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NUMERICAL INVESTIGATION OF FREE CONVECTION IN CONICAL ANNULAR SPACE WITH DISCRETE HEAT SURFACE

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ABSTRACT

Natural convection heat transfer in a vertical conical annular space with discrete heat sources is investigated numerically., The heating sources is maintained at the middle of the inner wall, while the top and bottom walls are considered adiabatic, the outer wall is kept at a lower temperature, and the non-heated parts of the horizontally wall are considered adiabatic. The steady-state continuity, Navier–Stokes and energy equations were discretized using the control volume method and solved numerically via the SIMPLER algorithm. The result shows that the discrete heat sources significantly altered the temperature distribution and the flow patterns. The results are plotted in streamlines and the isotherms as well as for different values of Rayleigh numbers and various boundary conditions in the annulus.

1. INTRODUCTION

The heat transfer analysis by natural convection in an enclosure is a large research topic owing to its wide variety of engineering applications involving energy conversion, storage and transmission systems. Case of using annulus geometry solar thermal collection, the field of heat exchangers and nuclear energy design. A comprehensive review has been documented in the literature considered to investigate the thermal natural convection in different cavity shapes. Among the very first investigations, "Chu et al. (1976)" and "Valencia and Frederick (1989)." has been analyzed numerically the heat transfer problem by natural convection flow of micropolar fluid in rectangular enclosure, to study the effect of the conductive vertical divider. The case of square and rectangular cavities was reported by "Aydin and Yang (2000)." numerically investigated the natural convection of air-filled in a vertical square cavity with the localized isothermal heating on the bottom wall. The free convection heat transfer in a vertical rectangular enclosure with three discrete flush-mounted heaters which produce time-dependent heating has been analyzed numerically by "Bae and Hyun (2004)." .The design of these electronic devices requires a deep understanding of the thermal behavior of the fluid flow in an enclosure with different arrangements of heat sources "Bazylak et al. (2006)" and "Sharif and Mohammad (2005).". Natural convection due to two and three discrete source-sink pairs on the vertical sidewalls of a square cavity has been numerically studied by

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"Deng (2008)." to understand the size and arrangement effects of the source-sink pairs on the fluid flow and heat transfer phenomena."*Muftuoglu* and Bilgen (2008)." Determined the optimum position of a discrete heater in an open square cavity by maximizing the conductance, and then studied the heat transfer and volume flow rate with the discrete heater at its optimum position. The numerical study of natural convection the phenomenon of heat transfer in vertical concentric annulus with isothermal inner and outer vertical walls. Several problems that have been widely studied in reason to its many applications of technologies have received much attention. The studies of free convection heat transfer conducted by "de Vahl Davis and Thomas (1969)" and "Keyhani et al. (1983)" and "Prasad and Kulacki (1985)" and "Kumar and Kalam (1991)."in a vertical annulus cavity. A parametric study numerically of the natural laminar convection within vertical closed annulus by the heat generation rod variation centrally vertically by "Venkata Reddy and Narasimhamn (2008).". "Khan and Kumar (1989)" and "Sankar and Do (2010)." have been extensively investigated numerically the natural convection heat transfer in a vertical annulus in the literature for uniform or discrete heating. The majority of the previous investigations in the annular cavities have treated with the uniform heating of inner wall "Sankar et al. (2006)" and "Nadeem and Akbar (2012)).". "Sankar and Do (2010)." Have investigated effect natural convection in a vertical annular bounded by to two finite size heaters in the size and location is fixed. More recently, the effects of size and the location of discrete heating on thermal convection in a filled porous vertical annulus were numerically investigated "Sankar et al. (2011).". Few research works have been reported for the case of conical, have numerically solved the coupled by natural convection heat transfer and thermal radiation problem in a conical annular cylinder porous fixed is presented by "Salman et al. (2009).". These studies have been limited to conduction heat transfer only. The present paper covers the laminar natural convection in discretely heated conical annular space. On the other hand, the direct numerical simulation necessary for well resolved the study of free natural convection flow, the computational resources required by (DNS) approach it's very well suited with of the actual capacities for the majority of real industrial problems. We will be concerned with the effect of the Rayleigh number and aspect ratio and radius ratio of the annulus as well as the size and location of heated surface has a very profound effect on the flow and heat transfer in these cavity geometry.

2. Physical and mathematical formulation

2.1. Physical Domain

In this present study, the analysis domain is delimited by two concentric conical cavity with discrete heater surface. It is placed at the middle inner wall of length h, but the upper and lower horizontal walls are kept adiabatic wall. While the outer and inner wall are maintained respectively at lower and height temperature, as shown in Fig. 1. The buoyancy induced flow is expected to be laminar, and the fluid studied is incompressible with constant fluid properties except the density variation. The Boussinesq approximation is used to compute the density variation with the temperature.

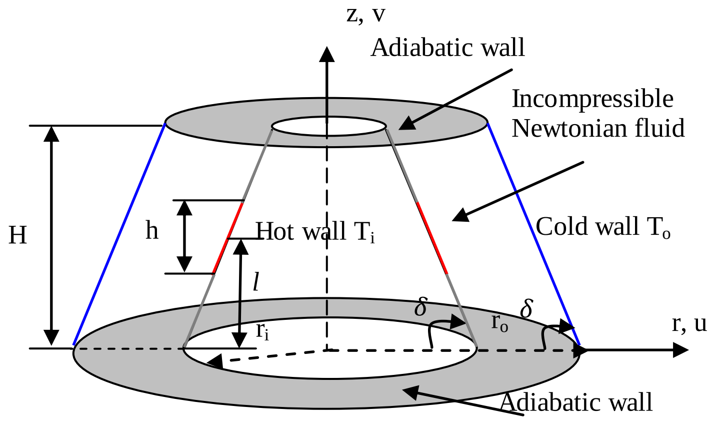


Fig.1. Physical Model

2.2. Governing Equations

The problem description and the assumptions application on the fluid properties, the governing differential equations in vector form can be written as:

Continuity,

$$\vec{\nabla}.\vec{V} = 0 \qquad (1)$$

Momentum,

$$\rho(\vec{V}.\vec{\nabla})\vec{V} = \mu\nabla^2\vec{V} - \vec{\nabla}p - \rho\vec{g} \qquad (2)$$

Energy,

$$\rho c_p(\vec{V}.\vec{\nabla})T = \lambda(\vec{\nabla}.\vec{\nabla})T \qquad (3)$$

In present study can be written as the governing dimensionless equations in cylindrical coordinates in the following forms.

Continuity,

$$\frac{1}{R}\frac{\partial}{\partial R}(RU) + \frac{\partial V}{\partial Z} = 0 \qquad (4)$$

R momentum,

$$U\frac{\partial U}{\partial R} + V\frac{\partial U}{\partial Z} = -\frac{\partial P}{\partial R} + Pr\left[\frac{\partial}{\partial R}\left(\frac{1}{R}\frac{\partial}{\partial R}(RU)\right) + \frac{\partial^2 U}{\partial Z^2}\right] \qquad (5)$$

Z momentum with the Boussinesq approximation for the buoy-ancy term,

$$U\frac{\partial V}{\partial R} + V\frac{\partial V}{\partial Z} = -\frac{\partial P}{\partial Z} + Pr\left[\frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial V}{\partial R}\right) + \frac{\partial^2 V}{\partial Z^2}\right] + PrRa(T^* - 0.5) \qquad (6)$$

Energy

$$U\frac{\partial T^*}{\partial R} + V\frac{\partial T^*}{\partial Z} = \frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial T^*}{\partial R}\right) + \frac{\partial^2 T^*}{\partial Z^2} \qquad (7)$$

Where the dimensionless variables and numbers are defined as follows:

$$U = \frac{uL}{\alpha}; V = \frac{vL}{\alpha}; R = \frac{r - r_i}{D};\ Z = \frac{z}{H}; Ar = \frac{H}{D}; K = \frac{r_o}{r_i}\ ; L = \frac{l}{H}\ ;\ D = r_o - r_i \qquad (8)$$

$$\varepsilon = \frac{h}{H}; T^* = \frac{T - T_o}{T_i - T_o};\ P = \frac{pL^2}{\rho\alpha^2};\ Pr = \frac{\nu}{\alpha};\ Ra = \frac{\beta g \Delta T L^3}{\alpha\nu} \qquad (9)$$

2.3. Boundary Conditions

The corresponding dimensionless boundary conditions for the radial and vertical velocity are equals to zero at all walls. The temperature boundary conditions are as follows.

-At $0 \leq Z \leq \frac{H}{D}$ and $R_o \leq R \leq R_o - \frac{H}{D}$ cotgδ: U= V= 0 and T^*=0 for the isothermal Cold tilted wall.

-At $L - \frac{\varepsilon}{2} \leq Z < L + \frac{\varepsilon}{2}$ and $R_i - (L - \frac{\varepsilon}{2})$cotgδ$\leq R \leq R_i - (L + \frac{\varepsilon}{2})$ cotgδ: U= V= 0 and T^*=1 for the isothermal Hot tilted wall.

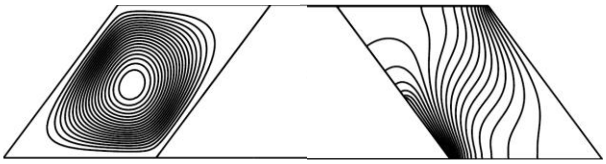
-At $L+\frac{\varepsilon}{2} \leq Z \leq \frac{H}{D}$ and $R_i-(L+\frac{\varepsilon}{2})\cot g\delta \leq R \leq R_i-\frac{H}{D}\cot g\delta$: U= V= 0 and $\frac{\partial T^*}{\partial R}=0$ for the adiabatic tilted wall.

-At $0 \leq Z < L-\frac{\varepsilon}{2}$ and $R_i \leq R \leq R_i-(L-\frac{\varepsilon}{2})\cot g\delta$: U= V= 0 and $\frac{\partial T^*}{\partial R}=0$ for the adiabatic walls wall.

-At $Z=\frac{H}{D}$ and Z =0: U= V= 0 and $\frac{\partial T^*}{\partial Z}=0$ for the adiabatic horizontal walls.

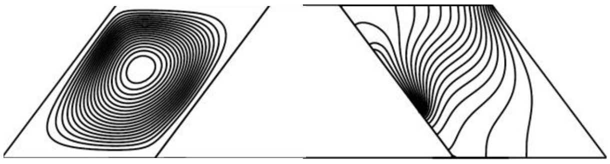
3. Results and discussion

These numerical results of the natural convection flow in vertical conical annular cavity are presented for several sizes of discrete heating surface. Three locations L = 0.25, 0.625 & 1 for various parameters of the Rayleigh number in range of ($10^4 \leq Ra \leq 10^6$).

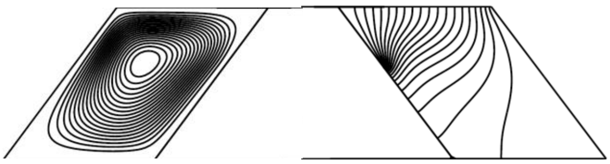


a) Ra=10^4, L=0.25

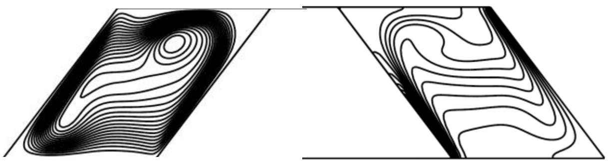
Ra=10^4



b) Ra=10^4, L=0.625

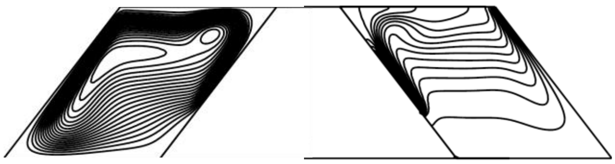


c) Ra=10^4, L=1

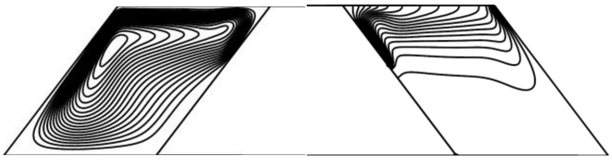


d) Ra=10^5, L=0.25

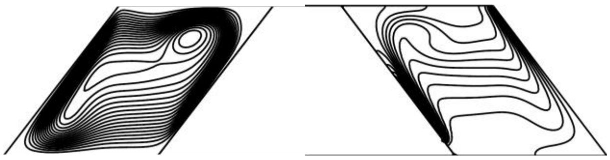
Ra=10^5



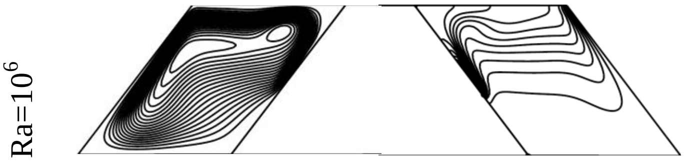
e) Ra=10^5, L=0.625



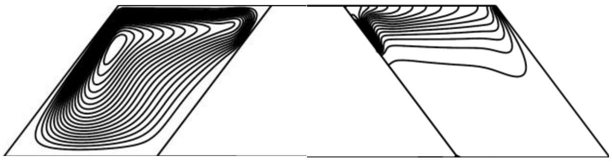
F) $Ra=10^5$, L=1



a) $Ra=10^6$, L=0.25



b) $Ra=10^6$, L=0.625



c) $Ra=10^6$, L=1

Fig. 3. Streamlines and isotherms for annulus Aspect ratio Ar=1, ε = 0.25, K=2 and δ= 45° at different locations of the heated wall.

The fluid flow and temperature distribution inside the conical annular space with discretely heated surface are presented by means of the streamlines and isotherms in Fig. 3. As Rayleigh number $Ra \leq 10^4$, the circulation intensity is weaker due to the weak buoyancy forces and the corresponding isotherms exhibit the characteristics of pure conduction. When Rayleigh number value is higher $Ra = 10^6$. the convection intensity is stronger and therefore the heat transfer flow by convection dominated the thermal field in the annulus. As the heated portion moves upwards, the fluid flow centre transformed into a single cell and becomes larger near the cold wall when the heated part is placed adjacent the horizontal wall, at lower Rayleigh number the isotherms are parallel, have deformed and move towards the cold wall.

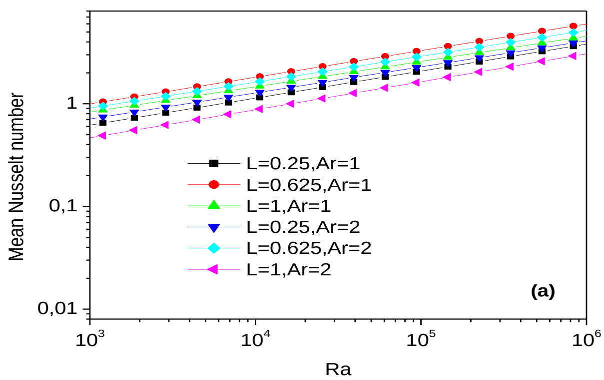


Fig. 4. Nusselt number variation with Rayleigh number for ε = 0.25 , K=2.

The Nusselt number variation with various locations L of the discrete source and aspect ratio Ar of the annulus are presented in Fig. 4. The figure clearly shows that size and location of the discreetly heated part plays an important role in maximum heat removal from the heated surface. In addition, The average heat transfer rate increases as the Rayleigh number is increased.

4. Conclusion

In this paper, the natural convection heat transfer in a Two-Dimensional vertical conical annular space for steady-state regime with discrete heat source has been investigated numerically. The effect of main parameters such as the location of heater length, Rayleigh number. From the discussion of the results, the following conclusions can be concluded:

- The heat transfer decreased when the heater length and aspect ratio increased.
- The increase of different Rayleigh numbers reinforces the natural convection flow which explains by the reducing of discrete heat source.

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