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ARTICLE

Convection Heat Transfer from Heated Thin Cylinders Inside a Ventilated Enclosure

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ARTICLE INFO

Article history
Received:16 May 2022
Revised: 20 June 2022
Accepted: 29 June 2022
Published Online: 15 July 2022

Keywords:
Experimentation
Ventilated enclosures
Convection heat transfer
Dimensionless data

ABSTRACT

Experimental study was conducted to determine the effect of velocity of axial fan, outlet vent height, position, area, and aspect ratio (h/w) of ventilated enclosure on convection heat transfer. Rectangular wooden ventilated enclosure having top and front transparent wall was made up of Perspex for visualization, and internal physical dimensions of box were 200 mm × 200 mm × 400 mm. Inlet vent was at bottom while outlet vents were at the side and top wall. Electrically heated cylindrical heat source having 6.1 slenderness ratio was fabricated and hanged at the centre of the enclosure. To calculate heat transfer rates, thermocouples were attached to the inner surface of heat source with silica gel. Heat source was operated at constant heat flux in order to quantify the effect of velocity of air on heat transfer. It was observed that average Nusselt number was increased from 68 to 216 by changing velocity from 0 to 3.34 m/s at constant modified Grashof number i.e. 5.67E+09. While variation in outlet height at the front wall did not affect heat transfer in forced convection region. However, Nusselt number decreased to 5% by changing the outlet position from top to the front wall or by 50% reduction in outlet area during forced convection. Mean rise in temperature of enclosure increased from 8.19 K to 9.40 K by increasing aspect ratio of enclosure from 1.5 to 2 by operating heat source at constant heat flux i.e. 541.20 w/m^2.

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DOI: https://doi.org/10.30564/ssid.v4i2.4719

1. Introduction

Convection heat transfer from cylindrical heated surfaces inside the ventilated enclosure is of practical importance related to designing of enclosure of electronic equipments and ventilation of building. From design point of view, aspect ratio of enclosure, position and area of vents, velocity of fan in case of forced convection are critical variables. These parameters should be opted in such a way that it yields in maximum cooling and optimum electrical energy consumption. Due to recent development in electronics industry, reliability of system is hugely dependent on working of its electronic component. Literature shows that main cause of electronic equipment failure is rise in temperature [1].

Ali Riaz et al. studied effect of heat source height inside the ventilated enclosure on mean rise in temperature numerically. This study suggested that heat source should be at minimum height relative to the base of enclosure or at maximum height relative to the outlet position of the enclosure to achieve maximum cooling [2,3]. Basavaraj Kusammanavar et al. numerically studied the effect of heat source position on natural convection in a square cavity. Two dimensional simulations were performed in ANSYS Fluent to expedite a problem. Main finding of the study is that heat transfer increases by splitting heat source [4]. Chavan S. and Sathe A. performed three dimensional numerical simulations to study the impact of inter board spacing and heat dissipation rate on heat transfer and flow field inside the enclosure. Through proper placement of electronic component temperature of enclosure can be reduced from 3 ºC to 5 ºC [5].

Fahad Sadekin Zaman et al. studied velocity and temperature fields inside the enclosure having multiple discrete heat sources [6]. V.C. Mariani and L.S. Coelho studied effect of aspect ratio of open enclosure and temperature difference of vertical walls and internal heat source intensity on Nusselt number [7]. Luiz Joaquim C.R. and Henor Arthur S. studied influence of internal heat source in naturally ventilated offices. They found that cross ventilation is more useful as compared to unilateral [8]. Masaru I. and Shinji N. experimentally studied effect of outlet area and height on natural convection inside the ventilated enclosure. To accomplish this study, they fabricated ventilate enclosure having dimensions: 220 mm × 230 mm × 310 mm, and heat source. At the end, results are reported in the form of dimensionless number [9].

Norma Alejandra R.M. et al. modelled room as ventilated cavity and performed numerical simulations on ANSYS Fluent 6.3 by applying k-ε turbulence model along with non-uniform grid. Results are reported in dimensionless form [10]. Satish K.A. and A.N. Mathur studied combined effect of forced and natural convection in a ventilated enclosure with different outlet vents arrangements. Surface temperature of bottom heated surface was recorded against different values of Grashof and Reynold number and outlet vent configuration [11]. Azhar Kareem Muhammad et al. performed experiments to determine heat transfer from cylindrical heated surface inside the ventilated enclosure. Experiments were performed for the range of Ra 3.47×10^3 to 5.66×10^4, aspect ratio $11 \leq L/D \leq 22$ and inclination angle of heat source 0º to 90º [12].

S.K. Ajmera and A.N. Mathur studied the effect of mixed (free and forced) convection heat transfer in ventilated enclosure by changing ventilation arrangements. For this, they performed numerical simulations by flush mounted heat source at bottom wall. Average heat source temperature at different value of Richardson number was recorded for different combination of vents [13]. E. Bilgen and A. Muftuoglu observed that optimum position of heat source inside the ventilated enclosure depends upon ventilation ports arrangements rather than Richardson number. Furthermore, they plotted Nusselt number as a function of Richardson number [14].

2. Motivation for the Present Research

Main purpose of manufacturing and fabricating indigenous experimental setup was to study the critical parameters of ventilated enclosure on convection heat transfer. Number of research studies are available on geometric effect in natural convection, while limited data is available in forced convection. This study can be used in designing of enclosures for electronics systems or ventilation of building. It can be noticed by studying literature that experimental data is limited and mostly numerical findings are available, therefore, this study is addition to experimental results archive. Furthermore, this could be used as a bench mark for numerical simulations and formulation of empirical correlation.

3. Energy Balance

Net heat generated from cylindrical heat source is equal to convection, radiation and heat loss from endcaps as shown in Equation (1).

$$Q_{total}=+Q_{convection}+Q_{radiation}+Q_{loss} \tag{1}$$

Radiation heat transfer can be calculated through Stephen Boltzman law, heat losses through Fourier law of heat conduction while convection heat transfer through Newton's law of cooling. Net heat generated was equal to the product of current and voltage.

4. Experimental Setup

Rectangular ventilated enclosure having internal dimensions: 200 mm × 200 mm × 400 mm, was manufactured. Cylindrical heat source having 6.10 slenderness ratio was hanged at the center of enclosure by strings. There were four outlet vents and one inlet vent in the ventilated enclosure. Position and size of each vent are given in mm as shown in Figure 1.

Axial fan was fixed at inlet that can be operated at maximum 12 volts. Furthermore, honey comb structure was used to remove the flow disturbances as shown in Figure 2. Complete experimental setup is shown in Figure 3. Thermocouples were installed inside the enclosure and to the inner surface of heat source to record mean rise in surface and enclosure temperature.

5. Uncertainty and Results Validation

After calibration of sensors, each experiment was repeated thrice to ensure the repeatability and to calculate the standard deviation. Maximum deviation in temperature measurement was ± 0.5 °C due to the least count of temperature measuring system. Additionally, maximum uncertainty in reporting dimensions i.e. heat source length, diameter and thermocouple locations is ± 2 mm. Experimental results are reported with 97% precision and average standard deviation of results from reported value is 1.14%. Before starting experiments, heat source was placed in air quiescent medium and results were compared with Al-Arabi and Khamis data [15]. Results were found satisfactory as maximum percentage difference was less than 5%.

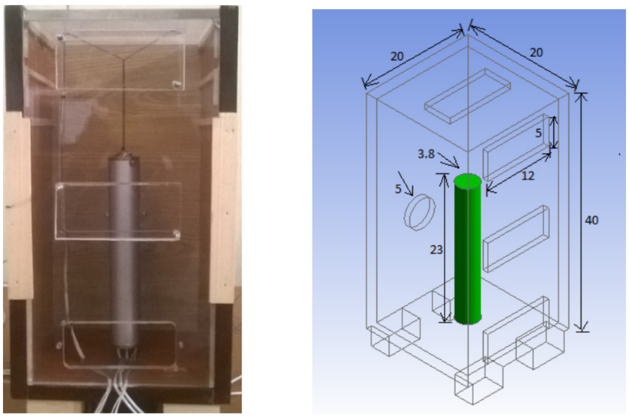

Figure 1. Manufactured and CAD model (left to right) of ventilated enclosure with dimensions (in mm)

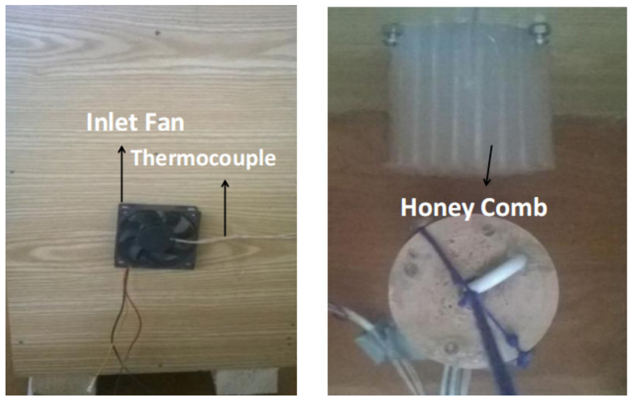

Figure 2. Inlet of ventilated enclosure (fan, honeycomb structure)

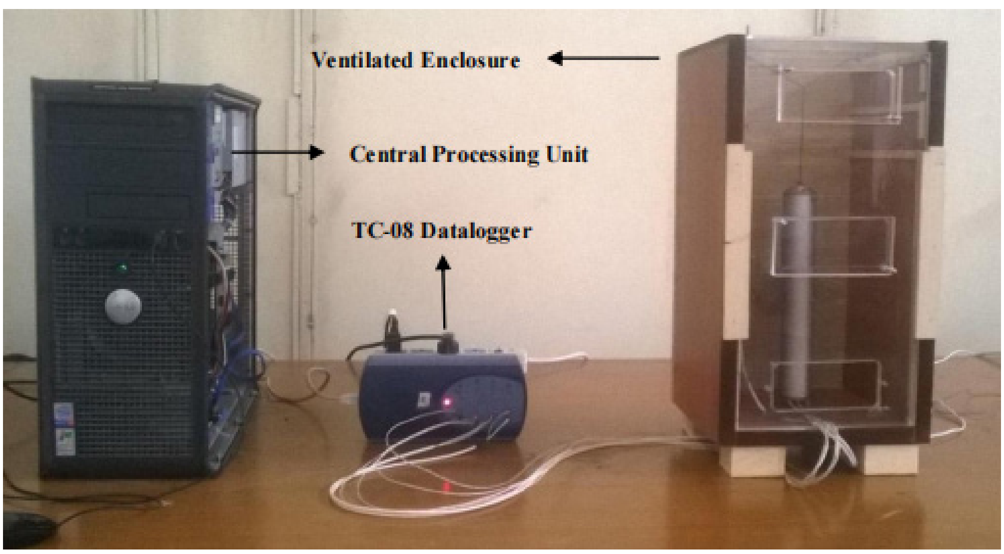

Figure 3. Experimental setup to study heat transfer inside the ventilated enclosure

6. Method and Measurements

After validation, heat source was hanged at the centre of the ventilated enclosure and operated at constant heat flux value while fan was operated at constant velocity. System took almost one hour to reach steady state. After reaching steady state, temperatures were recorded with the help of miniature K-type thermocouples and TC-08 data logger. Average heat transfer coefficient on the surface of heat source in various conditions was calculated through energy balance equations.

Total thirty five experiments were performed for the range of convective heat flux 236.76 w/m^2 to 976.08 w/m^2.

7. Results and Discussion

Effect of outlet area, height, position and fluid inlet velocity on convection heat transfer from the thin cylindrical heat source operated at various heat flux values inside the ventilated enclosure are discussed below.

8. Effect of Outlet Height, Area and Position

In order to study the effect of outlet height, three different outlet heights were used such as 45 mm (outlet:3), 200 mm (outlet:2) and 355 mm (outlet:1). Results were plotted in the form of average Nusselt number against the Richardson number as shown in Figure 4. It was observed that variation of outlet height at the front wall did not affect heat transfer in forced convection region. However, Nusselt number increased to 5% by changing the outlet position from front to the top wall as shown in Figure 5. This is because of placing the outlet at top wall of ventilated enclosure is assisting the natural convection flow.

By reducing the outlet area to 50%, Nusselt number decreases to 5% as shown in Figure 6. It was observed that volumetric flow rate of air increased by reducing the area but Nusselt number decreases due to resistance increment in the passage of air.

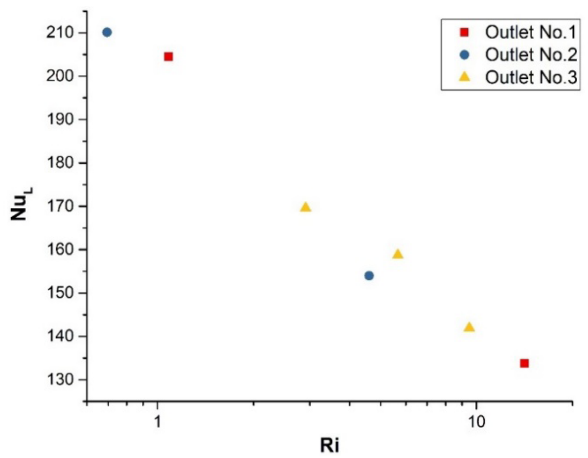

Figure 4. Effect of outlet height on convection heat transfer inside the ventilated enclosure having internal heating source

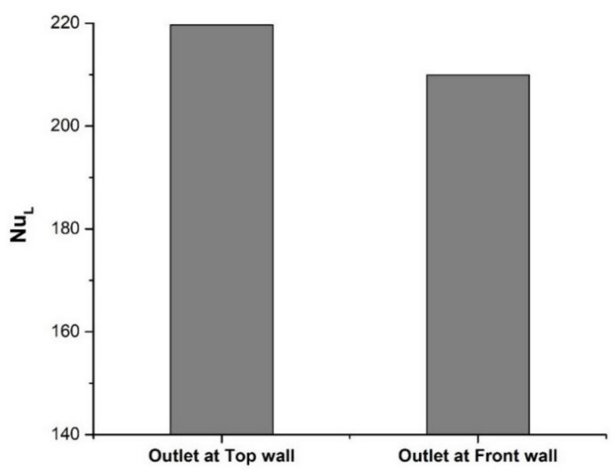

Figure 5. Effect of outlet position on convection heat transfer

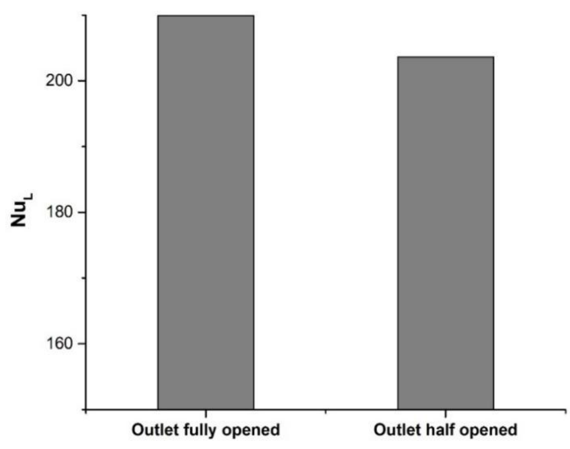

Figure 6. Effect of outlet area on convection heat transfer

9. Effect of Inlet Velocity

To study the effect of inlet velocity, heat source was operated at constant modified Grashof number i.e. 5.67E+09. Seven experiments were performed by varying velocity while outlet height, position, and heat flux were kept constant. It was observed that inlet velocity has a significant effect on convection heat transfer as shown in Figure 7. Figure 7 is divided into three sections i.e. forced, mixed and natural convection heat transfer. In mixed convection region, Nusselt number was varied from 120 to 160, but in forced convection there was a sharp increment in heat transfer and Nusselt number reached upto 216; while in natural convection region, Nusselt number suddenly decreased to 68. Trend of region "A" of Figure 7 was also observed in reference [16].

Maximum calculated heat losses were 4.1%. Almost one hour was required by the system to reach the steady state. Calculations showed that heat transfer coefficient was varied from 8.31 $w.m^{-2}.K^{-1}$ to 25.22 $w.m^{-2}.K^{-1}$ by changing inlet average velocity from 0 to 3.34 m/s.

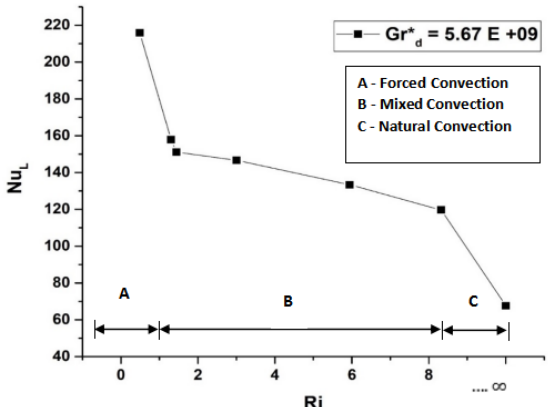

Figure 7. Effect of inlet velocity on convection heat transfer from cylindrical surface inside the ventilated enclosure

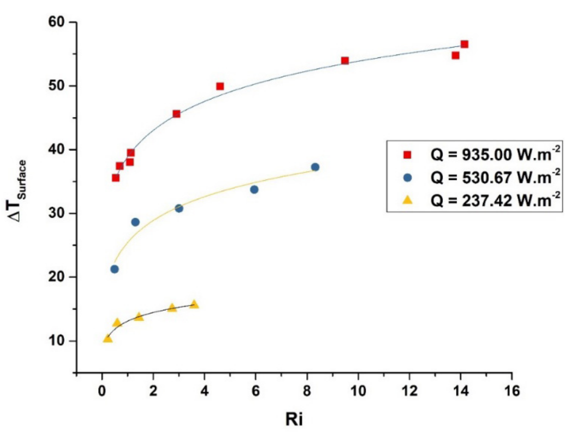

Figure 9. Mean rise in surface temperature of cylindrical heat source for three different values of heat flux

10. Mean Rise in Temperature

Mean rise in surface temperature of cylindrical heat source and air temperature inside the ventilated enclosure was measured against the Richardson number as shown in Figure 8 and Figure 9. For this, heat source was operated at three different values of heat flux such as 237.42 w/m^2, 530.67 w/m^2 and 935.00 w/m^2. Mean rise in temperature inside the enclosure and on the surface of heat source is important variable to consider before designing of ventilated enclosure. From the constraint of maximum allowable temperature of heat source, one can approximate the required velocity of air.

To observe the variation of mean rise in temperature inside the ventilated enclosure, five thermocouples were used. Thermocouples were placed at the distance of 50 mm from side wall and at the height of 50 mm, 100 mm, 150 mm, 200 mm and 250 mm with the help of string. It was observed that in forced convection, mean rise in temperature inside the ventilated enclosure was almost uniform. While in natural convection, mean rise in temperature increased from bottom to the top of the enclosure as shown in Figure 10.

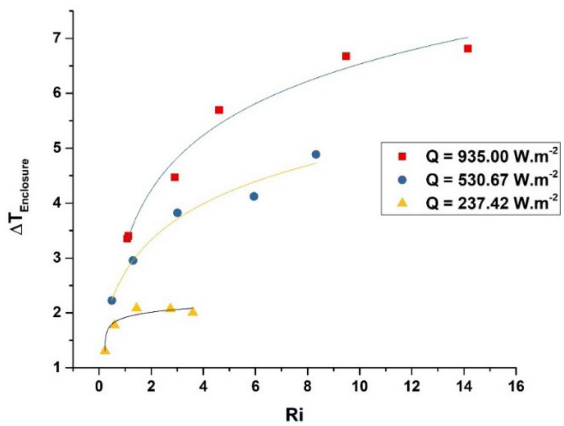

Figure 8. Mean rise in enclosure temperature for three different values of heat flux

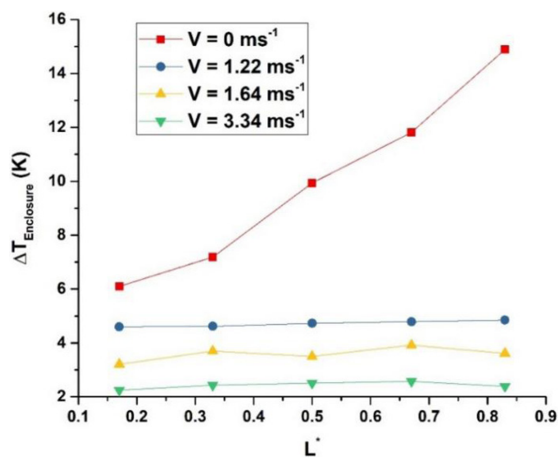

Figure 10. Variation of mean rise in temperature inside the air ventilated enclosure against the dimensionless height

11. Effect of Aspect Ratio

Aspect ratio (h/w) of ventilated enclosure was changed from 2 to 1.5 in order to study its effect on convection heat transfer. For this, two experiments were performed by operating inlet fan at 1.64 m/s and 0 m/s while heat source was operated at constant heat flux: 541.20 w/m^2. It was observed that surface temperature did not affect by changing aspect ratio, however mean rise in enclosure was affected by changing aspect ratio of enclosure under natural convection heat transfer as shown in Table 1. Heat transfer decreases by increase in aspect ratio, and same behaviour was observed by Vishnu C.S. and Anoop V. [17].

Table 1. Effect of aspect ratio on mean rise in temperature inside the enclosure

Convection Mode	Aspect Ratio = 2 $\Delta T_{Enclosure}$ (K)	Aspect Ratio = 1.5 $\Delta T_{Enclosure}$ (K)
Natural Convection	9.40	8.19
Forced Convection	3.82	3.62

12. Conclusions

Effect of axial fan velocity, aspect ratio of enclosure, outlet position, height and area on mixed convection is studied experimentally. It was observed that:

1) Average Nusselt number was increased from 68 to 216 by changing velocity from 0 to 3.34 m/s at constant modified Grashof number i.e. 5.67E+09.

2) Nusselt number decreased to 5% by changing the outlet position from top to the front wall or by 50% reduction in outlet area during forced convection.

3) Mean rise in temperature of enclosure increased from 8.19 K to 9.40 K by increasing aspect ratio of enclosure from 1.5 to 2 by operating heat source at constant heat flux i.e. 541.20 w/m^2.

Funding

This research was funded by Pakistan Institute of Engineering and Applied Sciences, Pakistan. It is hereby declared that authors have no conflict of interest of any kind.

Conflict of Interest

There is no conflict of interest.

Nomenclature

Gr^*	Modified Grashof number $Gr_L^* = \frac{g\beta L_c{}^4 q''}{kv^2}$
Nu_L	Average Nusselt number $Nu = \frac{hL_c}{k}$
Re	Reynold Number $Re = \frac{Inertia\,forces}{Viscous} = \frac{VL_c}{v} = \frac{\rho V L_c}{\mu}$
Ri	Richardson Number $Ri = \frac{Gr_L}{Re_L{}^2}$
Q_{total}	Total Heat Transfer $Q_{total} = Q_{in} = I \times V$
$Q_{convection}$	Convection Heat Transfer $Q_{convection} = h_{avg}\, A_s (T_{s,avg} - T_b)$
$Q_{radiated}$	Radiated Heat Transfer $Q_{radiated} = \varepsilon\, \sigma\, A_s\, (T_{s,avg}{}^4 - T_b{}^4)$
$Q_{conduction}$	Conduction Heat Transfer $Q_{conduction} = -kA\frac{dT}{dx}$
A_s	Surface Area
g	Gravitational acceleration
h	Convective heat transfer coefficient
I	Current
k	Thermal conductivity
L_c	Characteristic length
Pr	Prandtl number
Q	Heat source power
T_b	Ambient or bulk stream temperature
T_f	Film temperature
T_s	Surface temperature
V	Voltage
ρ	Density
μ	Dynamic viscosity
ε	Emissivity
v	Kinematic viscosity

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