Free convection cooling in modified L-shape enclosures using copper–water nanofluid

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Abstract

The aim of the present study is investigation of free convection cooling in an L-shape enclosure filled with copper–water nanofluid. The governing equations are solved numerically using finite volume approach. The effects of the volume fraction of the Cu nanoparticles, Rayleigh number and the aspect ratio of the L-shaped enclosure on the heat transfer coefficient, temperature and velocity profiles are studied. The results show that at high Rayleigh numbers, the dominant heat transfer mechanism shifts from conduction to free convection. By increasing Rayleigh number, the numbers of eddies will increase, but the maximum heat transfer coefficient will decrease. For all ranges of Rayleigh number, increasing the volume fraction of the Cu nanoparticles enhances heat transfer coefficient. The simulation results show that inclusion of a number of pins inside the enclosure has a significant effect on increasing the heat transfer coefficient. Also, inclination angle is another important improvement factor for increasing free convection in the enclosures. The numerical results demonstrate that the enclosure with an inclination angle of $\omega = 225^\circ$ has the maximum heat transfer coefficient.

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1. Introduction

Sensitive electronic devices such as central processing units (CPUs) are being used extensively in applications where the heat transfer rates are small. Facilitating effective heat transfer in order to maintain a specified temperature on the device and more efficient heat removal is an essential matter in cooling system development, since improper cooling can adversely affect device performance. Different developments proposed for improving the cooling of chips such as micro-channel, jet impingement, porous media flow, and etc. Though application of micro-channel water cooling is a well-known process for removal of high heat fluxes in electrical cooling systems, there are many limitations associated with this method such as high pumping power to keep the temperature gradient in the fluid within acceptable ranges, high freezing point of water and also low reliability concerns about application of water in electrical instruments [1,2]. Another process is liquid jet impingement cooling which presents some advantages such as low thermal resistances, no thermal interface and relatively uniform surface temperatures however the disadvantages associated with this method are high pumping power and erosion of the surfaces due to the high-velocity [3–5]. Application of porous media with single- or two-phase flow is favorable due to its large surface area however, the main issue with this process is high pumping power demand [6,7]. To overcome the issues related to the pumping power and coolant low thermal conductivity, natural convection in enclosures filled with nanofluids has been widely studied as possible technology and suitable alternative for high heat flux cooling (local heat fluxes on CPUs up to 400 W/cm² or more [8]) owing to increasing heat transfer coefficient over hot-spot locations. Free convection phenomenon in enclosures is promising for nanotechnology-based cooling applications such as Micro-Electro-Mechanical Systems (MEMS), electronic packaging, solar collectors, conductivity coolants, lubricants, concentric cryogenic tubes, hydraulic fluids and metal cutting fluids, building walls and double-glazed windows. In this study, thermal performance of an L-shape enclosure filled with a nanofluid is examined in detail. If the walls of the enclosure are not at a same temperature, the fluid in the enclosed space is affected by buoyancy forces which cause the fluid to circulate in the enclosure transferring heat from the hot side to the cold side by convection mechanism. On the other hand, if buoyancy forces are not large enough to overcome viscous forces, the dominant mechanism of heat transfer will essentially be conduction [9].

Heat transfer enhancement using nanofluids depends on the physical characteristics of the nanoparticle such as shape, size,
concentration and also their thermal properties. Several investigators reported that up to 20% enhancement in thermal conductivity can be achieved using low concentrations of nanoparticles (e.g., 1–5% by volume) [10–12]. Nanofluids behave like a fluid rather a mixture hence; several studies have been conducted to investigate the mechanisms involved in the enhancement of the thermal conductivity of nanofluids. Xuan and Roetzel have considered the mechanism of heat transfer enhancement in nanofluids [13]. They also studied the thermal dispersion effects and the transport properties of nanofluids using conventional and modified approaches [13]. Wen and Ding examined convective heat transfer of nanofluids made of water and $\gamma$-Al$_2$O$_3$ nano-particles under laminar flow conditions [14]. They observed significant convective heat transfer enhancement in the entrance region attributed to the enhancement of the effective thermal conductivity. Also, they concluded that the nanofluids have greater thermal developing length with respect to the pure base liquid, and it is enhancing with an increase in particle concentration [14]. Santra et al. [15] studied the heat transfer due to laminar flow of copper–water nanofluid through two isothermally heated parallel plates. Their simulation results represented that the rate of heat transfer enhances with the increase in flow as well as increase in solid volume fraction of the nanofluid.

In the conventional approach, the thermal properties of nanofluid are substituted in the existing heat transfer coefficient correlations for pure fluid. But in the modified approach, due to the random movement of particles, the thermal dispersion has been considered.

The mechanism and application of free convection in partially heated enclosures for different conditions have been studied extensively in the last decades [17–33]. Chu et al. analyzed the effect of location and size of heater, aspect ratio and boundary conditions on the free convection in a rectangular enclosure filled with air [16]. They concluded that heater size and location are essential parameters on temperature and velocity distributions. In another related work, Aminossadadi and Ghasemi considered the effects of size and location of heater, volume fraction of the nanofluid particles and Rayleigh number ($R$) on free convection in a square cavity [17,18]. Their numerical investigations indicated that type of nanoparticles, the length and location of the heat source have impressive effect on the maximum temperature of heat source and also nanoparticles increase the rate of heat transfer. Khanaf et al. reported that by increasing the solid volume fraction, the rate of heat transfer inside rectangular enclosures filled with nanofluids will be increased for the entire range of Grashof number [29]. Oztop and Abu-Nada considered free convection of nanofluids in partially heated rectangular cavities [30]. Their results showed that by increasing the solid volume fraction of the nanofluids and the size of heater, the average Nusselt number will increase over a wide range of Rayleigh numbers. The study of thermophysical properties of nanofluids such as effective dynamic viscosity, thermal conductivity, thermal expansion coefficient is another related field which considered by many researchers [34–39].

Kaluri and Basak analyzed the effect of differential and distributed heating methodologies on thermal mixing and temperature uniformity in the natural convection in square cavities [40]. They observed a remarkable uniformity in temperature across the cavity by moderate thermal mixing [40]. Heatline based thermal management for free convection within right-angled porous triangular enclosures is considered by Anandalakshmi et al. [41]. They reported that isothermal heating of walls enhances the heat distribution and thermal mixing.

Despite a large number of studies on free convection in the nanofluid filled enclosures, there is a fundamental lack of information on the optimum conditions and the extent of heat transfer rate one can achieve. This study examines the effect of various operating and design parameters such as Rayleigh number, geometry and the volume fraction of copper nanoparticles on the free

![Fig. 1. Schematic diagram of the L-shape enclosure under consideration (a) conventional and (b) modified structure.](image)
convection characteristics within an L-shape enclosure filled with nanofluids. The enclosure geometry was modified by insertion of a few round pins and by inclination to study their effects on the rate of heat transfer. The results of the present study can be used as a guideline for the design and operation of electronic cooling systems.

2. Modeling

2.1. Problem definition

The schematic of the L-shape enclosure under study is shown in Fig. 1. As shown, the enclosure height is \( H \) and width is \( W \). The dimension of the cavity perpendicular to its plane is assumed to be long enough, such that the problem can be considered two-dimensional. For the present case, the height and width are considered to be equal, \( H = W \). One side of the enclosure is in contact with a constant temperature heat source at \( T_h = 120^\circ \text{C} \) and the other side is exposed to the ambient temperature at \( T_c = 20^\circ \text{C} \). The other surfaces are considered to be insulated and impermeable. The hot, cold and adiabatic walls are specified in the figure. The aspect ratio of the cavity is defined in Eq. (1):

\[
AR = \frac{L}{H}
\]

(1)

The enclosure is filled with a nanofluid of water and solid spherical copper nanoparticles. The nanoparticles are considered to be uniform in shape and size. The thermo-physical properties of water (base fluid) and Cu nanoparticles are listed in Table 1. The nanofluid is considered as incompressible and Newtonian and the flow is assumed to be laminar. It is also assumed that there is no slip between the nanoparticles and the base fluid and they are in thermal equilibrium with each other. The Boussinesq approximation is also applied. The free convection was studied for cavities with various aspect ratios, nanoparticles volume fractions and Rayleigh numbers.

2.2. Mathematical model

To simplify mathematical calculations, the following conditions are considered; steady state, continuum, Boussinesq approximations, two-dimensional laminar flow (\( Ra < 10^9 \)), no viscous dissipation and no radiation. The governing equations for continuity, momentum and energy are as follows:

\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0
\]

(2)

<table>
<thead>
<tr>
<th>Table 1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermophysical properties of water and copper at 20 °C [48].</td>
</tr>
<tr>
<td>Physical properties</td>
</tr>
<tr>
<td>( C_p ) (J kg(^{-1}) K(^{-1}))</td>
</tr>
<tr>
<td>( \rho ) (kg m(^{-3}))</td>
</tr>
<tr>
<td>( k ) (W m(^{-1}) K(^{-1}))</td>
</tr>
<tr>
<td>( \beta ) (K(^{-1}))</td>
</tr>
<tr>
<td>( \alpha ) (m(^2) s(^{-1}))</td>
</tr>
</tbody>
</table>

Table 2

Effect of the grid density on the average Nusselt number of the hot walls of the cavity with \( AR = 0.4 \), \( Ra = 10^3 \), zero inclination angle and filled with the nanofluid with \( \phi = 0.06 \).

<table>
<thead>
<tr>
<th>Grid size</th>
<th>20 x 20</th>
<th>40 x 40</th>
<th>60 x 60</th>
<th>80 x 80</th>
<th>100 x 100</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nu</td>
<td>4.6598</td>
<td>4.6621</td>
<td>4.6734</td>
<td>4.6857</td>
<td>4.6861</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Comparison of the average Nusselt number in an air-filled L-shape enclosure at ( AR = 0.25 ) with the reported data [46,47].</td>
</tr>
<tr>
<td>(Tamin and Mahmoodi simulation)</td>
</tr>
<tr>
<td>( Ra = 10^3 )</td>
</tr>
<tr>
<td>( Ra = 10^4 )</td>
</tr>
<tr>
<td>( Ra = 10^5 )</td>
</tr>
</tbody>
</table>

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

(3)

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

(4)

\[
\alpha_{nf} = \frac{k_{nf}}{(\rho C_p)_{nf}}
\]

(7)

and \( \phi \) is the volume fraction of nanoparticles. The nanofluid thermal conductivity, \( k_{nf} \), can be calculated using Maxwell equation [42].

\[
k_{nf} = k_1 \left[ \left( k_s + 2k_T \right) - 2\phi \left( k_1 - k_s \right) \right] \]

(8)

where \( k_s \) and \( k_T \) are the thermal conductivity of dispersed nanoparticles and pure fluid, respectively. The heat capacitance and thermal expansion coefficient of the nanofluid are given by Ref. [22]:

\[
(\rho C_p)_{nf} = (1 - \phi)(\rho C_p)_f + \phi(\rho C_p)_s
\]

(9)

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

(10)

and the Brinkman correlation [43] can be used to calculate the effective dynamic viscosity of the nanofluid.

<table>
<thead>
<tr>
<th>Table 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Comparison of the average Nusselt number in a copper–water nanofluid filled L-shape enclosure at ( AR = 0.4 ) and ( \phi = 0.06 ) with the reported data [47].</td>
</tr>
<tr>
<td>(Mahmoodi simulation)</td>
</tr>
<tr>
<td>( Ra = 10^3 )</td>
</tr>
<tr>
<td>( Ra = 10^4 )</td>
</tr>
<tr>
<td>( Ra = 10^5 )</td>
</tr>
<tr>
<td>( Ra = 10^6 )</td>
</tr>
</tbody>
</table>
\[ \mu_{nf} = \frac{\mu_f}{(1 - \phi)^2 S} \]  

The temperature dependency of nanofluid density is obtained by expanding density in a Taylor series about temperature as follows:

\[ \rho = \rho_0 (1 - \beta(T - T_0)) \]  

Equations (2)–(5) can be converted to non-dimensional forms, using the following non-dimensional parameters:

\[ X = \frac{x}{L} \]  
\[ Y = \frac{y}{L} \]  
\[ U = \frac{uL}{\alpha_f} \]  
\[ V = \frac{vL}{\alpha_f} \]  
\[ P = \frac{PL^2}{\rho_f \alpha_f^2} \]  

\[ \theta = \frac{T - T_f}{\Delta T} \]  
\[ \Delta T = \frac{q''L}{k_f} \]  

The non-dimensional continuity, momentum and energy equations are obtained as follows:

\[ \frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \]  
\[ U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \frac{\mu_{nf}}{\rho_{nf} \alpha_f} \left( \frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} + \frac{\rho \beta_{nf}}{\rho_{nf} \alpha_f} \right) \]  
\[ U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \frac{\mu_{nf}}{\rho_{nf} \alpha_f} \left( \frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} + \frac{\rho \beta_{nf}}{\rho_{nf} \alpha_f} \right) \]  
\[ U \frac{\partial \theta}{\partial X} + V \frac{\partial \theta}{\partial Y} = \frac{\alpha_{nf}}{\alpha_f} \left( \frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} \right) \]

where Rayleigh and Prandtl numbers are defined as

\[ Ra = 10^3 \]  
\[ Ra = 10^4 \]  
\[ Ra = 10^5 \]  
\[ Ra = 10^6 \]

**Fig. 2.** Isotherms profile in the L-shape enclosure with $AR = 0.2$ and $\phi = 0$. 
The following equations are used:

\[ Ra = \frac{g \beta L^3 \Delta T}{\nu \alpha_f} \]  \hspace{1cm} \text{(24)}

\[ Pr = \frac{\nu}{\alpha_f} \]  \hspace{1cm} \text{(25)}

No-slip condition is imposed for all velocities on the walls, thermal boundary conditions are \( T = T_h \) for the hot walls, \( T = T_c \) for the cold walls and \( \partial T/\partial n = 0 \) for the adiabatic walls. By applying the dimensionless parameters, the following boundary conditions are obtained.

\[ U = V = 0, \quad \theta = 1, \quad \text{On the hot walls} \]  \hspace{1cm} \text{(26)}

\[ U = V = 0, \quad \theta = 0, \quad \text{On the cold walls} \]  \hspace{1cm} \text{(27)}

\[ U = V = 0, \quad \frac{\partial \theta}{\partial n} = 0, \quad \text{On the adiabatic walls} \]  \hspace{1cm} \text{(28)}

where \( n \) is normal direction to the walls. The local Nusselt number can be expressed as

\[ Nu_l = \frac{hH}{k_f} \]  \hspace{1cm} \text{(29)}

where the heat transfer coefficient and thermal conductivity are calculated from the following equations:

\[ h = \frac{q_w}{T_h - T_c} \]  \hspace{1cm} \text{(30)}

\[ k_{nf} = \frac{q_w}{\partial T/\partial x} \quad \text{on the vertical walls} \]  \hspace{1cm} \text{(31)}

\[ k_{nf} = \frac{q_w}{\partial T/\partial y} \quad \text{on the horizontal walls} \]  \hspace{1cm} \text{(32)}

By substituting Eqs. (30)–(32) in Eq. (29), the Nusselt number can be written as

\[ Nu_l = \left( \frac{k_{nf}}{k_f} \right) \frac{\partial \theta}{\partial x} \quad \text{On the vertical walls} \]  \hspace{1cm} \text{(33)}

\[ Nu_l = \left( \frac{k_{nf}}{k_f} \right) \frac{\partial \theta}{\partial y} \quad \text{On the horizontal walls} \]  \hspace{1cm} \text{(34)}

The average Nusselt number along the hot wall is calculated by integrating the local Nusselt number along the hot walls as follows:

![Fig. 3. Isotherms profile in the L-shape enclosure with AR = 0.2 and \( \phi = 0.03 \).](image-url)
\[ Nu = \frac{1}{2} \left( \int_0^1 N_t dX + \int_0^1 N_t dY \right) \]  \hspace{1cm} (35)

2.3. Modified L-shape enclosures

Structural modifications in L-shape enclosures could be useful for enhancing the heat transfer rate. In this study, the conventional L-shape enclosure has been modified by insertion of a few cylindrical pins and also through an inclination angle as illustrated in Fig. 1b. The pins are in fact inside the enclosure and in direct contact with the nanofluid. They are perpendicular to the side walls. The objective of using such pins is to modify flow regime inside the nanofluid and enhance heat transfer rate. In the present study, inclination angles, \( \omega = 0^\circ, 45^\circ, 135^\circ, 225^\circ \) and \( 315^\circ \) are considered.

3. Numerical approach and validation

Velocity and temperature distributions are coupled in free convection and the solution to such problems requires simultaneous integration of the continuity, momentum, and energy equations. In this work, numerical techniques based on finite-volume method are used to discretize the governing mass, momentum and energy equations [44]. SIMPLER algorithm is used to couple the pressure and velocity fields in the momentum equation and since the governing equations are nonlinear, successive over-under relaxation method is used to solving the equations [45]. All calculations are executed on a Core Duo CPU 2.33 GHz system and the CPU time for each simulation was about 20 s.

The convective and diffusive terms in the governing equations are approximated by second order upwind and central differential schemes, respectively. The convergence criterion in this study is based on a tolerance function which considered to be less than \( 10^{-6} \).

\[ \text{Tolerance} = \frac{\sum_{i=1}^m \sum_{j=1}^n |\phi^{k+1} - \phi^k|}{\sum_{i=1}^m \sum_{j=1}^n |\phi^k|} < 10^{-6} \]  \hspace{1cm} (36)

In Eq. (36), \( \phi \) is a transport quantity. The indices \( k, m \) and \( n \) are the number of iterations and the number of grid points in the \( x \) and \( y \) directions, respectively.

To test and assess the grid independency of the solution scheme, an extensive mesh testing procedure was examined to determine the appropriate grid density for free convection inside the L-shaped cavity. The present code was tested for grid independency by calculating the average \( Nu \) of the hot walls at five different mesh combinations, \( 20 \times 20, 40 \times 40, 60 \times 60, 80 \times 80 \) and \( 100 \times 100 \). Table 2 presents the grid independency results for the case of \( AR = 0.4 \) filled with the Cu–water nanofluid with \( \varphi = 0.06 \) and zero inclination angle while the Rayleigh number is kept constant at

![Fig. 4. Isotherms profile in the L-shape enclosure with AR = 0.2 and \( \varphi = 0.06 \).](image-url)
$Ra = 10^5$. It is found that a grid size of $80 \times 80$ ensures a grid independent solution.

For model validation, the numerical results are compared with the available literature data. For this purpose, according to Fig. 1, the average Nusselt number for an enclosure at $AR = 0.25$ and for different Rayleigh numbers are considered. As indicated in Table 3, comparison of the present results with those reported by Tasnim and Mahmud, and Mahmoodi shows reasonable agreement [46,47]. As indicated in this table, the average Nusselt number in an air-filled L-shape enclosure at different Rayleigh numbers is approximately the same. Model validation is also conducted based on the presented results with Cu–water nanofluid in Ref. [47] at $AR = 0.4$, $\phi = 0.06$ and different Rayleigh numbers. The results in Table 4 indicate that the present model can be used to predict heat transfer rates in an L-shape enclosure.

4. Results and discussion

To investigate the heat transfer performance of L-shape enclosures, temperature and velocity profiles inside a thermal enclosure filled with Cu–water nanofluid with isothermal vertical walls are obtained. The effects of volume fraction of solid particles in the nanofluid, the aspect ratio of the enclosure and the Rayleigh number on the heat transfer rate are examined. The influence of structural modifications (e.g. insertion of pins) and inclination angle on heat transfer coefficient are also studied. The Rayleigh number, the aspect ratio of the enclosure and the solid volume fraction of the nanofluid are ranging from $10^7$ to $10^5$, $0.2-0.4$ and $0-0.1$, respectively. The results are presented in the following sections.

4.1. The effects of $Ra$, $AR$ and $\phi$ on free convection

Figs. 2–9 show the isotherm profiles in the L-shape enclosure with different parameters. As shown in Figs. 2–5, at an aspect ratio of 0.2, the isotherms are parallel for low Rayleigh number such as $10^7$ and $10^4$ however, as Rayleigh number is increased, the isotherms are disturbed. This means that at high Rayleigh numbers, the dominant heat transfer mechanism shifts from conduction to free convection. This behavior can also be observed in Figs. 6–9 with an aspect ratio of 0.4.

The velocity profiles are shown in Figs. 10–17. According to these figures, heated fluid moves upward toward the cold wall and some eddy are created. Also, the flow in the horizontal section of the enclosure is nearly stagnant. As shown in the figures, the number of eddies formed is increased and the flow circulation is augmented inside the enclosure at higher Rayleigh numbers.

The addition of nanoparticles to pure water influences the flow and temperature profiles by increasing viscosity and effective thermal conductivity of fluid. Examinations of Figs. 9–17 indicate that as $\phi$ increases the maximum velocity decreases due to increased density and the viscosity of the nanofluid. Though $k_{eff}$ increases with $\phi$, the effect of buoyancy will be decreased. At a particular $Ra$, because of the decrease in vertical velocity with increase in $\phi$, the strength of circulation reduces. On the other hand, at a fixed $\phi$ (the same viscosity and density), the

**Fig. 5.** Isotherms profile in the L-shape enclosure with $AR = 0.2$ and $\phi = 0.1$. 

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strength of circulation enhances as $Ra$ increases. A similar trend is reported by Mahmodi [47] for isotherm and velocity profiles at different solid volume fractions, aspect ratios and Rayleigh numbers.

The primary objective of using nanofluids in free convection systems is to enhance heat transfer rates. Local heat transfer coefficients are presented in Figs. 18–25. From these figures it is evident that for all range of the Rayleigh number, the heat transfer coefficient will be increased by increasing $\varphi$. Also, it is observed that at a constant $\varphi$, the maximum heat transfer coefficient at low Rayleigh numbers is more than that of high Rayleigh numbers. For example, in an enclosure with $AR = 0.2$ and $\varphi = 0.06$, the maximum heat transfer coefficients for $Ra = 10^3$, $10^4$, $10^5$ and $10^6$ are $6733.8$, $3067.4$, $1045.2$, and $1043.6 \text{ W m}^{-2} \text{ K}^{-1}$ respectively. The difference becomes negligible at higher $Ra$ numbers (e.g. between $Ra = 10^5$ and $10^6$). Aspect ratio is another important factor for increasing heat transfer. Comparison of various aspect ratios indicates that at $AR = 0.4$, since the circulations are more prevalent, the resulting heat transfer coefficients are higher than those associated with $AR = 0.2$. Also, at low Rayleigh numbers, the stagnant zone in the horizontal section of the enclosure with $AR = 0.4$ is smaller than that of $AR = 0.2$. Hence, higher heat transfer rates can be achieved at larger aspect ratios.

4.2. Performance analysis of the modified L-shape enclosure

In this part, the effects of possible structural modifications on an enclosure with $AR = 0.4$, $\varphi = 0.1$, $Ra = 10^6$ and different pin sizes are examined. Fig. 26 represents the effect of number and size of inserted pins on the isotherm profiles in an enclosure. As demonstrated in these figures, in general, insertion of pins of any sizes at any location will cause isotherms to approach to each other particularly near the enclosure walls and therefore temperature gradient will increase. It is also evident that application of larger pins creates more disturbances within the enclosure and as a result, closer isotherms. However, as shown in Fig. 27c and d, very large pins restrict flow circulation in the enclosure and as a result, decrease convective heat transfer. Fig. 27 indicates that insertion of three large diameter pins in the enclosures generates more eddies as compared with two pins with smaller diameter. Therefore, the number pins and their diameter are key parameters in enhancing free convection.

Local heat transfer coefficients in modified enclosures for different configurations at $AR = 0.4$, $\varphi = 0.1$ and $Ra = 10^6$ are presented in Fig. 28. Comparison of conventional (Fig. 25) and modified enclosures (Fig. 26d) shows that application of pins in the enclosure structure increases significantly the heat transfer coefficient from $984.3$ to $1093.8 \text{ W m}^{-2} \text{ K}^{-1}$.

In the present work, numerical simulations are conducted to study the effect of inclination angle on free convection in a two-dimensional inclined L-shape enclosure filled with Cu–water nanofluid. Fig. 29 shows variations of local heat transfer coefficients for different inclination angles. As shown in this figure, the maximum heat transfer coefficients in a modified enclosure with two pins and inclination angles of $\omega = 45^\circ$, $135^\circ$, $225^\circ$ and $315^\circ$ are $947.56$, $979.39$, $1260.2$ and $980.49 \text{ W m}^{-2} \text{ K}^{-1}$, respectively.

![Fig. 6. Isotherms profile in the L-shape enclosure with AR = 0.4 and \( \varphi = 0 \).](image)
Fig. 7. Isotherms profile in the L-shape enclosure with AR = 0.4 and $\varphi = 0.05$.

Fig. 8. Isotherms profile in the L-shape enclosure with AR = 0.4 and $\varphi = 0.06$. 
Fig. 9. Isotherm profile in the L-shape enclosure with AR = 0.4 and φ = 0.1.

Fig. 10. Velocity profile in the L-shape enclosure with AR = 0.2 and φ = 0.