# Convection in vertical annular gap effected by rotating outer cylinder and imposed axial flow at inlet

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## ABSTRACT *(12 pt)*

Experiments are conducted for convection in vertical annular gap effected by cold rotating outer cylinder and heated stationary inner cylinder with imposed axial flow at inlet. Experiments were conducted by varying the rotational speed of outer cylinder from 0 to 1000 rpm with a step size of 100 rpm, corresponding to rotation parameter in the range $0 \leq \zeta \leq 2812$. The axial flow is imposed by varying the inlet mass flow rate viz:1.108g/s, 2.0157g/s and 4.101g/s. The heat flux to the inner heated stationary cylinder, channel aspect ratio and radius ratio is kept fixed as, $q$ =80 W/m$^2$ , AR(d/H) = 38.75 η=0.614 respectively. The interdependence of rotation and imposed axial flow on the flow and thermal field in the annular gap is analyzed and discussed.

## NOMENCLATURE

A surface area of heated inner cylinder, m$^2$
$A_N$ Inlet cross sectional area of the nozzle m$^2$
d annular gap, $d_o$-$d_i$, m
g acceleration due to gravity, m/s$^2$
h average convective heat transfer coefficient, W/m$^2$K
Δh manometer head difference, mm
k thermal conductivity, W/mK
Δp pressure drop
I current, A
Q Heat load, W
T temperature, K
V Voltage, V
Gr Grashof number, $\frac{g\beta\Delta T d^3}{\nu^2}$

***Greek symbols***

η radius ratio, $\frac{r_i}{r_o}$
ζ rotation parameter, $\frac{Re_\omega^2}{Gr}$
ω angular velocity, rad/s
ρ density, kg/m$^3$
α thermal diffusivity, m$^2$/s
β isobaric cubic expansivity of fluid, 1/K
ν kinematic viscosity, m$^2$/s
σ standard deviation

***Subscripts***

*avg* average
*atm* atmospheric
*b* bulk
*f* fluid
*i* inner
*o* outer
∞ free stream

## INTRODUCTION

Convection and associated fluid dynamics in vertical annular gap between concentric cylinders received considerable attention due to their occurrence in wide range of engineering fields viz: rotating electrical motor, rotating heat exchangers, gas centrifuges and many others [1]. In electric motors studies by Vallenzuela and Tapia [2], Oualil et al. [3] have demonstrated the importance of convective heat transfer within the cylindrical gap(area separating stator and rotor). Flow and heat transfer in the annular gap in such types of geometries is quite complex due to the interaction of the swirl component of velocity induced by the centrifugal force due to the rotation either inner/outer cylinder with the axial and radial component of velocities. Experimental investigation of turbulent heat transfer in a large gap annulus with both rotating and non rotating inner cylinder and electrically heated outer cylinder was done by Kuzay et al. [4]. They reported that the rotation of the inner cylinder caused significant variation in the radial temperature profile for a large portion of the cross section of the annulus. A finite-difference scheme was proposed by Sarhan et. al.[5]

for solving boundary layer equations of laminar convection in open ended vertical concentric annulus of radius ratio 0.5 with rotating inner wall. It was reported by the authors that for high values of dimensionless volumetric flow rate, rotational effect causes fluid to move from regions close to the two boundaries to the core. But, for low values of volumetric flow rate the fluid from the region close to the adiabatic wall was observed to displace to the region close to the heated wall. Numerical results were presented by El-Shaarawi et al.[6] for the combined free and forced convection developing laminar boundary layer flow in a vertical concentric annulus with a rotating inner cylinder. In this study, they analyzed the effect of rotating inner cylinder on the hydrodynamic developing length, critical distance at which axial velocity gradient normal to the wall vanishes and the heat transfer performance. Ball et al. [7] conducted experiments to study interdependence between the heat transfer mechanism and the structure of the secondary flows in an annular gap between concentric vertical cylinders. The heated and rotating inner cylinder was aligned concentrically with stationary and cooled outer cylinder. They elucidated the qualitative aspects of the mixed convection flows in vertical annular enclosures in terms of the value of Froude number. Experimental investigation of laminar flow in a circular annulus with a rotating inner cylinder in the presence of laminar axial flow using sublimation technique was performed by Molki et al.[8]. They reported that the axial flow had a major influence on the location of vortices in the annulus. Stability of axial flow between concentric cylinders with the inner cylinder rotating was experimentally investigated by Lueptow et al. [9]. With an imposed axial pressure gradient, seven flow regimes of toroidal vortices including Taylor vortices, wavy vortices, random wavy vortices, modulated wavy vortices, turbulent modulated wavy vortices, turbulent wavy vortices, and turbulent vortices were identified. Lee et al.[10] conducted numerical simulations for upward air flow in a vertical pipe with rotating outer wall. Their study revealed that the rotation of the outer wall greatly influences the turbulent structures, causing suppression of turbulent motion near the rotating wall and enhancement of the turbulent intensity close to the stationary wall. Also, they have noticed that with increase in rotational speed the turbulence at the stationary wall was found to increase which in turn resulted in the hike of turbulent velocity near the stationary wall. Ouali et. al.[11] presented an experimental identification technique for the convective heat transfer coefficient inside a rotating cylinder with an axial air flow. It was reported that the heat transfer from the heated rotating cylinder is mainly influenced by the rotation and the axial flow has only marginal influence. A comprehensive review of heat transfer between concentric rotating cylinders with and without axial flow was conducted by Fenot et al.[12]. In this paper the results of the work of previous researchers for different gap thickness, axial and radial ratio, rotational velocities were compared.

In light of the above literature review it can be concluded that many of the previous investigators have analyzed the influence of the vortex flows in rotating annular space of vertical concentric cylinders on the heat transfer from the heated cylinder. However, in all these studies the inner heated cylinder is heated and rotating with/without imposed axial flow. The objective of the present work is to experimentally investigate the convection in a vertical annulus wherein the heated inner cylinder is stationary and the cold outer cylinder is rotating and the annular gap is subjected to imposed axial flow. It is worth to mention here that due to experimental limitations the results reported are mainly qualitative in nature but are not intended as an exact measure of the stability limit of the various regimes of flow.

## EXPERIMENTAL SETUP

The schematic of the experimental setup depicted in Figure 1, consists of a vertical annular space between two concentric vertical cylinders. The inner cylinder is a polished hollow copper tube of outer diameter 51mm, thickness 3mm and height 620mm. The outer cylinder is made of acrylic tube of inner diameter 83mm, thickness 5mm and height 620mm. The aforesaid dimensions of the cylinders forms an annular gap having a radius ratio of $\eta$=0.614 that comes under the limit of moderate gap annulus. The inner copper tube is heated with a cartridge heater that has been inserted from the top of the copper tube. A ceramic bush of outer diameter same as the inner diameter of the copper tube is placed at the top of the cartridge heater acts as an insulator to heat transmission from the end face of the copper tube. The cartridge heater, ceramic bush and copper tube form as an integral system which is clamped to a frame with provisions for adjusting the vertical alignment of the inner cylinder. At the bottom end of the copper tube, a teflon rod of diameter same as the outer diameter of the copper tube is inserted to prevent the heat loss from bottom end face of the copper tube. Further, the teflon rod is fixed to the frame of the assembly, thereby, enacting as a support for the bottom end of the copper tube. The inner copper tube is aligned vertically, before fixing the outer tube, and the vertical alignment inner tube is ensured using a spirit level. The cartridge heater is heated at various heat loads by applying regulated DC power (Aplab India Ltd).

The outer cylinder is mounted concentrically by means of two nylon bearings, one at the top and the other at the bottom. The bearings are supported by brackets which in turn are fixed to the frame of the assembly. The nylon bearings facilitate smooth rotation of the outer cylinder and at the same time they arrest the axial movement of the outer cylinder that may happen due to rotation. The outer cylinder carries a nylon pulley which is driven by a pulley connected to three phase AC motor (ABB India Ltd), through an open belt drive. The speed of rotation of the motor is controlled by Variable Frequency Drive, (Fujii, Electric, Japan). Care is taken to arrange the driver and driven pulleys close to each other such that the central distance between the pulleys is 150mm. The speed of rotation of the outer cylinder is measured using a stroboscope (Mextech Technologies India Pvt Ltd). A dial gauge is used to check the eccentricity of outer cylinder during rotation and the eccentricity was found to be within ± 5 microns. A closed plenum is arranged at the bottom of the concentric annulus(test section).

The experimental setup for the imposed axial flow is the same as the one discussed earlier. The only change was the addition of a centrifugal blower, which forced air into the closed plenum at the base of the test section through a circular opening having diameter 38mm. The suction of the blower was connected to the suction plenum, in which air entered through an axisymmetric nozzle with upstream and downstream diameters equal to 30and 10mm, respectively; the pressure difference across this nozzle, measured using U- tube manometer, was used for calculating the air mass flow rate(figure 2). Both plena and the blower assembly were sealed to prevent air leakage, which would affect the accuracy of flow rate measurements. the pressure difference across this nozzle measured using U-tube manometer was used for calculating the mass flow rate. The mass flow rate was varied with the help of a valve located at the downstream of the blower. Both plena and the blower assembly were sealed to prevent air leakage which could affect the accuracy of flow rate measurement. The photograph of the complete experimental setup for imposed axial flow is depicted in figures 3.

A total of sixteen T-type calibrated thermocouples (32 SWG), eight thermocouples diametrically opposite to each other, are fixed to the heated inner cylinder to measure the temperature at different locations of it. Longitudinal grooves were machined along the axial direction at different locations on the inner surface of the copper tube for embedding the thermocouple wires. The thermocouple wires were taken out from the top end of the inner cylinder. The thermocouples are located equidistantly from bottom to top of the inner cylinder. All thermocouples are fixed to the respective positions by using highly conducting cement thermobond (Fabricka India ltd). Four stainless steel sheathed thermocouples arranged on a ring at the exit section of the annulus are used to measure the bulk outlet fluid temperature. Another two T-type thermocouples are used in the plenum to measure the temperature of the fluid at the inlet of the annulus. The calibration error of the thermocouples is estimated as 0.2ºC. The thermocouples are connected to a personal computer based wireless data acquisition system (HIOKI Japan). The entire assembly is mounted on an optical bench with the help of vibration isolators (rubber boots).

**Data analysis:** The average temperature rise of inner heated cylinder is estimated by eq.1

$$\Delta T_{avg} = \frac{\sum_{j=1}^{N} T_j}{N} - T_f \qquad (1)$$

Where $N = 16$

$$T_f = \frac{T_b + T_\infty}{2} \qquad (2)$$

where $T_b$ is the bulk fluid temperature at the outlet, evaluated as per the procedure outlined in [13]

The rotational Reynolds number is calculated as:

Rotational Reynolds number, $Re_\omega = \frac{\omega r_o d}{\nu}$ (3)

Mass flow rate, $\dot{m} = A_N\sqrt{2\rho\Delta p}$

The average heat transfer coefficient and average Nusselt number are defined, respectively, as:

$$h_{avg} = \frac{Q}{A\Delta T_{avg}} \qquad (4)$$

Nusselt number, $Nu_{avg} = \frac{h_{avg} d}{k_f}$ (5)

**Uncertainty estimation:** The uncertainty in the estimated parameters is computed by the method suggested by Kline and McClintock [14, 15]. The uncertainty in the values of the estimated parameters for a given heat load Q = 80W/m$^2$, and mass flow rate =2.0g/s is shown in Table 1 reveals that maximum uncertainty in the estimated parameter is ≤2.6%. The uncertainty in the estimated parameters for other heat loads and mass flow rates were also calculated but not shown here for brevity.

$$\sigma_R = \sqrt{\pm\left\{\sum_{i=1}^{n}\left[\left(\frac{\partial R}{\partial X_i}\right)^2 \sigma_{X_i}^2\right]\right\}} \qquad (6)$$

Table 1: Uncertainty of the various measured and estimated parameters

| | Measured | | | | | | | Estimated | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | $D_o$ d | H | T | V | I | N | $\Delta h$ | A | Q | $h_{avg}$ | $Nu_{avg}$ | $Re_{\Omega}$ | Gr | $\zeta$ | $\dot{m}$ |
| | (mm) | (mm) | (°C) | (V) | (A) | (rpm) | (mm) | (%) | (%) | (%) | (%) | (%) | (%) | (%) | (%) |
| Q=80W/m$^2$ N=100rpm | 0.01 | 1.0 | 0.20 | 0.10 | 0.01 | 1.0 | 1.0 | 0.16 | 2.2 | 2.5 | 2.6 | 1.9 | 1.5 | 0.41 | 2.2 |

## RESULTS AND DISCUSSION

## ZERO IMPOSED AXIAL FLOW

Initially experiments were conducted by varying the rotational speed of outer cylinder from 0 to 450 rpm with a step size of 10 rpm, corresponding to rotation parameter in the range 0≤ ζ ≤526, heat flux of 80W/m$^2$, channel aspect ratio AR(d/H) = 38.75 and radius ratio η=0.614, without imposed axial flow. The experimental run conducted up to ζ=526 illustrated in figure 4 reveals that heat transfer from the heated cylinder is affected by the rotation of the outer cylinder. It can be inferred from figure 4 that the heat transfer rate is relatively constant up to rotational parameter ζ=6, above which it increases progressively with increasing rotation parameter. Interestingly, the slope of Nusselt number versus rotation parameter curve appears to change at rotation parameter ζ = 6 and thereafter there is a steep increase in the Nusselt number up to rotation parameter ζ = 212.5 , with further increase in rotation parameter there is rapid reduction in Nusselt number. The observation leads to the conclusion that buoyancy driven convection dominates up to ζ=6. However for rotation parameter ζ=6 to 15 the centrifugal and buoyant force complement each other thereby promoting heat transport. With further increase in rotation parameter from 15 ≤ ζ ≤ 212.5 centrifugal force dominates over buoyant force and the flow and thermal field is strongly influenced by the rotation of the outer cylinder. The rotation of the outer cylinder causes increased mass flow rate resulting the greater heat transport from the heated cylinder. As the rotation parameter is increases above 212.5 the centrifugal and gravity forces compliment each other and as a result secondary flows are produced by the body forces. These secondary vortex flows are responsible for the sharp decline in heat transfer from the heated cylinder. The numerical simulations were performed using commercial computational fluid dynamics package Ansys CFX and we could observe various flow regimes. Stream lines, velocity distribution and temperature distribution inside the annulus which we obtained from numerical simulations are not presented here, because a paper based on this work without imposed axial flow is under review in the International Journal of Thermal Science, Elsevier.

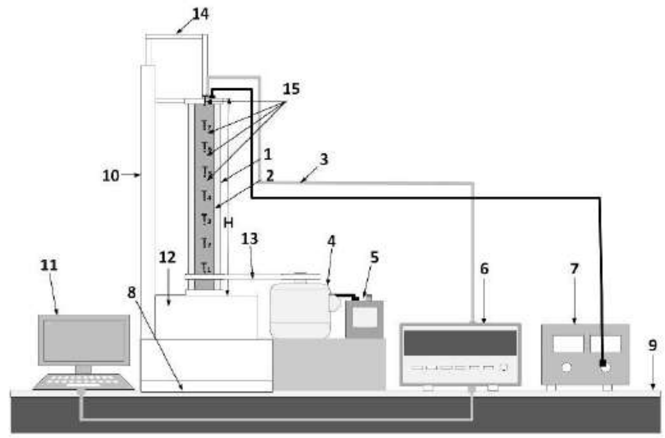
## IMPOSED AXIAL FLOW

Experiments were repeated for wide range rotational speed (0≤N≤1000rpm) corresponding to rotation parameter 0≤ ζ ≤2812. The imposed axial flow was varied by changing the mass flow rate viz: 1.108g/s, 2.0157g/s and 4.101g/s. at the inlet of the annular gap. It can be seen from figure 5, when axial flow is imposed at the inlet of the annulus the flow field inside the annular gap will be altered and the heat transfer characteristics of the heated inner cylinder is different. As seen in figure 5, inlet mass flow rate corresponding to m=1.1g/s, up to ζ = 346 the axial flow suppresses the centrifugal and buoyant forces. Also the imposed axial flow increases the fluid velocity in the annular gap which in turn enhances the heat transfer from the heated inner cylinder in comparison with when no imposed axial flow at the inlet of the annulus. Further increase in rotation parameter (ζ>346) causes the centrifugal forces due to the rotation of the outer cylinder aid the axial flow. The heat transfer from the heated inner cylinder increases with rotation parameter up to ζ≤1889. This observation leads to the conclusion that convection regime in the rotation parameter range 346≤ζ≤1889 persist in the forced convection regime. There after the flow becomes fully developed and therefore any further increase in rotational speed of the outer cylinder does not have impact on the heat transfer from the heated inner cylinder. Similar observations are obtained for other mass flow rate. But the rotational speed at which the centrifugal forces influences the axial flow is shifted to higher rotational speeds. This is due to increased mass flow rate. It is also worth to mention here as the mass flow rate increases the transition to fully developed flow takes place quickly.

## CONCLUSIONS

Experiments have been performed to study convection in vertical annular gap effected by rotating outer cylinder, with and without imposed axial flow at inlet. The salient conclusions from the study are

- In the case of flow in the annulus without imposed axial flow, the Nusselt number of the inner cylinder indicates that corresponding to rotation parameter $\zeta \leq 6$ the effect of centrifugal forces on the mean flow is very negligible and the flow is dominated by buoyant forces.
- In contrast the dependence on rotational Reynold's number is quite strong for rotation parameter in the range $6 \leq \zeta \leq 212.5$ wherein the centrifugal forces and buoyant forces compliment each other, there by increasing the heat transport.
- For the rotation parameter $212.5 \leq \zeta \leq 526$ the centrifugal and gravity forces compliment each other as a result secondary flows are produced by these body forces, resulting in deterioration in heat transfer performance of the heated inner cylinder.
- The imposed axial flow at inlet suppresses the effect of centrifugal and buoyant forces and the imposed axial flow increases the velocity of fluid in the annulus which in turn increases the heat transfer from the heated inner cylinder.
- As the rotational speed increases the centrifugal forces assist the forced flow and the heat transfer from the heated cylinder will be in the forced convection regime.
- At very high rotational speeds the flow becomes fully developed and any further increase in rotational speed does not influence the heat transfer characteristics of the inner cylinder.




1. Rotating Outer Cylinder, 2. Heated Inner Cylinder, 3. Thermocouple wires, 4. Three Phase AC motor, 5. Variable Frequency drive, 6. Data Acquisition System, 7. DC Power Supply, 8. Opening, 9. Optical Bench, 10. Frame, 11. Computer 12. Plenum, 13. Belt drive, 14. Adjustable clamp holding the heater, 15. Thermocouple positions

Figure 1: Schematic of Experimental setup without imposed axial flow


**1. Suction plenum, 2. Nozzle, 3. U – tube manometer, 4. Centrifugal blower, 5. Blower inlet, 6. Blower outlet, 7. Flow control valve**

Figure 2: Photograph of suction plenum and flow measuring Nozzle


1. Rotating Outer Cylinder, 2. Heated Inner Cylinder, 3. Thermocouple wires, 4. Three Phase AC motor, 5. Variable Frequency drive, 6. Data Acquisition System (Receiver), 7. DC Power Supply, 8. Data Acquisition System (wireless unit), 9. Optical Bench, 10. Frame, 11. U- Tube Manometer, 12. Inner Cylinder Support, 13. Belt, 14. Plenum, 15. Flow control valve

Figure 3: Photograph of the experimental setup with imposed axial flow

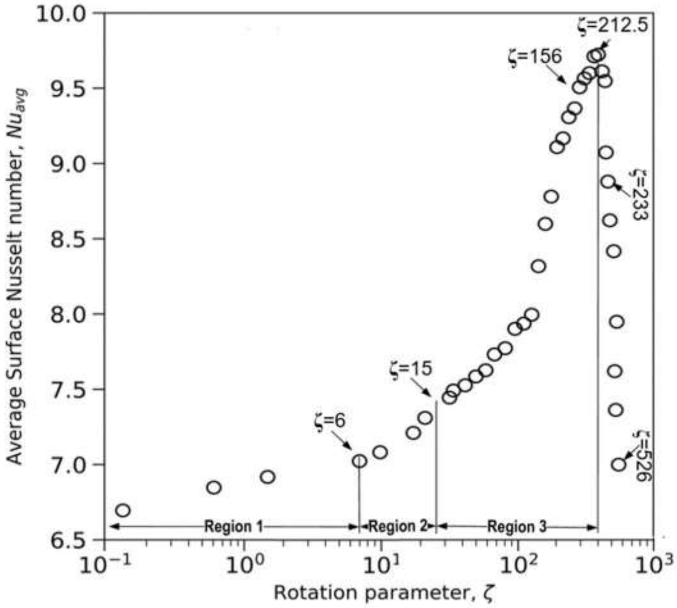



Figure 4: Variation of Nusselt number with rotation parameter, ζ (q″=80W/m2, η=0.614).

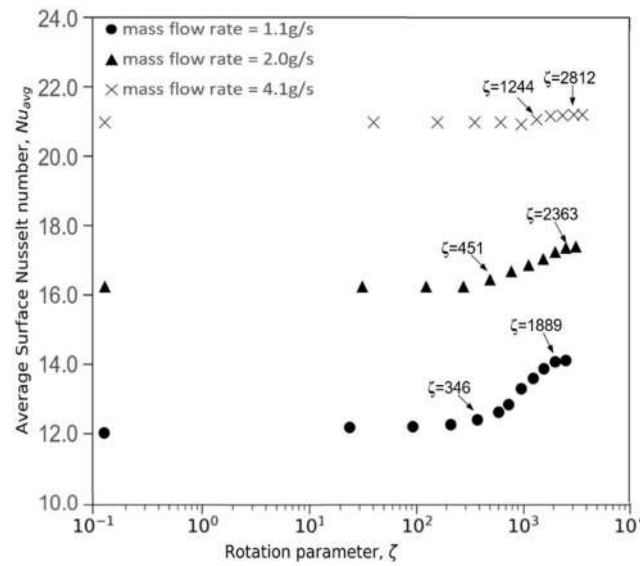



Figure 5: Variation of Nusselt number with rotation parameter, ζ, for different mass flow rates (q″=80W/m2, η=0.614)

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