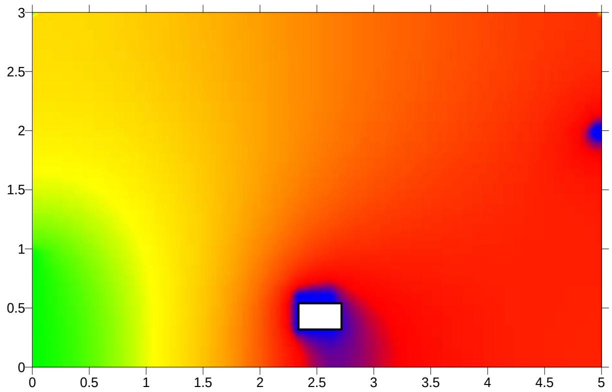

Fig 4: color contour at Re=160, Pr=0.71

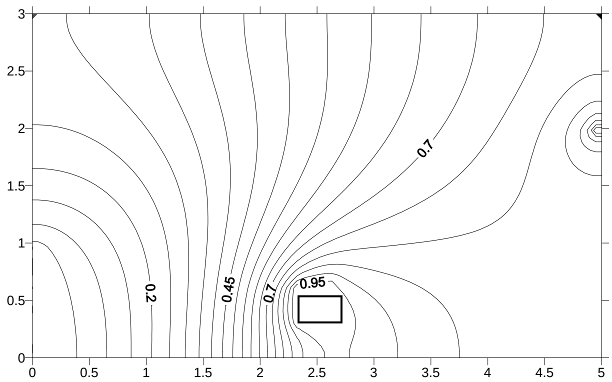

Fig 5: Isotherm at Re=160,Pr=0.71

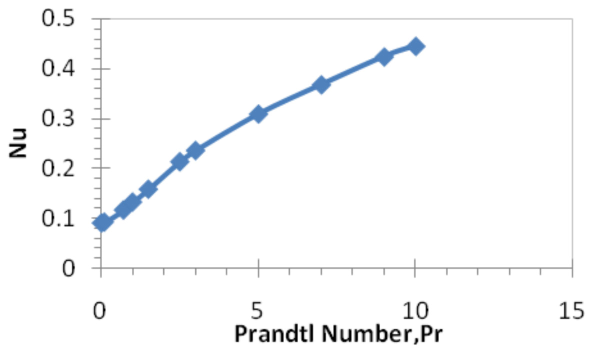


Fig 6:Relation between average Nusselt Number &Pr at Re=80,Clearance,C=0.3

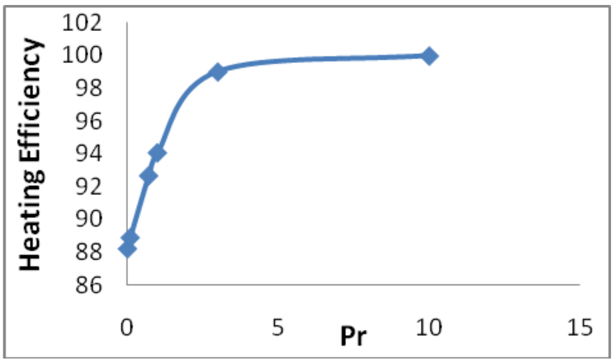


Fig 7: Relation between Pr and heating efficiency at Re=80, Clearance,C=0.3.

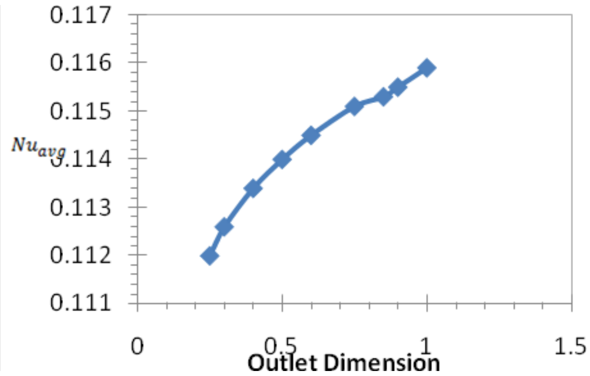


Fig 8: Relation between average Nu & outlet dimension at Pr=0.71, Re=80, Clearance,C=0.3

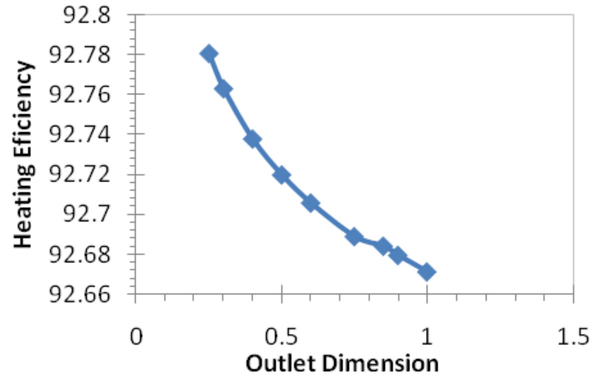


Fig 9:Relation between heating efficiency & outlet dimension at Pr=0.71, Re=80, C=0.3

Numerical results have been presented in order to determine the effects of presence of dimensionless parameters in a rectangular enclosure. The dimensionless governing parameters that must be specified for the systems are Reynolds number (Re), Prandlt number (Pr) and the physical parameters in the system is the clearance between blunt heat source and bottom wall and the various dimensions of outlet.

**4.1 Effects of Reynolds Number on Heat Transfer Characteristics**

From graphical representation in Fig 2 and fig 3, it is shown that thecirculation zone becomes prominent with higher Re i.e. Re=160 with Pr=0.71 for both velocity vector and stream function. For higher Re number circulation zones take place in conjugate corners.In the colour contour of temperature distribution and isotherms from Fig 4 at Re=160 and Pr=0.71, cold fluid spread more and bottom corner of the outlet is being heated more with increasing Re number. Also isotherm lines concentrate more at bottom corner of the outlet. Local Nu number increases with increasing Re number. Heating efficiency also increase with increase of Re number.

**4.2 Effects of Prandtl Number on Heat Transfer Characteristics**

Fig 6 shows that the local Nusselt number increases along with Prandtl number and most of the fluid remain cold which enter through inlet in the domain of enclosure. Fig 7 shows that heating efficiency also increase with Pr number at Re=80 and clearance 0.3.

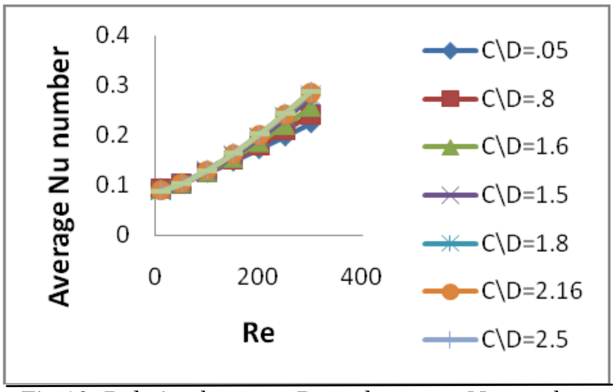


Fig 10: Relation between Re and average Nu number for various clearance at Pr=0.71.

**4.3 Effects of Outlet Dimensionon Heat Transfer Characteristics**

Fig 8 and Fig 9 shows that, with increasing outlet dimension, Nu number increases and heating efficiency decreases at Pr=0.71, Re=80, Clearance,C=0.3. For higher Re number, as the outlet dimension decrease, recirculation zone of two corner is increased in case of velocity vector and stream function. In colourcontour of temperature distribution, outlet temperature in increased with decrease of outlet dimension.There is no specific change of local Nu number for the change of outlet dimension.

**4.4 Effects of Clearanceon Heat Transfer Characteristics**

From Fig 10, it has been seen that for low Re, Nu deceases with increasing clearance between the blunt body and bottom wall. But for higher Re, this phenomenon does not occur. It has been done with Pr=0.71. As the clearance increased of heated object from the bottom wall, temperature of the outlet bottom corner increased.

**5. CONCLUSION**

In many modern buildings, mechanical ventilation is provided as a means of room load removal and provision of good indoor air quality. Two kinds of convection are involved in a ventilated room, i.e., internal buoyancy-induced natural convection by the discrete heat sources and external mechanical-driven forced convection by the ventilation.In this study, a finite difference method for steady state incompressible forced convection has been developed for laminar forced convection flow in a rectangular enclosure with adiabatic side walls and a blunt heated object. The difference equation has been derived from the governing equation that consists of conservation of mass, momentum and energy equation.

1. Higher circulation zones formed at the conjugate corner of left wall and top wall. And small circulation zones formed at the conjugate corner of bottom wall and right wall.
2. Heating efficiency and average Nu number both increases with the increase in Re.

3.Heating efficiency and average Nu number both increases with the increase in Pr.
4. Strength of circulation increases with increasing Re.
5. Average Nu number decreases with the increase of clearance for low Re ($0 \leq Re \leq 100$).
6. Heating efficiency decreases and average Nu number increases with the increase of Re.

In future, performance evolution can be done by considering following recommendations:

1. Arrangement of forced convection with heat source with the inlet at upper side and centre of the left wall of the enclosure.
2. Arrangement of forced convection with heat source using two inlet and one outlet in the enclosure.
3. Arrangement of forced convection with heat source using one inlet and two outlet in the enclosure.
4. Arrangement of forced convection with heat source using inlet at the upper side and outlet in the lower side of the left wall along with the blunt object adjacent to the right wall of the enclosure.

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## 8. NOMENCLATURE

| Symbols | Description |
|---|---|
| D | Height of the Block |
| c | Clearance between wall and Block |
| C | c/D |
| H | Height of the Domain |
| L | Length of the Domain |
| $H_{obj}$ | Filament Height |
| k | Thermal conductivity of the fluid |
| Nu | Nusselt Number |
| p | Pressure of the flowing fluid |
| Pr | Prandtl Number, $Pr = \frac{\nu}{\alpha}$ |
| Gr | Grashof number, $Gr = \frac{g\beta\Delta\theta W^3}{\nu^2}$ |
| T(x,y) | Local fluid temperature |
| $U$ , $V$ | Dimensionless velocity components, $U = \frac{uW}{\alpha}$, $V = \frac{vW}{\alpha}$ |
| x, y | Cartesian coordinates |
| X, Y | Dimensionless Cartesian coordinates, $\frac{x}{W}, \frac{y}{W}$ |
| W | Inlet and outlet width |
| ρ | Density of fluid |
| ν | Kinematics viscosity of fluid |
| ω | Vorticity |
| ω* | Dimensionless Vorticity, $\frac{\omega}{\omega_o}$ |
| θ | $= \frac{T - T_\infty}{T_{obj} - T_\infty}$ |
| ψ | Stream function |
| $\psi_o$ | Initial value of Stream function |
| ψ* | Dimensionless Stream function, $\frac{\psi}{\psi_o}$ |