

NUMERICAL ANALYSIS OF MIXED CONVECTION CHARACTERISTICS INSIDE A VENTILATED CAVITY INCLUDING THE EFFECTS OF NANOPARTICLE SUSPENSIONS

by

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A numerical study of mixed convection flow and heat transfer inside a square cavity with inlet and outlet ports is performed. The position of the inlet port is fixed but the location of the outlet port is varied along the four walls of the cavity to investigate the best position corresponding to maximum heat transfer rate and minimum pressure drop in the cavity. It is seen that the overall Nusselt number and pressure drop coefficient vary drastically depending on the Reynolds and Richardson numbers and the position of the outlet port. As the Richardson number increases, the overall Nusselt number generally rises for all cases investigated. It is deduced that placing the outlet port on the right side of the top wall is the best position that leads to the greatest overall Nusselt number and lower pressure drop coefficient. Finally, the effects of nanoparticles on heat transfer are investigated for the best position of the outlet port. It is found that an enhancement of heat transfer and pressure drop is seen in the presence of nanoparticles and augments with solid volume fraction of the nanofluid. It is also observed that the effects of nanoparticles on heat transfer at low Richardson numbers is more than that of high Richardson numbers.

Key words: *mixed convection, enhancement, heat transfer optimizing, nanofluid, numerical study*

Introduction

Increasing the heat transfer rate while incurring acceptable pressure drop is an important objective in many industrial applications such as electronic device cooling, microfluidic components, heat exchangers, and so on. Conventional heat transfer fluids such as water, oil, and ethylene glycol possess low thermal conductivity values thus limiting their utilization in challenging conditions. Adding some solid nanoparticles with high thermal conductivity to the fluid is one of the ways to overcome this problem. The resulting fluid is a suspension of the solid nanoparticle in the base fluid which is called a *nanofluid*. The thermal conductivities of nanofluids are believed to be greater than the base fluid due to the high thermal conductivity of the nanoparticles. In the recent years, many experiments have been carried out by researchers to identify the thermal conductivity of nanofluids [1, 2]. The results revealed that the thermal conductivity of nanofluids depends to various parameters such as interfacial layer at the particle/liquid interface [3], nanoparticle size and shape [4], nanoparticle clustering [5], Brownian motion of the nanoparticle in the base fluid [6], temperature and volume fraction concentration of the nanoparticles in the base fluid.

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In the recent years many researches have investigated numerically and experimentally the enhancement of heat transfer utilizing nanofluids [7-13]. Aminossadati [14] performed a numerical analysis of natural cooling of a right triangular heat source by a CuO-water nanofluid in a right triangular cavity that is under the influence of a horizontal magnetic field. It was found that the presence of nanoparticles in the base fluid drastically impressed the fluid flow and enhanced heat transfer rate. Khanafer *et al.* [15] carried out a numerical study of laminar natural convection in a square cavity. They have used three theoretical models for prediction of viscosity and thermal conductivity of nanofluids and deduced that the variances within different models have substantial effects on the results. Cho *et al.* [16] performed a numerical study of mixed convection heat transfer characteristics of water based nanofluids confined within a lid-driven cavity with wavy walls. They considered three different nanofluids of Cu-water, Al_2O_3 -water, and TiO_2 -water to explore the effects of utilization of nanofluids and deduced that the heat transfer is augmented in the presence of nanofluids.

This paper has focused on the characteristics of the mixed convection heat transfer in a square cavity with ventilation ports for various combinations of the Reynolds and Richardson numbers and the position of the outlet port. For a fixed inlet port, the position of the outlet port is varied along the four walls of the cavity to find out the best configuration of the system corresponding to maximum heat transfer rate and minimum pressure drop in the cavity. In doing so, the best position of the outlet port will be obtained. Eventually the utilization of Al_2O_3 -water nanofluids is performed to study its effects on heat transfer rate and pressure drop in the cavity for the best position of the outlet port.

Problem formulation

Figure 1 shows a 2-D square cavity with inlet and outlet ports. The height and width of the cavity are denoted by H and L , respectively, and are assumed to be identical ($H = L$). The depth of the enclosure perpendicular to the plane of the diagram is assumed to be long. Hence, the problem can be considered to be 2-D. The four walls of the cavity are maintained

at a constant high temperature, T_w , whereas the temperature of the fluid entering the cavity is at a constant low temperature of T_{in} . The inlet port is fixed and positioned on the top of the left wall whereas the outlet port can be present on any of the four walls. In this study the width of the inlet and outlet ports are identical ($w_i = w_o = w = 0.25 H$). To denote the position of the outlet port, a special co-ordinate system, s , along the walls is adopted with its origin at $x = 0$ and $y = H$ as identified by the dashed lines in fig. 1. The s co-ordinate of the symmetry planes of the inlet and outlet ports are $0.5w_i$ and S_o , respectively.

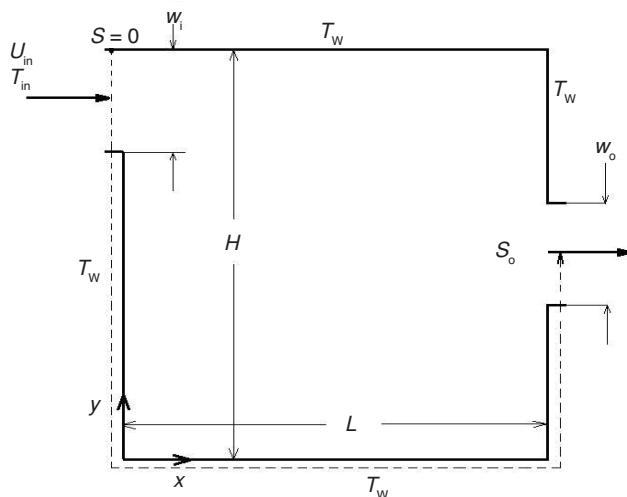


Figure 1. Schematic diagram of a cavity with inlet and outlet ports

Dimensionless form of the governing equations

The flow field is considered to be steady and in general, the fluid in the enclosure is a water based nanofluid containing Al_2O_3 nanoparticles. The thermophysical and transport properties of the fluid are assumed constant except for the density which is estimated by the Boussinesq's approximation. The thermophysical properties of the fluid phase and the nanoparticles at $T = 25^\circ\text{C}$ are presented in tab. 1.

Table 1. Thermophysical properties of the base fluid and the Al_2O_3 nanoparticles at $T = 25^\circ\text{C}$

	c_p [$\text{Jkg}^{-1}\text{K}^{-1}$]	ρ [kgm^{-3}]	k [$\text{Wm}^{-1}\text{K}^{-1}$]	$\beta \cdot 10^{-5}$ [K^{-1}]	μ [$\text{kgm}^{-1}\text{s}^{-1}$]
Water	4179	997.1	0.613	21	0.001
Al_2O_3	765	3970	40	0.85	–

With these assumptions, the dimensional transport equations are as:

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

$$\text{– momentum} \quad u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -\frac{1}{\rho_{\text{nf}}} \frac{\partial P}{\partial x} + \nu_{\text{nf}} \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) \quad (2)$$

$$u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = -\frac{1}{\rho_{\text{nf}}} \frac{\partial p}{\partial y} + \nu_{\text{nf}} \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) + \frac{1}{\rho_{\text{nf}}} g (\rho\beta)_{\text{nf}} (T - T_{\text{in}}) \quad (3)$$

$$\text{– thermal energy} \quad u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \alpha_{\text{nf}} \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \quad (4)$$

where $\alpha_{\text{nf}} = k_{\text{nf}} / (\rho c_p)_{\text{nf}}$. The density of the nanofluid is given by:

$$\rho_{\text{nf}} = (1 - \phi) \rho_f + \phi \rho_p \quad (5)$$

The heat capacitance of the nanofluid and part of the Boussinesq term are expressed:

$$(\rho c_p)_{\text{nf}} = (1 - \phi) (\rho c_p)_f + \phi (\rho c_p)_p \quad (6)$$

$$(\rho\beta)_{\text{nf}} = (1 - \phi) (\rho\beta)_f + \phi (\rho\beta)_p \quad (7)$$

with ϕ being the volume fraction of the solid particles and subscripts f, nf, and p stand for base fluid, nanofluid, and particle, respectively.

The effective viscosity of nanofluid is determined using the following relation [17]:

$$\mu_{\text{eff}} = (150\phi^2 + 2.5\phi + 1) \mu_f \quad (8)$$

This is based on experimental data in the literature for Al_2O_3 -water nanofluids.

The effective thermal conductivity of the nanofluid was given by Murshed *et al.* [18]:

$$k_{\text{nf}} = \frac{(k_p - k_{lr}) \phi k_{lr} (2\beta_1^3 - \beta^3 + 1) + (k_p + 2k_{lr}) \beta_1^3 [\phi \beta^3 (k_{lr} - k_f) + k_f]}{\beta_1^3 (k_p + 2k_{lr}) - (k_p - k_{lr}) \phi (\beta_1^3 + \beta^3 - 1)} \quad (9)$$

with $1 + h/a = \beta$, $1 + h/2a = \beta_1$, and $k_{lr} = 2k_f$. Here, h is the interfacial layer thickness, a – the particle radius, and subscripts *eff* and *lr* denote effective and interfacial layer, respectively. In this study diameter of the utilized spherical nanoparticles is 36 nm and h is about 1 nm.

Dimensionless form of the governing equations can be obtained via introducing dimensionless variables. The lengths can be scaled by the length of the cavity, H , and the velocities can be scaled by the inlet fluid velocity, u_{in} . As for the temperature, the two extreme values T_{in} and T_w are used. The dimensionless variables are then:

$$\begin{aligned} X = \frac{x}{H}, \quad Y = \frac{y}{H}, \quad U = \frac{u}{u_{in}}, \quad V = \frac{v}{u_{in}}, \quad \theta = \frac{T - T_{in}}{T_w - T_{in}} \\ P = \frac{p}{\rho_f u_{in}^2}, \quad \text{Pr} = \frac{\nu_f}{\alpha_f}, \quad \text{Re} = \frac{u_{in}(2W)}{\nu_f}, \quad \text{Gr} = \frac{g\beta_f \Delta \theta H^3}{\nu_f^2} \end{aligned} \quad (10)$$

Based on the previous dimensionless variables, the dimensionless transport equations for mass, momentum, and thermal energy are:

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \quad (11)$$

$$U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} = -\frac{\rho_f}{\rho_{nf}} \frac{\partial P}{\partial X} + 2 \frac{w}{H} \frac{1}{\text{Re}} \frac{\mu_{eff}}{\nu_f \rho_{nf}} \left(\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} \right) \quad (12)$$

$$U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} = -\frac{\rho_f}{\rho_{nf}} \frac{\partial P}{\partial Y} + 2 \frac{w}{H} \frac{1}{\text{Re}} \frac{\mu_{eff}}{\nu_f \rho_{nf}} \left(\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} \right) + 4 \frac{w^2}{H^2} \frac{\phi \rho_p \beta_p + (1 - \phi) \rho_f \beta_f}{\rho_{nf} \beta_f} \frac{\text{Gr}}{\text{Re}^2} \theta \quad (13)$$

$$U \frac{\partial \theta}{\partial X} + V \frac{\partial \theta}{\partial Y} = 2 \frac{w}{H} \frac{\alpha_{nf}}{\alpha_f} \frac{1}{\text{Pr Re}} \left(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} \right) \quad (14)$$

In the energy equation, the viscous dissipation terms are neglected and parameters α and ν are the thermal diffusivity and kinematic viscosity of the fluid, respectively.

Denoting $W = w/H$, the dimensionless form of the boundary conditions are:

- at the inlet [$X = 0$, $Y = (1 - W)$ to 1]; $U = 1$, $V = 0$, and $\theta = 0$
- on the four solid walls $U = V = 0$, and $\theta = 1$, and
- at the outlet port ($S = s/H = S_0 - 0.5 W$ to $S_0 + 0.5 W$), the outflow boundary condition was used for the velocity and temperature fields.

Based on the formulation, it is clear that the dimensionless parameters governing this problem are Re , Pr , Gr , S_0 , and W .

In this study, the Prandtl number of the fluid is fixed to 5 and the Reynolds numbers considered are 10, 40, 100, and 500, and the Richardson numbers considered are 0, 1, and 10.

In order to evaluate the heat transfer enhancement along the walls, it is necessary to observe the variations of the local Nusselt number on the walls which is defined:

$$\text{Nu} = -\frac{k_{nf}}{k_f} \frac{\partial \theta}{\partial n} \Big|_{\text{wall}} \quad (15)$$

Also the dimensionless pressure drop which is related to the difference between the average pressures of the inlet and outlet ports is defined by the following equation:

$$c_p = \frac{\bar{p}_{in} - \bar{p}_{out}}{0.5\rho u_{in}^2} \quad (17)$$

Where the average pressure is calculated by integrating the pressure over the inlet and outlet ports.

Computational details

The governing equations were solved iteratively by the finite volume method using Patankar [19] SIMPLE algorithm. A 2-D uniformly spaced staggered grid system was used. The QUICK scheme was utilized for discretizing the convective terms, whereas the central difference scheme was used for the diffusive terms. The under-relaxation factors for the velocity components, pressure correction, and thermal energy were all set to 0.2. Tolerance of the normalized residuals upon convergence was set to 10^{-5} for all cases. The validation of the present computational code has been performed in forced convection flow with $Re = 500$ and the outlet port located at the bottom of the right wall for the local Nusselt number distribution with those obtained by Saeidi and Khodadadi [20] which is shown in fig. 2. It is clear that the computed results obtained from two different codes are in excellent agreement.

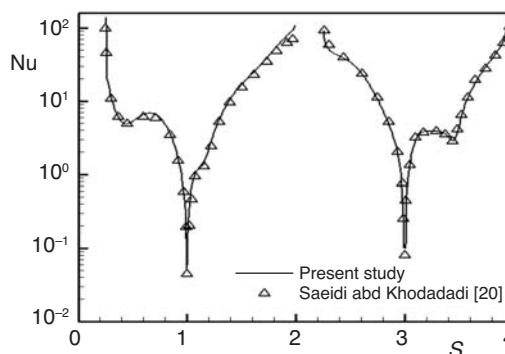


Figure 2. Comparison of the local Nusselt number predicted by the present code to those of Saeidi and Khodadadi [20]

In order to get a detailed understanding of the flow field and heat transfer characteristics of this problem, a total of 180 cases were considered. For a fluid with no particle additives, this involved studying the effect of placing an outlet port at nine different positions for a fixed position of the inlet port.

Results and discussion

Numerical solutions were obtained for $Re = 10, 40, 100$, and 500 and $Ri = 0, 1$, and 10 (108 cases). Then, the effect of the presence of the nanoparticles was investigated for the volume fraction range of 0 to 5% (remaining 72 cases). The cases with no suspended particles are presented first followed by the cases corresponding to nanofluids.

The streamlines corresponding to eight positions of the outlet port with $Re = 500$ and $Ri = 0, 1$, and 10 are shown in fig. 3. It is seen (especially for $S_o = 3.125$ and 2.875) that when the buoyancy force exists ($Ri = 1$), the small CCW rotating vortex which is located at the right bottom corner will gain in size and the other vortex at the left bottom corner will disappear. This is because when the buoyancy force exists, the presence of a hotter-than-fluid right wall promotes lifting of the colder neighboring CCW vortex fluid that was already traveling up-ward in the vertical direction. Thus, the forced and buoyancy induced convective modes combine to form a noticeable larger CCW-rotating vortex next to the right bottom corner. But the effect of the buoyancy force on the CCW vortex on the left bottom corner is opposite, thus leading to its loss of strength and eventual disappearance. However, with further increase of the Richardson number ($Ri = 10$), the buoyancy force is strong enough to push the fluid molecules near the left wall up-ward. So it is seen that a small CW vortex (not CCW) is created at the bottom left corner of the cavity. It can be seen from the other positions of the outlet port that as the Richardson number increases the

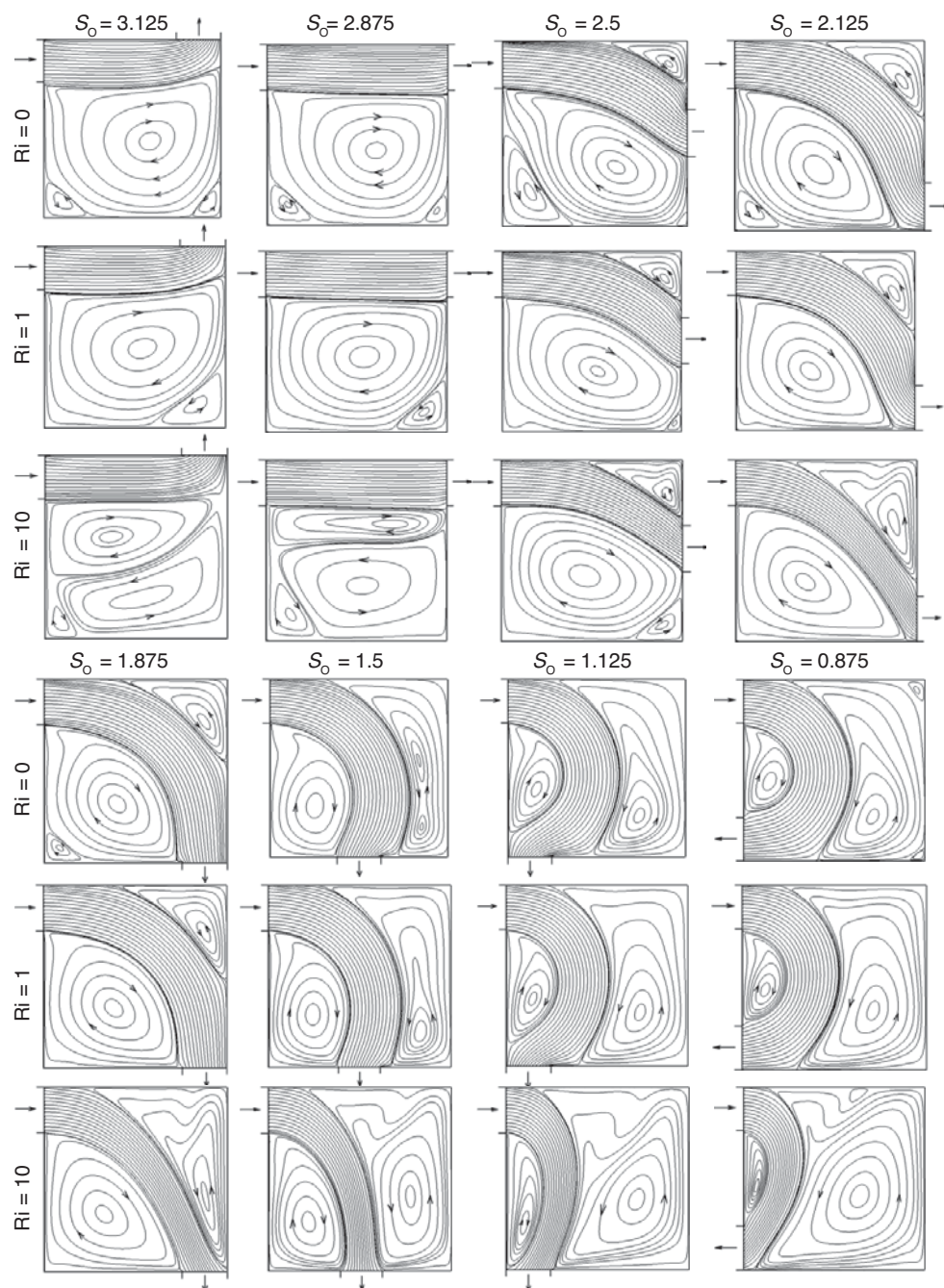


Figure 3. Streamlines for $Re = 500$ and various Richardson number and positions of the outlet port

CW vortex at the left side of the mainstream and the CCW vortex of its right side are augmented by the buoyancy force and cover a larger portion of the cavity. It is also seen that the width of the mainstream fluid becomes narrower when Richardson number increases because the two strengthened CW

and CCW rotating vortices squeeze the incoming fluid from two sides. Based on the results presented in fig. 3, an important conclusion is obtained to the effect that the buoyancy force augments the CCW rotating vortices that are on the right side of the cavity because of the extra assistance offered by the buoyancy force to the existing forced convection. On the contrary, the CCW rotating vortex on the the left side is degraded and might even disappear. The contours of the dimensionless temperature, θ , for various Reynolds and Richardson numbers are shown in fig. 4. The value of θ on the four solid walls is 1, whereas the value of θ of the fluid entering the cavity is zero and the temperature contour levels are incremented by 0.05. It is observed that the fluid temperature gradients are steep next to the walls and at the interface between the mainstream and CW and CCW rotating vortices. The cores of the vortices are generally isothermal because the rotating fluid away from the core of the vortex exchanges the major part of the thermal energy. Therefore, heat transfer at the core is dominated by conduction. It is also seen that the temperature gradient and therefore heat transfer is raised with the increasing of the Reynolds and Richardson numbers.

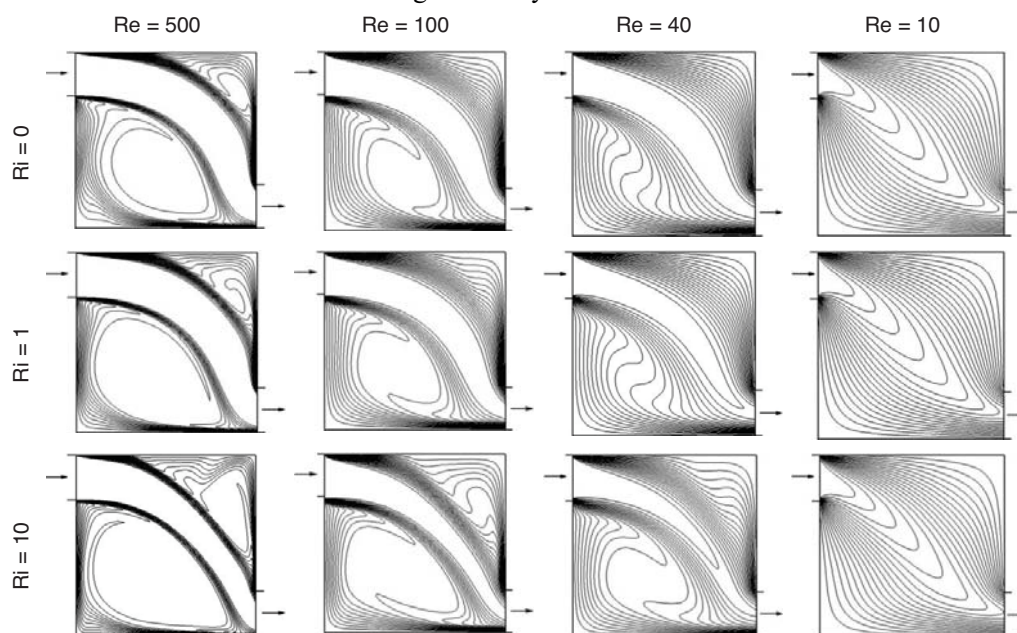


Figure 4. Temperature field in the cavity for various Reynolds and Richardson numbers

The total Nusselt number and pressure drop of the cavity as a function of the position of the outlet port for different Reynolds and Richardson numbers are presented in fig. 5. It is seen that by placing the outlet port with one end at three corners, maximum overall Nusselt number of the cavity can be achieved whereas its minimum is seen when the outlet port is located at the middle of the walls. Moreover, when the Richardson number increases, the total Nusselt number is generally enhanced. It is also observed that the pressure drop is strongly dependent on the position of the outlet port and when it is positioned at the middle of the walls, the pressure drop in the cavity attains its minimum value, whereas it gains the minimum value with the outlet port at the end of the walls. It is observed that when the Richardson number increases, the values of the dimensionless pressure drop for the cases corresponding to $S_o = 0.875$ to 2.5 increase, whereas it decreases for two cases corresponding to $S_o = 3.125$ and 3.5 (outlet on the top wall). To choose the best position of the outlet port in order to realize the highest heat exchange, one must consider

both of the average Nusselt number and system pressure drop in the cavity simultaneously. It can be concluded that by placing the outlet port on the right side of the top wall corresponding to $S_o = 3.125$, the best operating condition considering greater overall Nusselt number and lower pressure drop coefficient can be achieved.

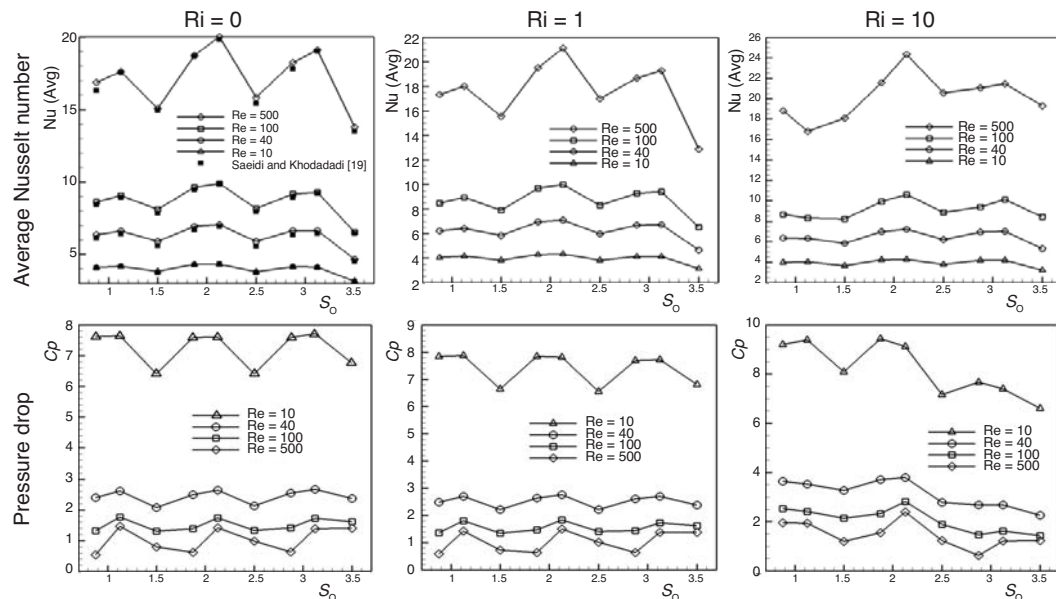


Figure 5. Variations of the total Nusselt number and dimensionless pressure drop of the cavity for different positions of the outlet port and various Reynolds and Richardson numbers

The influence of the presence of nanoparticles in the base fluid on the heat exchange rate and pressure drop in the cavity is shown in fig. 6 for the best position of the outlet port. The diagram has been plotted for various Reynolds, Richardson numbers, and volume fractions of the solid nanoparticles. In these figures, the enhancement of heat transfer in the cavity is clearly observed with the presence of the nanoparticles in the base fluid for all the Reynolds and Richardson numbers. As the volume fraction increases, the heat transfer rate is enhanced due to promotion of the thermal conductivity of the nanofluid and therefore augmentation of conduction and convection mechanism in heat transfer. For instance the presence of the nanoparticles in the base fluid enhances the average Nusselt number by about 18.45% for $Re = 10$, $Ri = 0$ and 15.48% for $Re = 40$, $Ri = 1$ for a volume fraction of 0.05. Also the increase of the Richardson number augments the heat transfer rate and hence the average Nusselt number, as discussed before. Another important matter which can be discussed is that the percentage of enhancement of heat transfer at higher Richardson number ($Ri = 10$), is less than that of lower Richardson numbers. For example, it is seen that at $Re = 500$, $Ri = 10$ the enhancement of the average Nusselt number is about 5.5% and at $Re = 500$, $Ri = 0$ is about 12.1% for a volume fraction 0.05. This is because at higher Richardson numbers, the convection mechanism is the dominant mechanism for heat transfer compared to conduction mechanism, hence with adding the nanoparticle to the base fluid, the raising of the viscosity of the resulting fluid, led to decreasing the role of the convection term in heat transfer. Therefore less noticeable enhancements are observed. As mentioned before, through adding nanoparticle to the base fluid, the viscosity of the resulting nanofluid becomes greater. Therefore, it is expected

that the pressure drop in the cavity will be raised by adding the nanoparticle to the fluid. The pertinent predictions of the pressure drop are given in fig. 6. It is seen that when the solid volume fraction increases, the pressure drop in the cavity is also raised, due to increasing of the viscosity of the nanofluid.

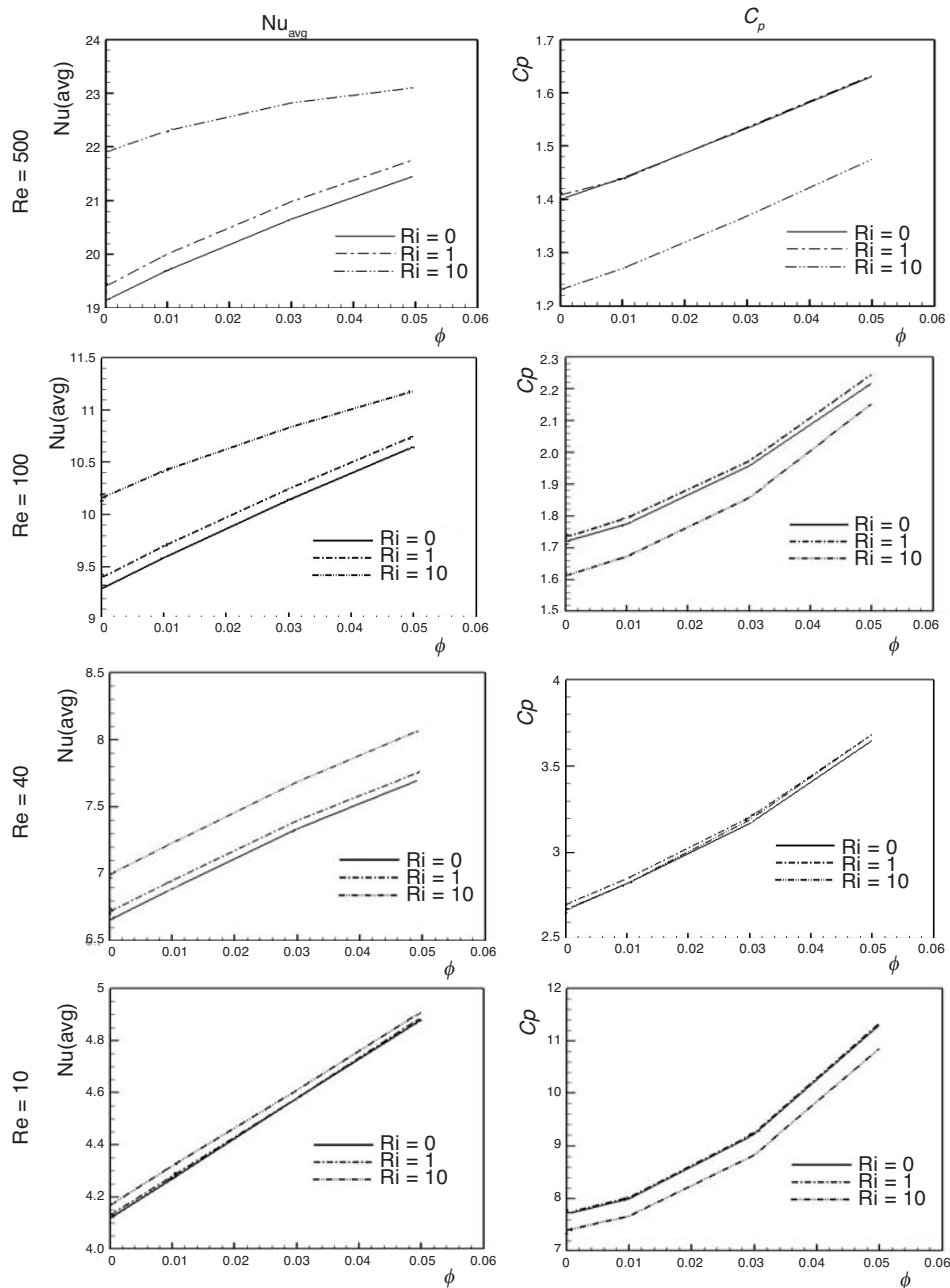


Figure 6. Effects of the nanoparticle on the average Nusselt number and pressure drop coefficient of the cavity for $Re = 500, 100, 40$, and 10 and $Ri = 0, 1$, and 10

Conclusions

- As the Richardson number increases, the CCW rotating vortices on the right side of the cavity are augmented due to favorable heating of the vortex. Similarly, the CW rotating vortices on the left side of the cavity become stronger. Also, the width of the through flow becomes narrower as Richardson number increases.
- By placing the outlet port with one end at three corners, maximum overall Nusselt number of the cavity can be achieved. Minimum overall heat transfer of the cavity is observed when the outlet port is located at the middle of the walls. Also as the Richardson number increases, the overall Nusselt number generally rises. It can be concluded that by placing the outlet port on the right side of the top wall corresponding to $S_o = 3.125$, the best operating condition considering greater overall Nusselt number and lower pressure drop coefficient can be achieved.
- The present results show that an enhancement of heat transfer can be achieved due to presence of nanoparticles. It is shown that the average Nusselt number increases by about 18.45% for $Re = 10$, $Ri = 0$ and 15.48% for $Re = 40$, $Ri = 1$ for a volume fraction of 0.05. It is also seen that the enhancement of heat transfer at low Richardson numbers is marked compared to the high Richardson numbers. Furthermore, the presence of the nanoparticles increases the pressure drop in the cavity due to the increasing the viscosity of the nanofluid.

Nomenclature

c_p	– constant pressure specific heat, [Jkg ⁻¹ K ⁻¹]
Gr	– Grashof number
g	– gravitational acceleration, [ms ⁻²]
H	– height of cavity, [m]
L	– width of cavity, [m]
k	– thermal conductivity, [Wm ⁻¹ K ⁻¹]
Nu	– Nusselt number
P	– dimensionless pressure, ($= p / \rho_{nf} u_{in}^2$)
p	– pressure, [Nm ⁻²]
Pr	– Prandtl number, ($= \nu_f / \alpha_f$)
Re	– Reynolds number, ($= u_{in} 2w / \nu_f$)
Ri	– Richardson number, ($= Gr Re^{-2}$)
S	– dimensionless co-ordinate, ($= s / H$)
s	– co-ordinate adopted for distance along the walls, [m]
T	– temperature, [K]
U, V	– dimensionless velocity components,
u, v	– components of velocity, [ms ⁻¹]
w	– width of the inlet and outlet ports
X, Y	– dimensionless Cartesian co-ordinates
x, y	– Cartesian co-ordinates, [m]

Greek symbols

α	– thermal diffusivity, [m ² s ⁻¹]
β	– thermal expansion coefficient, [K ⁻¹]
θ	– dimensionless temperature
μ	– dynamic viscosity, [kgm ⁻¹ s ⁻¹]
ν	– kinematic viscosity, [m ² s ⁻¹]
ρ	– density, [kgm ⁻³]
θ	– particle volume fraction

Subscripts

avg	– average
eff	– effective
f	– fluid
in	– inlet
le	– interfacial layer
nf	– nanofluid
o	– outlet
p	– particle
w	– wall

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