

# Mixed Convective Heat Transfer in Water-Based $\text{Al}_2\text{O}_3$ Nanofluid in Horizontal Rectangular Duct

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**Abstract**—In the present study, mixed convection in a horizontal rectangular duct using  $\text{Al}_2\text{O}_3$  is numerically investigated. The effects of different Rayleigh number, Reynolds number and radiation on flow and heat transfer characteristics are studied in detail. This study covers Rayleigh number in the range of  $2 \times 10^6 \leq \text{Ra} \leq 2 \times 10^7$  and Reynolds number in the range of  $100 \leq \text{Re} \leq 1100$ . Results reveal that the Nusselt number increases as Reynolds and Rayleigh numbers increase. It is also found that the dimensionless temperature distribution increases as Rayleigh number increases.

**Keywords**—Numerical simulation, Mixed convection, Horizontal rectangular duct, Nanofluids.

## I. INTRODUCTION

CONVECTIVE heat transfer and fluid flow inside horizontal rectangular ducts have received special interest and consideration by many researchers because of their important application in recent industrial demand. It has a wide variety of engineering applications such as heat exchangers technology, the design of solar collectors and modeling processes in electronic devices. Forecast of fluid flow convective heat transfer in horizontal rectangular ducts is necessary for the proper design of compact heat exchangers and other heat transfer equipment. Recently, the emerging technology using nanofluids has a tendency to increase the heat transfer compared to other base fluids. This technology lies under the concepts of using suspended nanoparticles in order to enhance the rate of heat transfer. Nanofluids are fluids that contain suspended nanoparticles such as metals and oxides. These nano-scale particles keep suspended in the base fluid and no sedimentation occurs. Thus, it does not cause increase in pressure drop in the flow field. Past studied showed that nanofluids exhibit enhanced thermal properties, such as higher thermal conductivity and heat convective transfer coefficient compared to the base fluid.

In last decade, many researchers have done numerical and experimental studies on the mixed convective heat transfer in rectangular duct. Huang and Lin [1] performed numerical simulation of transitional aiding convection flow of air through a bottom heated horizontal rectangular duct. The parameters varied were the Reynolds number, Prandtl number, aspect ratio, buoyancy-to-inertia ratio and dimensionless heating length. Results have shown that at the entry region a

significant reduction in Nusselt number. This is because the forced convection dominated in the heat transfer process. Yan [2] obtained a numerical study of mixed convection heat and mass transfer in horizontal rectangular duct. It is observed that the heat transfer performance is better for the channel with smaller aspect. This is due to the heated side-wall effect, which is more significant when the aspect ratio is small. Chiu et al. [3] studied numerically mixed convection heat transfer in horizontal rectangular ducts with radiation effects. It reveals that radiation effect tends to balance the temperature in the flow at the downstream. It is seen that the bulk temperature development was enhanced with the larger Rayleigh number and smaller conduction-to-radiation parameter.

The present study numerically investigates three-dimensional laminar mixed convective heat transfer characteristics in a horizontal rectangular duct using  $\text{Al}_2\text{O}_3$ . The results covered Rayleigh number range of  $2 \times 10^6$  to  $2 \times 10^7$  and Reynolds number range of 100 to 1100. Results of interest such as temperature distribution, Nusselt number and friction factor are reported to illustrate the effect of Rayleigh number, Reynolds number and radiation on the thermal fields of laminar mixed convection in a horizontal rectangular duct.

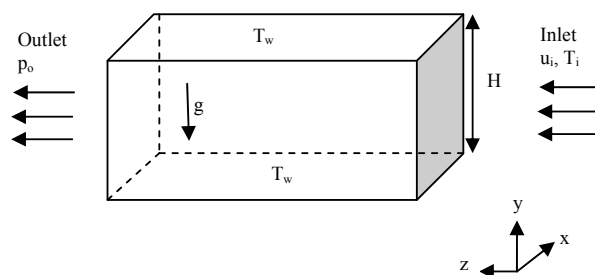


Fig. 1 The physical geometry and flow configuration of the horizontal rectangular duct

## II. PROBLEM DESCRIPTION AND GOVERNING EQUATIONS

The physical geometry of the horizontal duct and flow configuration in this study is schematically shown in Fig. 1. The height and width of the duct are  $H$  and  $W$ , respectively. All walls are maintained at a uniform and constant temperature higher than the inlet fluid temperature. The fluid flow is assumed to be laminar, steady and incompressible. The flow entering the duct is set to be at uniform velocity,  $u_i$ , and uniform temperature,  $T_i$ .

By utilizing Boussinesq approximation, the dimensionless three-dimension steady-state governing conservation

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equations for the physical problem under consideration can be written as follows [4], [5]:

Continuity equation:

$$U \frac{\partial U}{\partial X} + V \frac{\partial V}{\partial Y} + W \frac{\partial W}{\partial Z} = 0 \quad (1)$$

X-momentum equation:

$$U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} + W \frac{\partial U}{\partial Z} = -\frac{1}{\rho_{nf}} \frac{\partial P}{\partial X} + \mu_{nf} \left[ \frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} + \frac{\partial^2 U}{\partial Z^2} \right] \quad (2)$$

Y-momentum equation:

$$U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} + W \frac{\partial V}{\partial Z} = -\frac{1}{\rho_{nf}} \frac{\partial P}{\partial Y} + \mu_{nf} \left[ \frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} + \frac{\partial^2 V}{\partial Z^2} \right] + \beta g (\theta - \theta_0) \quad (3)$$

Z-momentum equation:

$$U \frac{\partial W}{\partial X} + V \frac{\partial W}{\partial Y} + W \frac{\partial W}{\partial Z} = -\frac{1}{\rho_{nf}} \frac{\partial P}{\partial Z} + \mu_{nf} \left[ \frac{\partial^2 W}{\partial X^2} + \frac{\partial^2 W}{\partial Y^2} + \frac{\partial^2 W}{\partial Z^2} \right] \quad (4)$$

Energy equation:

$$U \frac{\partial \theta}{\partial X} + V \frac{\partial \theta}{\partial Y} + W \frac{\partial \theta}{\partial Z} = \alpha_{nf} \left[ \frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} + \frac{\partial^2 \theta}{\partial Z^2} \right] \quad (5)$$

The boundary conditions are given by:

$$U = V = W = 0, \theta = 1 \text{ (at the duct walls)} \\ W = 1, U = V = \theta = 0 \text{ (at the entrance, } Z = 0)$$

The dimensionless forms are interpreted as follows:

$$X = x / D_h; \quad Y = y / D_h; \quad Z = z / D_h; \quad Z^* = Z / Pr \\ U = u / u_i; \quad V = v / v_i; \quad W = w / w_i \\ \theta = T / T_w; \quad \theta_i = T_i / T_w; \quad P = p / (\rho v^2 / D_h^2)$$

In above equations, the viscosity of the nanofluids is approximated as viscosity of a base fluid  $\mu_f$  containing dilute suspension of fine spherical particle as given by Brinkman [6]:

$$\mu_f = \frac{\mu_f}{(1-\phi)^{2.5}} \quad (6)$$

The density of the nanofluids is given as:

$$\rho_{nf} = (1-\phi)\rho_f + \phi\rho_s \quad (7)$$

The heat capacitance of the nanofluids is stated as given by Khanafer et al. [7]:

$$(\rho C_p)_{nf} = (1-\phi)(\rho C_p)_f + \phi(\rho C_p)_s \quad (8)$$

The effective thermal conductivity of the nanofluids is approximated by using Maxwell-Granetts (MG) model as shown in Khanafer et al. [7]:

$$\frac{k_{nf}}{k_f} = \frac{k_s + 2k_f - 2\phi(k_f - k_s)}{k_s + 2k_f + \phi(k_f - k_s)} \quad (9)$$

The thermal expansion coefficient of the nanofluids is given by Khanafer et al. [7]:

$$(\rho\beta)_{nf} = \phi\rho_s\beta_s + (1-\phi)\rho_f\beta_f \quad (10)$$

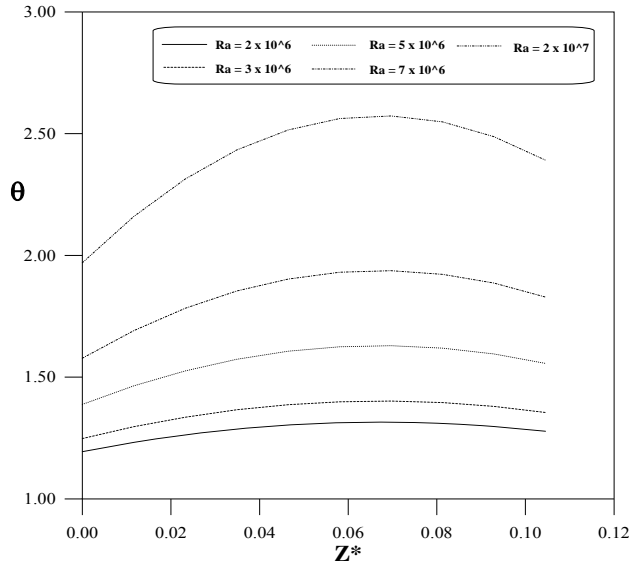


Fig. 2 Temperature distribution for various Rayleigh numbers

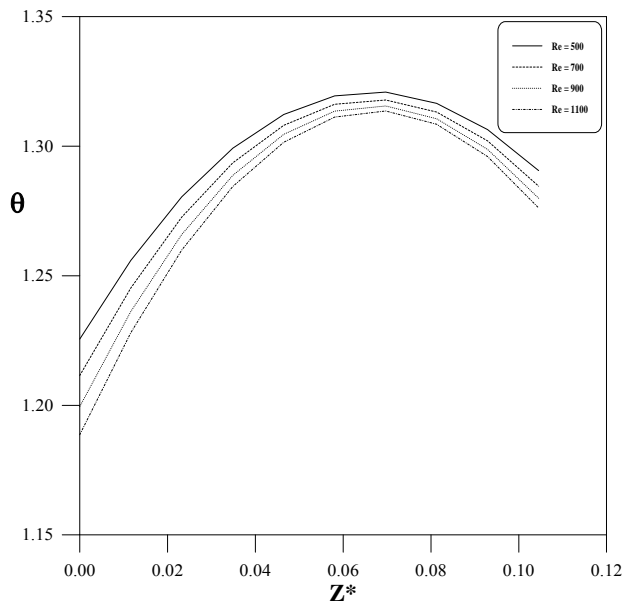


Fig. 3 Temperature distribution for various Reynolds numbers

### III. RESULTS AND DISCUSSIONS

#### A. Dimensionless Temperature Distributions

The effects of the Rayleigh number, Reynolds number and radiation on the dimensionless temperature distribution along the horizontal rectangular duct are presented, respectively, in Figs. 2-4. The dimensionless temperature increases at the entrance of the duct and then it increases to reach a maximum

value at specific position. Then, it slightly decreases at the exit of the duct due to buoyancy effect and end losses.

The effects of Rayleigh number and Reynolds number on the dimensionless temperature distribution of  $\text{Al}_2\text{O}_3$  are presented in Figs. 2 and 3, respectively. It is observed that dimensionless temperature distribution increases as Rayleigh number increases. This contributed to the developing of the thermal boundary layer faster due to buoyancy effect as Rayleigh number increases. On the other hand, the dimensionless temperature distribution decrease as Reynolds number increases. Fig. 4 presents the effect of radiation on the dimensionless temperature distribution of  $\text{Al}_2\text{O}_3$ . The figure shows that the radiation tends to increase the dimensionless temperature very slightly.

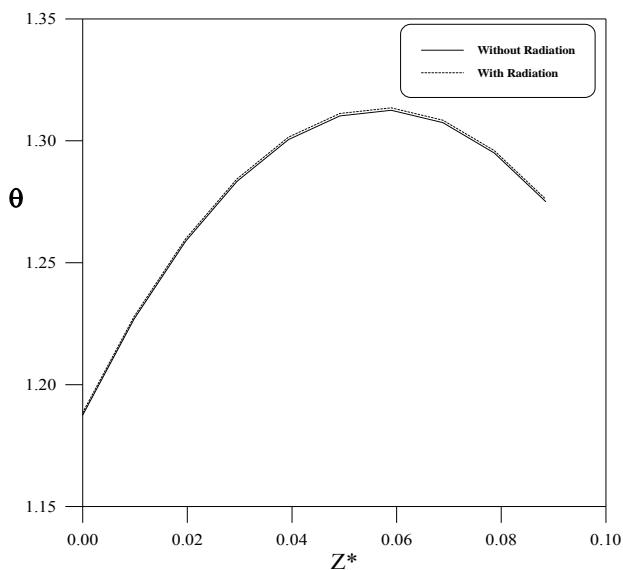


Fig. 4 Temperature distribution with and without radiation effect

#### B. Nusselt Number

The effects of the Rayleigh number, Reynolds number and radiation on the Nusselt number distribution along the horizontal rectangular duct are presented, respectively, in Figs. 5-7. As these figures show, Nusselt number decreases from its highest value at the inlet of the rectangular duct as the distance increases downstream from the inlet until it reaches its lowest value (forming a valley), due to development of thermal boundary layer, and then it starts to increase again due to the upstream axial conduction in the duct solid walls.

Figs. 5 and 6, respectively, show the effects of Rayleigh number and Reynolds number on the Nusselt number distribution of  $\text{Al}_2\text{O}_3$ . It is observed that the Nusselt number increases as Rayleigh number increases (i.e., as the buoyancy force increases). It is also observed that Nusselt number increases with increasing Reynolds number. This is because of forced convection is dominant on the heat transfer process.

Fig. 7 presents the effect of radiation on the Nusselt number distribution of  $\text{Al}_2\text{O}_3$ . The results indicate that Nusselt number

distribution with radiation is just slightly higher than Nusselt number without radiation (1% difference).

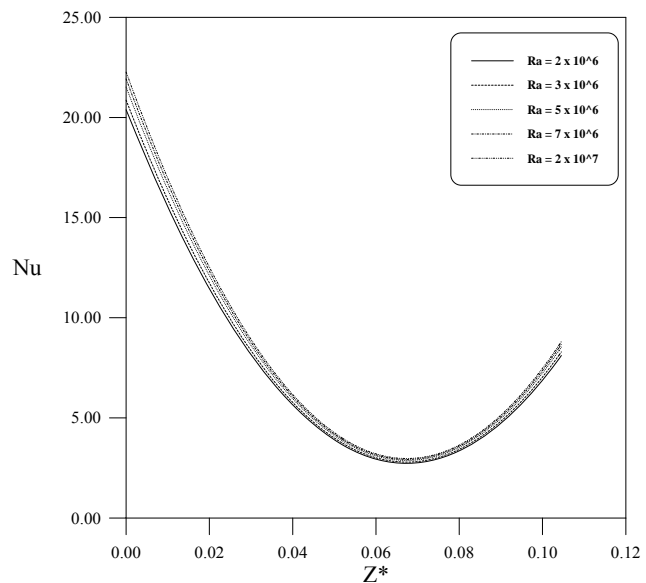


Fig. 5 Nusselt Number for various Rayleigh numbers

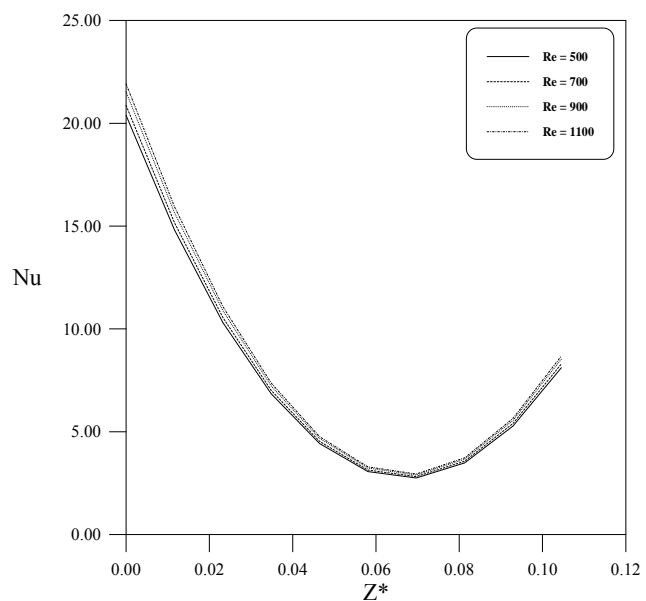


Fig. 6 Nusselt Number for various Reynolds Numbers

#### C. Friction Factor

The effects of Reynolds number on the friction factor distribution along the horizontal rectangular duct for  $\text{Al}_2\text{O}_3$  nanofluid are presented in Fig. 8. As this figure shows, the friction factor decreases from its highest value at the inlet of the rectangular duct as the distance increases downstream from the inlet until it reaches its lowest value (forming a valley), and then it starts to increase again due to the buoyancy effect. Fig. 8 shows that friction factor increases as Reynolds

number increases because of forced convection is dominant on the heat transfer process.

It should be noted that the results (not shown) reveal that the effects of Rayleigh number and radiation on friction factor are insignificant.

number. The present of radiation effect tends to increase the temperature distribution and Nusselt number. On the other hand, the friction factor increased when the Reynolds number is increased. Rayleigh number and radiation have no significant effect on friction factor.

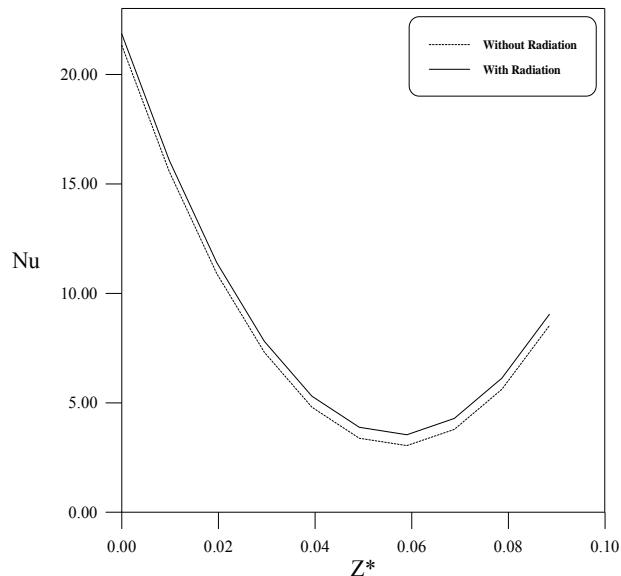


Fig. 7 Nusselt Number with and without radiation effect

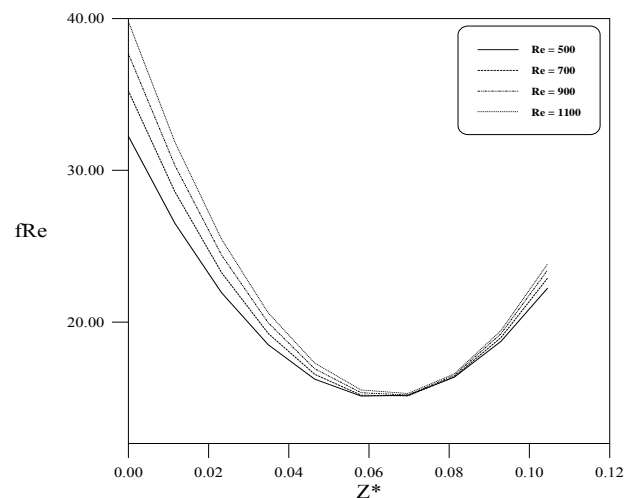


Fig. 8 Friction factor distribution for various Reynolds numbers

#### IV. CONCLUSIONS

Numerical investigations of mixed convective heat transfer in horizontal rectangular duct using  $\text{Al}_2\text{O}_3$  are presented. The effect of different Rayleigh number, Reynolds number and radiation on thermal and flow fields are intensively interpreted. It is found that the temperature distribution increases as Rayleigh number increases and Reynolds number decreases. It is also inferred the high value of Rayleigh number and Reynolds number have the highest Nusselt

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