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Effect of Length Ratio on Heat Exchange Rate by a Triangular Heat Generating Conductive Body Inside an Enclosure

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ABSTRACT:

This paper presents 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 operated at a consistently low temperature. The second order upwind scheme is used for the convective term and SIMPLE algorithm, to lead the numerical analysis and solve the discrete equations using the commercial software FLUENT 15.0. The consequences of the numerical investigations are then used to clear up the effect of length-ratio and transfer of heat. As per observations, the expansion in the length-ratio influences the rate of heat transfer.

KEYWORDS:

Conjugate heat exchanger; Enclosure; Nano-fluid; Natural convection; Nusselt Number; Rayleigh Number

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NOMENCLATURE

C_p	Specific heat (J/kg-K)
G	Gravitational acceleration (m/s^2)
H	Convection transfer of heat coefficient (W/m^2K)
K	Thermal conductivity (W/ m-K)
K	Source of heat thermal conductivity ratio Ks/Kf
L	Source of heat length (m)
Lx	Length-ratio l/L
Nuy	Local Nu on the left side of the heat source
Nux	Local Nu on the upper & bottom sides of heat source
Num	Average Nu
P_	Fluid pressure (Pa)
$\bar{p}$	Modified pressure p+p_c gy
P	Dimensionless pressure $\bar{p}L^2$/ρ_(n_f) α_f
Pr	Prandtl numberν_f⁄α_f
q'''	Heat generation per unit volume(W/m3)
Ra	Rayleigh number g_(B_f) ΔTL^3,ν_f α_f
T	Temperature (K)
u,v	Velocity parts in x, y directions (m/s)
U, V	Dimensionless velocity components, uL⁄α_f,νL⁄α_f
x,y	Cartesian coordinates (m)
X,Y	Dimensionless coordinates x/L, y/L
α	Thermal diffusivity (m^2/s)
β	Thermal expansion coefficient (1/K)
ϕ	Solid-volume proportion
u, M	Dynamic (Ns/m^2) and Kinematic viscosity (m^2/s)
H	Dimensionless temperature (T-T_c)⁄ΔT
Q	Density (kg/m^3)
Ψ	Stream function

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 and cooling of structures, solidifying of solar collectors etc. 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, 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 the subject, copious amount of work has been done to study the impacts of heat generating or simple bocks on conjugate heat exchange in enclosures. House et al [4] quantified the impacts in a square-moulded, 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 Nusselt 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 works talked about enclosures having just one non-heat producing and one heat conduction object. Be that as it may, a consistent state transfer of heat was investigated 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. At higher temperature, the flow produced due to the temperature variation between the perpendicular walls continued with the flow produced by the difference in temperature resulted from the 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, the heat produced by the heat conducting object had minimal affects. Raisi [7] studied the 2-D conjugate 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 nano-fluid. 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 nano-fluid and containing a heat-producing triangular conductive body (as a source of heat). The current research intends on exploring all the impacts of length-ratio, average Nusselt number and transfer of heat.

2. Mathematical modelling

An illustrative diagram of the square-shaped enclosure has been outlined in Fig 1. Each side length is L. The isothermal perpendicular walls are at a temperature of Tc. The horizontal adiabatic walls are filled with a Newtonian, incompressible Cu-water nano-fluid. Equilibrium condition is assumed between the Cu and water nano-particles. A source of heat of triangular-shape (each side length = l) having a thermal conductivity of Ks and volumetric heat generation rate of "q" is applied at the enclosure centre. The flow is assumed 2-D, steady and laminar having no impact of radiation. Table 1 gives the thermo-physical Cu properties and the base fluid (un-adulterated water).

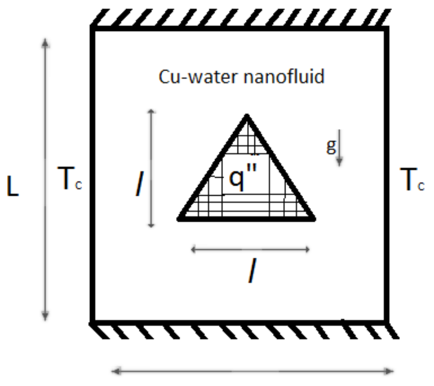

Fig. 1: An illustrative outline of the physical model

Table 1: Thermo-physical properties of water and nanoparticles

Type	Pr	P (kg/m^3)	C_P (J/kg-K)	K (W/m-K)	β (K^{-1})
Base fluid	6.2	997.1	4179	0.613	3.3881×10^{-3}
Cu	-	8933	385	401	1.67×10^{-5}

3. Mathematical modelling

All thermo-physical nano-fluid properties (except for the density variation, which has been found using the Boussinesq approximation) are viewed as constant. The fundamental equations for the heat source are as follows:

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

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

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=\frac{1}{\rho_{nf}}\left[-\frac{\partial p}{\partial y}+\mu_{nf}\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)+(\rho\beta)_{nf}g(T-T_c)\right] \quad (3)$$

The non-dimensional parameters used are as follows,

$$X=\frac{X}{L}, Y=\frac{Y}{L}, U=\frac{\mu L}{\alpha_F}, V=\frac{vL}{\alpha_F}, P=\frac{PL^2}{P_{nf}\alpha_f^2},$$

$$\Delta T=\frac{q'''L^2}{K_s}, K^*=\frac{K_s}{K_f}, Ra=\frac{g\beta_f\Delta TL^3}{v_F\alpha_f}, Pr=\frac{v_f}{\alpha_f} \quad (4)$$

The properties of the nano-fluid used can also be defined based on the properties of water and Copper,

$$\rho_{nf}=(1-\emptyset)p_f+\emptyset p_p \quad (6)$$

$$(pC_p)_{nf}=(1-\emptyset)(\rho C_p)_f+\varphi(\rho C_p)_f+\varphi(\rho C_p)_p \quad (7)$$

$$(p\beta)_{nf}=(1-\emptyset)(\rho\beta)_f+\varphi(\rho\beta)_p \quad (8)$$

$$\alpha_{nf}=\frac{k_{nf}}{(pC_p)_{nf}} \quad (9)$$

The dynamic viscosity and thermal conductivity of the nano-fluid can be characterized as,

$$\mu_{nf}=\frac{\mu_f}{(1-\vartheta)^{2.5}} \quad (10)$$

$$K_{nf}=K_f\left[\frac{(K_p+2K_f)-2\emptyset(K_f-K_p)}{(K_p+2K_f)+\emptyset(K_f-K_p)}\right] \quad (11)$$

4. Results and discussions

Enclosure filled with nano-fluid under natural convection was investigated. A triangular rigid source was likewise set at the centre of the enclosure. The analysis was performed for various length proportions (0.2 ≤Lx≤ 0.7). The solid-volume proportion (ϕ) and the thermal conductivity proportion in the solid heat generating source to the liquid (K*) are set to 1 and 0.04 respectively. The velocity vectors at three values of length ratio for Ra=10^3, 10^5 are illustrated in Figs. 2 and 3. The velocity vector is increased when we increase the Ra value. The amount of velocity vector in the region of horizontal walls is less than the perpendicular walls. Because of the high conductive heat exchange rate in the region of perpendicular walls, the extent of the perpendicular part of velocity vector is seen to be increasing as compared to that of the horizontal part.

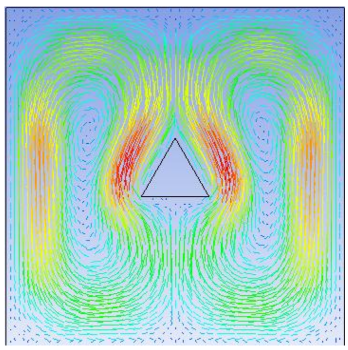

Fig. 2(a): Velocity vector (K* = 1, ϕ = 0.04) at Ra=10^3 and B=0.2

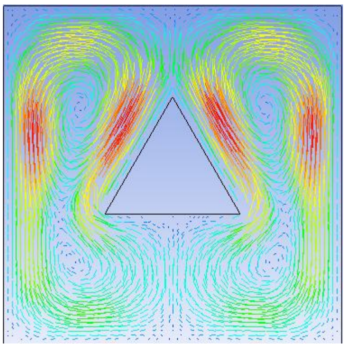

Fig. 2(b): Velocity vector ($K^* = 1$, $\phi = 0.04$) at Ra=10^3 and B=0.4

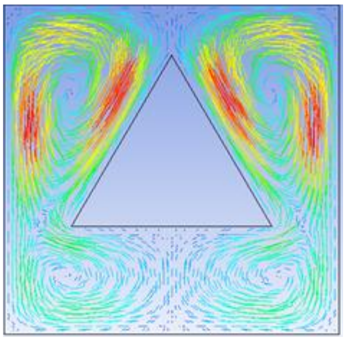

Fig. 2(c): Velocity vector ($K^* = 1$, $\phi = 0.04$) at Ra=10^3 and B=0.6

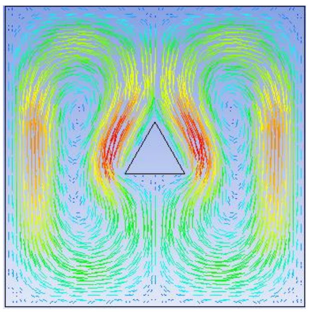

Fig. 3(a): Velocity vector ($K^* = 1$, $\phi = 0.04$) at Ra=10^5 and B=0.2

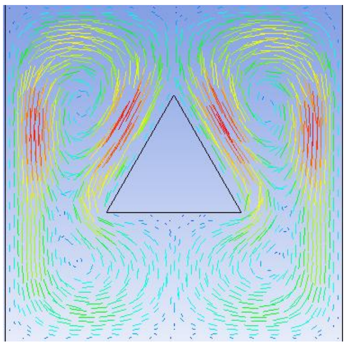

Fig. 3(b): Velocity vector ($K^* = 1$, $\phi = 0.04$) at Ra=10^5 and B=0.4

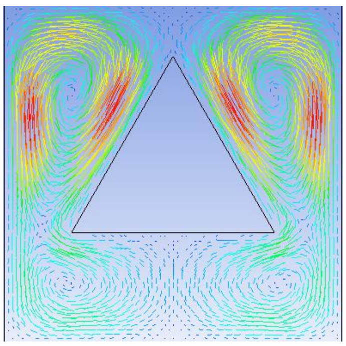

Fig. 3(c): Velocity vector ($K^* = 1$, $\phi = 0.04$) at Ra=10^5 and B=0.6

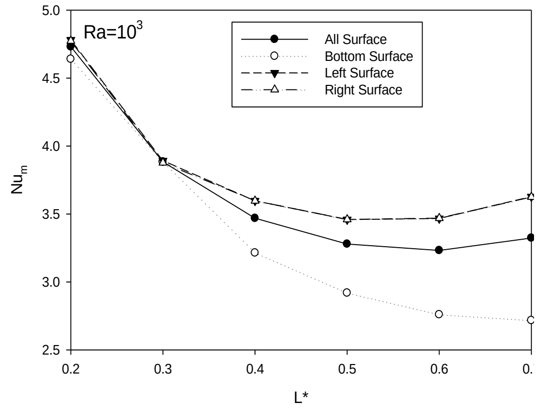

Fig. 4(a): Avg. Nu vs. Length ratio (K^*=1, $\phi = 0.04$)at Ra = 10^3

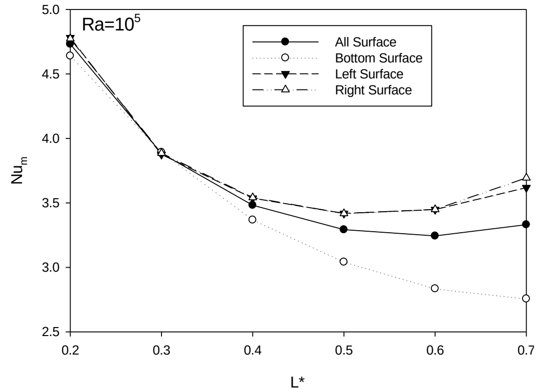

Fig. 4(b): Avg. Nu vs. Length ratio (K^*=1, $\phi = 0.04$)at Ra = 10^5

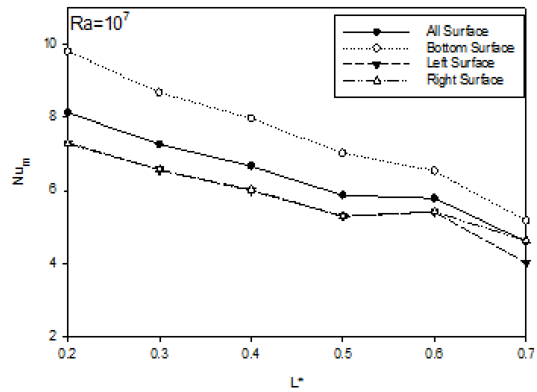

Fig. 4(c): Avg. Nu vs. Length ratio (K^*=1, $\phi = 0.04$)at Ra = 10^7

The variations in the length ratio with the average Nu number (L*) for Ra = 10^3,10^5, 10^7 is shown in Fig. 4(a) to (c) respectively. At a Rayleigh number of 10^3, conduction dominates the heat exchange (as opposed to convection). In this way, the right and the left walls of the enclosure have the same average Nu number. The range of the velocity vector in the area between the horizontal walls and heat source of the enclosure diminishes when the length ratio is increased. Along these lines, the average Nu number of the left and right walls of the heat source reduces for an increase in the length ratio. In spite of the starting reductions in the average Nu number of the heat source at left side (because of the diminished strength in the coursing cells), the reduced distance between the perpendicular cold walls and the triangular source of heat of the enclosure causes a reduction in the average Nu number at the left side. For the most part, at Ra = 10^3, the expansion in the triangular source of heat length at first diminishes, then increases the overall average Nu (the average Nu of all source of heat surfaces). On the other hand, a comparable pattern of the average Nu is seen at Ra=10^5, the average Nu in the base surface in the triangular source of heat did not considerably increase. At Ra=10^7, the heat exchange is basically an after effect of convection. In this Ra number, the natural convection stream's strength is lower at higher lengths of the heat source. Thus, average Nu number of all regions in the heat source diminishes when the length ratio increases. There is a huge gap between the average Nu number when compared to that of the base surfaces.

5. Conclusions

Conjugate heat exchange analysis inside a square enclosure filled with a Cu-water nano-fluid is examined numerically. A rigid triangular shaped heat source was additionally considered at the enclosure centre. In this paper, the impacts of length ratio have been examined. Different length ratio has various effects on the heat exchange dependent upon the Ra number. At lower Ra number, conduction is seen for the most part as the fundamental heat exchange system. Although the average Nu number initially decreases, it then ascends with additional increments in the length ratio. At higher Ra number, convection is chiefly to be blamed for the heat exchange. Thus, the average Nu number always diminishes as the length ratio is increased.

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