Determination of heat exchange rate by a triangular heat generating conductive body in an enclosure using CFD

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Abstract This paper performs a numerical analysis of the conjugate heat exchange inside a square enclosure full of a copper-water nanofluid. The enclosure also contains a heat-generating solid triangular block (a source of heat) at the center. While the horizontal walls of the enclosure are viewed as adiabatic, its perpendicular walls are kept up at a consistently low temperature. The second order upwind scheme is used for convective term and SIMPLE algorithm, to lead the numerical analysis and solve the discrete equations using the commercial software FLUENT15.0. The consequences of the numerical investigations are then used to clear up the effect of relevant factors, e.g., Rayleigh number, solid-volume proportion, thermal conductivity of the source of heat and transfer of heat. The result is accounted in terms of streamlines, isotherms, velocity and temperature profiles over the enclosure, and local and average Nu numbers. As per my findings, the growth of Rayleigh number and the solid volume fraction enhance the thermal performance of the enclosure. At last, the higher thermal conductivity of the source of heat is related to increases in the temperature of the nanofluid in the enclosure, the temperature of the source of heat, and the average Nu.

Keywords: solid-volume portion, conjugate heat exchange, enclosure, nanofluid, Natural convection, Nusselt Number, Rayleigh Number

1. Introduction
Heat which is transferred by natural convection is used as a part of an extensive variety of designing rehearses for electronic hardware heating, cooling and heating of structures, solidifying of solar collectors furthermore, sustenance firm. In this manner, many researchers and scientists have researched on enclosures having natural convection mostly simple ones using varying boundary conditions [1–3]. The solid material, which is present inside the enclosure like the block under conduction, inside an enclosure, alters the performance and fluid flow direction related to thermodynamics in the enclosure. Yet, on adding, it brings about an improvement of a conjugate heat conduction-convection transfer of heat. Because of sensitivity in subject, copious amount of work has been done to study the impacts of heat
generating or simple blocks on conjugate heat exchange of heat in enclosures. House et al. [4] quantified the impacts in a square-molded, heat-conductive piece put in the natural convection having a focus on the square enclosure in the system. As their discoveries indicated, changes in the body size due to Nu number (Nu) is fundamentally relied upon the proportion of the thermal conductivity of the fluid and the object. Apart from this, the presence of a comparatively little object has no huge consequences over the Nu number.

All the previously referred research worktalked about enclosures having just one non-heat producing and one heat conduction object. Be that as it may, a consistent state transfer of heat investigation was done by Oh et al. [5] in a perpendicular square enclosure with differently heated perpendicular walls and horizontal wall at adiabatic thermal boundary condition. They set a conductive and generative heat object inside the enclosure. It was observed that at higher temperature conditions, the flow produced due to the temperature variation between the perpendicular wall continued with the flow produced by the difference in temperature resulted from generation of heat. Numerical analysis on a hot bottom walls with natural convection was performed by Lee, and, Ha [6]. The perpendicular walls were insulated and the top wall was comparatively colder. A solid body with heat-generation was kept in the enclosure. As learnt from their observations, at low-temperature contrast ratio, heat produced by the heat conducting object had minimal affects.

A.Raisi [7] studied the 2-D conjugate heat heat exchange inside a square enclosure. According to his findings, including Cu nanoparticles in the base liquid (unadulterated water) expanded the resulting thermal conductivity of nanofluid. Besides, an increment of Rayleigh number supported the strength of the flow having buoyancy.

While past researchers have surveyed coupled transfer of heat in an enclosure brimming with unadulterated fluids or fluids mixed with Nano particles and having a heat conductive medium (as a source of heat), no scholar has researched conjugate transfer of heat in a square shaped enclosure filled with nanofluid and containing a heat-producing triangular conductive body (as a source of heat). Since quite a few electronic parts whose operation prompts generation of heat, can be observed along these lines. The current research intendson exploring all the impacts of various variables, e.g. Rayleigh number, the length, volumetric fraction and thermal conductivity of the source of heat, on conjugate heat exchange of heat.

### Nomenclature

| Symbol | Definition |
|--------|------------|
| $C_p$ | specific heat (J/Kg-K) |
| $g$ | gravitational acceleration (m/s²) |
| $h$ | convection transfer of heat coefficient (W/m²K) |
| $K_{th}$ | thermal conductivity (W/m-K) |
| $K_{s/Kf}$ | source of heat thermal conductivity ratio |
| $L$ | enclosure length (m) |
| $L_{source}$ | source of heat length (m) |
| $L_{length-ratio}$ | $L_{source}/L$ |
| $Nu_{left}$ | Nu on the left side of the heat source |
| $Nu_{upper bottom}$ | Nu on the upper and bottom sides of the source of heat |
| $Nu_{avg}$ | average Nu |
| $P$ | fluid pressure (Pa) |
| $P_{mod}$ | modified pressure $p + p_{g}gy$ |
| $Pr$ | Prandtl number $\nu f/K_{f}$ |
| $Ra$ | Rayleigh number $g B_{f} T L^{3}$ |
| $q'''$ | heat generation per unit volume (W/m³) |
| $T$ | temperature (K) |
| $u,v$ | velocity parts in x, y directions (m/s) |
| $U, V$ | dimensionless velocity components, $u L/\alpha_{f}, v L/\alpha_{f}$ |
| $x,y$ | Cartesian coordinates (m) |
| $X,Y$ | dimensionless coordinates $x/L, y/L$ |
Greek symbols
\( \alpha \) thermal diffusivity (m\(^2\)/s)
\( \beta \) thermal expansion coefficient (1/K)
\( \phi \) solid-volume proportion
\( \mu \) dynamic viscosity (Ns/m\(^2\))

m Kinematic viscosity (m\(^2\)/s)

h dimensionless temperature \((T - T_c) / \Delta T\)

q density (Kg/m\(^3\))

\( \Psi \) stream function

2. Problem Specification

An illustrative diagram of the square-shaped enclosure has been outlined in Figure 1. (each side length = L) The isothermal perpendicular walls are at a temperature = \( T_c \) and the horizontal adiabatic walls are filled with a Newtonian, incompressible Cu-water nanofluid. Equilibrium condition is assumed between the Cu and water nanoparticles. A source of heat of triangular-shape (each side length = l) having a thermal conductivity of \( K_s \) and heat generation rate which is volumetric of “q” is put at the enclosure centre. The flow is assumed 2-D, Steady and laminar having no impact of radiation. Table 1 tells about thermo-physical Cu properties and the base fluid (which is unadulterated water). All thermo-physical nanofluid properties (except for the density variation, which has been found using the Boussinesq approximation) are viewed as constant.

|                | \( \text{Pr} \) | \( \rho \) (Kg/m\(^3\)) | \( C_p \) (J/Kg·K) | \( K \) (w/m·K) | \( \beta \) (K\(^{-1}\)) |
|----------------|-----------------|-----------------|---------------------|-----------------|-----------------|
| Unadulterated Water | 6.2             | 997.1           | 4179                | 0.613           | 3.3881\times10\(^3\) |
| Cu              | 8933            | 385             | 401                 | 1.67\times10\(^5\) |

3. Problem Formulation

The governing equations of continuity, momentum and energy for the nanofluid and energy for the heat, source is as per the following:
\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0
\]

\[(1)\]

\[
u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = \frac{1}{\rho_{nf}} \left[ -\frac{\partial p}{\partial x} + \mu_{nf} \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) \right]
\]

\[(2)\]

\[
u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = \frac{1}{\rho_{nf}} \left[ -\frac{\partial p}{\partial y} + \mu_{nf} \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) + (\rho\beta)_{nf} g(T - T_c) \right]
\]

\[(3)\]

The given below non-dimensional parameters have been used:
\[
X = \frac{x}{L}, \quad Y = \frac{y}{L}, \quad \frac{u}{a_f} = \frac{V}{L}, \quad \frac{v}{a_f} = \frac{P}{L}, \quad \frac{R_1^2}{L^2}, \quad \frac{\Delta T}{K_s} = \frac{K_s}{K_f}, \quad \frac{Ra}{v_f a_f}, \quad \frac{Pr}{v_f a_f}
\]

\[(4)\]

The properties of the used nanofluid can also be defined based on the properties of water and copper(5),
\[
\rho_{nf} = (1 - \phi)p_f + \phi p_p
\]

\[(6)\]

\[
(p\mathcal{C}_p)_{nf} = (1 - \phi)(p\mathcal{C}_p)_f + \phi(p\mathcal{C}_p)_{p} + \phi(p\mathcal{C}_p)_{p}
\]

\[(7)\]

\[
(p\beta)_{nf} = (1 - \phi)(p\beta)_f + \phi(p\beta)_{p}
\]

\[(8)\]

\[
\alpha_{nf} = \frac{k_{nf}}{(p\mathcal{C}_p)_{nf}}
\]

\[(9)\]

The dynamic viscosity and thermal conductivity of the nanofluid can be characterized as,
\[
\mu_{nf} = \frac{\mu_f}{(1 - \phi)^{2.5}}
\]

\[(10)\]

\[
K_{nf} = K_f \left[ \frac{(K_p + 2K_f) - 2\phi(K_f - K_p)}{(K_p + 2K_f) + \phi(K_f - K_p)} \right]
\]

\[(11)\]

4. RESULTS

Enclosure filled with nanofluid under natural convection was investigated. The Nano-fluid used was a homogeneous mixture of unadulterated Cu and water nanoparticles. A triangular rigid source was likewise set at the centre of the enclosure. The analysis was performed for various Rayleigh numbers lying in the region of $10^3 \leq Ra \leq 10^7$, solid-volume proportion (0 $\leq \phi \leq 0.08$) and also the solid source having varying thermal conductivities of heat (1 $\leq k^* \leq 10$). The findings of the solution have been shown in the accompanying two segments:

4.1 Ra And Solid Volume Propotion

To assess the impacts of Ra and solid-volume proportion on the temperature fields and the stream functions, and also additionally the heat exchange performance within the enclosure, it is accepted that the length proportion (L = 0.2) and the thermal conductivity of the triangular solid source of heat (K $= 1$) has to be constant.
Figure 2 clearly showcases the isotherms and streamlines of the base fluid i.e. unadulterated water and the copper-water nanofluid ($\phi=0.04$) for different range of Rayleigh number lying between $10^3$ and $10^7$. At all Rayleigh numbers, flow cells were formed inside the enclosure for the nanofluid. While these flows were weak at low Ra, a strong buoyant flow was seen at higher Ra. Also, by the increment of force of buoyancy, the flow cells started to move towards the top portion of the enclosure. The weak flow of buoyancy and the isotherms patterns showed that a 2-D conductive heat exchange was set up around the source of heat at a low Ra. Then again, a strong buoyant flow and isotherm patterns proposed that heat exchange through convection is a dominant mechanism at high Rayleigh number.

At Ra=10^3

At Ra=10^5

At Ra=10^6
At Ra=10^7

Figure 2. Streamlines (on the L.H.S) and isotherms (R.H.S) for the Copper-water Nano-fluid filled in the enclosure, \( \phi = 0.04 \); at various Ra (L*=0.2, K*=1).

Figure 3 demonstrates the impacts of the solid-volume proportion of the y-velocity and the subsequent temperature both in dimensionless forms through the horizontal mid-section at different Ra numbers in the enclosures (Ra = 10^3, 10^5, 10^7). As observed and told so earlier also, higher Rayleigh number causes stronger flow of buoyancy and increases the maximum y-velocity. However, a high solid-volume proportion increases the viscosity of the Nano-fluid but leads to a decrease in the maximum y-Velocity. This reduction is highly significant at low Rayleigh numbers. Also, simultaneous increments in Ra and solid-volume proportion upgrade the thermal execution of the enclosure and diminishes the extreme temperatures.

![Figure 2 streamline and isotherm](image1)

![Figure 3 streamline and isotherm](image2)
Figure 3. Variation of y-velocity (LHS) and temperature (RHS) through the mid-section in the enclosure at various Ra numbers and solid-volume proportion ($L^*=0.2; K^*=1$).

Figure 4. exhibits the impacts of solid-volume proportion on local Nu number at different Rayleigh numbers ($Ra=10^3, 10^5, 10^7$). As expected, elevated local Nu is seen on all sides of the source of heat following additions in Ra and solid-volume proportion. Since conduction is exclusively responsible for heat exchange at low Rayleigh numbers, the solid-volume proportion could apply more prominent influences on local Nu number. At the same Ra number, both left and right sides of the triangular source of heat have balanced and convection is weak in the local Nu number. However, as the Rayleigh number increases, the base side’s temperature falls considerably more definitively, and along these lines, as indicated by condition (19), the local Nu number is higher on the source of heat side than on the left portion.

At $Ra=10^3$
At $Ra=10^5$

![Graph 1](image1)

At $Ra=10^7$

![Graph 2](image2)

Figure 4. Profile for local Nu number on the LHS of the source of heat and right and base sides (RHS) for different solid-volume proportion ($L^*=0.2, K^*=1$)

4.2 Source of heat and thermal conductivity

Figure 5 brings in the impacts of solid-volume proportion and the thermal conductivity of the source of heat by profiling the average Nu number. As told to us before by Equation (11), there is a visible increase in the thermal conductivity with an increment in the solid source of heat, not just the rate of heat generation (the triangular source the term in the energy equation), but also the addition of the amount of heat exchanged from the source to the nanofluid. This tends to increase the average Nu number with an increase in $K^*$. Besides, the rate is more appreciable to bring down the solid-volume proportion. Addition
of Cu-nanoparticles to the unadulterated water increases the average Nu number, however, it causes no progressions in $K^*$.

Fig. 5. Variation of the average Nu number with solid-volume proportion at different $K^* (L^* = 0.2, Ra = 10^5)$.

5. Conclusions

Counjugate heat heat exchange analysis inside a square enclosure filled with a Cu-water nanofluid isexamined numerically. A rigid triangular shaped source of heat was additionally put at the enclosure centre. As said before, the conjugate heat exchange inside an enclosure full with a nanofluid and containing a heat generating triangular conductive body, hadn’t been inspected by the past researchers. Hence, in my paper, the impacts of Rayleigh number, solid-volume proportion, the flow, temperature fields and heat source thermal conductivity on heat exchange rate and have been examined. It was observed that adding Cu nanoparticles to the fluid of the base (unadulterated water) increased the total thermal conductivity of the nanofluid. Besides, with an increase of the Rayleigh number boosts the strength of the flow of buoyancy. Subsequently, going with the significant conclusions can be taken through this examination:

- The temperatures of the nanofluid and the source of heat are contrarily related to Ra and solid-volume proportion. So, the specified temperatures reduce with increasing Ra and solid-volume proportion.
- Heat exchange will be enhanced by improving the Ra and solid volume fraction (which will improve the average Nu). Whereas, the major heat exchange in conduction part at low (Rayleigh Number $\leq 10^3$), increases in the Ra would not fundamentally convert the heat exchange rate.

At last, heat transfer rate from the source of heat to the nanofluid will be enhanced by higher thermal conductivity of source of heat, the enclosure nanofluid temperature, the source term in the equation of energy, the average Nu number and the temperature of the heat source.

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