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Numerical Analysis of Radiative Effects on Natural Convection in Asymmetrically Heated Vertical Convergent Channels

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Abstract —*This paper presents a numerical analysis of the effects of radiation on natural convection flow and heat transfer inside the asymmetrically heated convergent channels formed by two flat plates. One of the side walls is oriented vertically and is thermally insulated, and the opposite wall is tilted to various degrees to form convergent angles and is heated uniformly with a constant heat flux. Air (Pr~ 0.71) is considered as the working fluid. The flow is assumed to be steady-state, two-dimensional, laminar, and incompressible with negligible viscous dissipation. Results in terms of wall temperature profiles as a function of the wall inclination angle, channel aspect ratio and the heat flux are given for two values of the wall emissivity. Flow visualization is also carried out to show the peculiar pattern of the flow between the plates in several configurations. Dimensionless maximum temperatures are correlated to the Rayleigh number, in the investigated range from* 10^2 *to* 10^8 *and* $0° \leq \delta \leq 10°$.

Keywords-*Natural convection; vertical channel; convergent channel; asymmetrically heating; Radiative Effects*

I. INTRODUCTION

The heat transfer characteristics of natural convection in channels and parallel plates received a large attention in research community for its application in thermal control of various modern systems. These configurations have been employed in the cooling process of nuclear reactors, geothermal energy systems, heat exchangers, solar collectors and thermal control of electronic systems.

To reduce the chances of overheating in modern electronic devices by the large power dissipation, suitable methods are applied to carry away large amount of heat and to maintain reliable performance of the system. Usually the coolant is circulated between the parallel-plates passages formed by the mounting of electronics components on the arrays of vertical circuit boards and for easy handling, easy maintenance and reliability, air is preferred as a coolant. Under some circumstances the circuit boards maybe tilted slightly. This makes the flaw passage convergent or divergent. The convergence of the channel can accelerate the upward-flowing mainstream while the divergence can decelerate the flow. The convergence or divergence can change the heat transfer characteristics inside the channel because of their significant effects on the structure and occurrence of the reversed flaw.

Efforts have been made to enhance the thermal performance of geometrical simple configurations in natural convection [1-4]. Nowadays trends in natural convections are inclined more towards the optimization of simple configurations and the derived configurations from simple ones [4-9]. Among them the natural convection heat transfer in convergent vertical channel with one vertical insulated flat plate and other inclined uniformly heated flat plate is very important problem. Because of the large number of geometric and thermal parameters, it is difficult to evaluate the thermal performance of these kind of configurations. To predict the heat transfer characteristics and the fluid flow accurately the effects of surface radiation must be considered while analyzing a natural convection problem. Then it is interesting to investigate the radiative effects on thermal performance of natural convection in air in convergent channels.

An in-depth experimental study was carried out by Sparrow and Ruiz [10] in order to analyze the effects of natural convection in a vertical convergent-divergent channels. The principal walls were at uniform temperature and experiments were performed with water. Flow visualization indicated the presence of boundary layers adjacent to the channel walls with a recirculation loop occupying the remainder of the cross section. A single correlation between channel Nusselt and Rayleigh numbers for convergent, divergent and parallel walled channels was proposed. Khim et al. [11] carried out an experimental investigation of air natural convection in a converging channel with uniform wall temperature using a specklegram technique. The authors also proposed the correlations for both average and local Nusselt numbers. A numerical study of air natural convection in vertical convergent channels having uniform wall temperature was carried out by Said [12]. Using the maximum channel spacing as the characteristic length, the best correlation was achieved at Rayleigh numbers less than 10^2. The same problem given in [12] was investigated both experimentally and numerically by Shalash et al. [13]. The authors found out that for low Rayleigh numbers, if the convergence angle is increased, the Nusselt number also increases significantly and for High Rayleigh numbers, if the convergence angle is increased, the Nusselt number decreases. A numerical investigation on natural convection in isothermally heated converging channels with different convergence angles was carried out by Kaiser et al. [14]. The authors compared experimental and numerical results and also proposed a correlation for average Nusselt number In the Rayleigh number range 1 to 10^6 and in the convergence angle range 0° to 30°.Bejan et al. [15] carried out a numerical study on natural convection in air in vertical diverging and converging channels. Results showed that the Rayleigh number, based on wall length, was large when the angle between the two walls was approximately zero. A numerical investigation was carried out by Bianco and Nardini [16] on air natural convection in symmetrically heated convergent channel considering the conduction in the channel as well. The authors found out that at low Rayleigh numbers the wall temperature is significantly dependent on the convergence angle, whereas at high Rayleigh numbers the wall temperature is slightly dependent on convergence angle. Marcondes et al. [17] carried out a numerical investigation on natural convection in parallel, convergent and divergent channels with isothermal walls. Results were given for Prandtl number ranging from 0.7 to 88 and a correlation for average Nusselt number, Rayleigh number, maximum channel width, channel aspect ratio and Prandtl number was proposed.Bianco et al. [18] carried out a numerical study on the natural convection in a symmetrically heated convergent channel, with finite-thickness principal walls considering both the heat conduction and the effects of wall emissivity. Results showed that the radiation effects were more significant at low Rayleigh numbers. Bianco et al.[19] presented a graphical procedure to evaluate the thermal and geometrical parameters of air natural convection in symmetrically heated vertical convergent channels. Bianco et al. [20] carried out an experimental study on natural convection in air, in uniformly heated vertical convergent channels. The results showed that the higher the spacing, the lesser the effects of convergence angle. A good relationship was also observed between experimental flow visualization and stream function fields obtained numerically.

Dehghan and Behnia [21] found out that the effects of thermal radiation is significant in natural convection flows both in closed cavities and partially open-ended cavities.Carpenter et al. [22] carried out a numerical investigation on the interaction of radiation with developing laminar natural convection in an asymmetrically heated channel. The results showed that for black surfaces the maximum wall temperature was half of the value attained in the case of no radiation. Sparrow et al. [23] carried out a numerical study on radiative effects in a vertical channel with one plate adiabatic and the other at uniform temperature. Results showed that the effects of radiation heat transfer increased with the Grashof number. Moutsoglou and Wong [24] numerically studied the effects of uniform heat flux at the wall, channel width, channel height, surface emissivity and inlet air temperature in a vertical channel with arbitrarily prescribed heat fluxes on each wall. Yamada [25] carried out both analytical and experimental study considering the same configuration given in [24], with an absorbing and emitting medium. The results showed that the wall emissivity played a major role on combined heat transfer. Manca and Naso[26] found that the effect of the wall emissivity were appreciable in the case of asymmetrically heated channel. Moutsoglou et al. [27] numerically analyzed the natural convection– radiation cooling in a vented channel. The authors found out that the vents worsen the overall cooling process in a continuously heated channel. Sathe and Sammakia[28] carried out a numerical analysis for the thermal management of a tape ball grid array (TGBA) package. Krishnan et al. [29] carried out experimental and numerical study to investigate the multi-mode heat transfer between vertical parallel plates. An experimental correlation for combined convection and radiation between parallel vertical plates was given by Krishnan et al. in [30]. Mezrhab et al. [31] carried out a numerical study on radiation–natural convection interactions in air, in a vertical divided vented channel. The authors studied the effects of the vent opening position, the surface emissivity, size of the heat transfer and the flow structures inside the channel.Bouali and Mezrhab [32] studied the effects of surface radiation on developing laminar natural convective heat transfer in a divided channel. The authors found out that for Ra≥ 1600, the radiation plays a significant role on thermal field and at high Rayleigh numbers the mass flow rate and the average Nusselt number increased. Natural convection in air, in a convergent channel, uniformly heated at the principal walls, is experimentally investigated by Nicola Bianco et al. [33] in order to analyze the effects of the radiative heat transfer. The authors found that the effect of thermal radiation is more pronounced for larger convergence angles and for a wall emissivity equal to 0.90 and for small values of the minimum channel spacing, heat transfer in slightly convergent vertical channels is stronger than in a vertical parallel channel.

The aforementioned review points out that the radiative effects on air natural convection in asymmetrically heated vertical convergent channels has not been studied. Hence, in this paper a numerical study on the effects of radiation on thermal performance of air natural convection in asymmetrically heated convergent channels with one vertical insulated flat plate and opposite tilted heated slant plate is described. Results in terms of wall temperature distribution as a function of the angle of convergence, wall emissivity values, the heat flux and the channel spacing are presented. Nusselt numbers are correlated to the significant process parameters. Furthermore, flow visualization is carried out in order to show the peculiar pattern of the flow between the plates in several configurations.

II. PHYSICAL PROBLEM AND COMPUTATIONAL DOMAIN

2.1. Statement of the problem

The schematic view of the geometry considered in the present study is shown in the Fig. 1, where X and Y are the coordinates along the spacing and length of the channel respectively. The channel is made of two principle plates. One of the vertical plates is tilted slightly to form convergent angle, so that the channel is narrower at the exit than the entrance. Hence, the length of the tilted plate is more than the vertical one. The vertical wall is thermally insulated and the opposite tilted wall is heated over its entire surface with a uniform heat flux. It is open to the ambient along the top and the bottom edges of the channel.

The principle objective of the present work is to study the effects of the geometrical parameters as well as the thermal parameters on the heat transfer characteristic within the convergent channel. Hence, the length of the tilted flat plate is taken constant, L= 200×10^{-3} m throughout the study. The results are investigated for the different values of the ratio of the length of the tilted wall L to the minimum spacing between the two plates, Smin; convergent angle with the direction of the gravity, δ; Rayleigh number Ra and the different values of the emissivity of the adiabatic plate ε_a and heated plate ε_h.

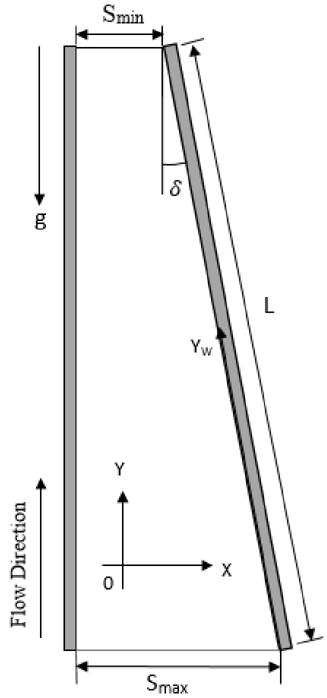

Figure 1. Geometrical configuration of vertical convergent channel

2.2. Mathematical description and governing equations

In this present study, the geometry consists of two non-parallel plates that form a vertical convergent channel is considered. The tilted plate is heated uniformly with constant heat flux. Both plates are assumed having gray surfaces and thermally conductive. The air flow is drawn into the channel due to the difference between the temperature of the heated plate and temperature of the ambient air. The flow in the convergent channel is assumed to be steady-state, two-dimensional, laminar, and incompressible with negligible viscous dissipation. Air (Pr~0.71) is considered as a working fluid. For the air density, the Boussinesq approximation holds good as the other thermophysical properties of the fluid are assumed to be constant and evaluated at the ambient temperature, To which is equal to 300 K. The external ambient was considered to be a black body at the temperature of 300 K.

Based on the above mentioned assumptions, the non-dimensional form of the governing equations for the convergent channel in steady-state and primitive variables for the fluid side can be written as follows:

Continuity:

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

X-momentum:

$$U\frac{\partial U}{\partial X} + V\frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \frac{1}{Re_s}\left(\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2}\right) \qquad (2)$$

Y-momentum:

$$U\frac{\partial V}{\partial X} + V\frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \frac{1}{Re_s}\left(\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2}\right) - \frac{Gr_S^*}{Re_S^2}\theta \quad (3)$$

Energy:

$$U\frac{\partial \theta}{\partial X} + V\frac{\partial \theta}{\partial Y} = \frac{1}{Re_s Pr}\left(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2}\right) \quad (4)$$

At the boundaries of the extended computational model, the pressure employed is an ambient pressure which is assumed equal to zero. In this present study, a two dimensional conductional model is employed.

The heat transfer equation in the converging channel considering steady-state conditions, with constant thermophysical properties and homogenous material can be given as,

$$\frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} = 0 \quad (5)$$

Reference to dimensionless quantities is made in the following. A dimensionless maximum wall temperature can be defined as,

$$\theta_{max} = \frac{(T_{max} - T_o)k}{q_w S} \quad (6)$$

Where q_w can be attained by dividing the overall heat rate by the total surface area of the wall surface, which was assumed to be uniform along the heated plate. Considering no heat losses due to the conduction to the ambient from the channel, q_w can be given as,

$$q_w = q_c + q_r \quad (7)$$

Where q_c is local convective heat flux and q_r is the local radiative heat flux from the plates. It is worth noticing that a separate evaluation of q_c from q_r is very difficult in practice. The terms q_c and q_r are evaluated by the following relations:

$$q_c = \frac{1}{L}\int_0^L q_c(y)dy \quad (8)$$

$$q_r = \frac{1}{L}\int_0^L q_r(y)dy \quad (9)$$

The net radiative heat flux from the surface is computed as sum of the reflected fraction of the incident and emitted fluxes

$$q_r(y) = (1 - \varepsilon_W)q_{in}(y) + \varepsilon_W \sigma T_W^4(y) \quad (10)$$

To compute the global heat transfer from the channel plate, q_w is taken into account. The channel modified Rayleigh number and average Nusselt number can be defined as,

$$Ra_S = \frac{g\beta q_w S^5}{v^2 kL} P_r \quad (11)$$

$$Nu_S = \frac{q_w S}{(T_w - T_O)k} \quad (12)$$

Where S is Smin or Sav or Smax and T_w is the average wall temperature.

$$T_w = \frac{1}{L}\int_0^L T_w(y_w)dy \quad (13)$$

The properties of the air are evaluated at the reference temperature $(T_w + T_O)/2$.

2.3. Boundary conditions

This problem is solved numerically considering a computational model of finite extent (Fig. 2) as two flat plates are placed in an infinite medium. The fluid zone is initialized as being at rest. This computational domain allows to account for the diffusive effects, peculiar in the elliptic model. The imposed boundary conditions are expressed mathematically in Eqs. (14) - (17).

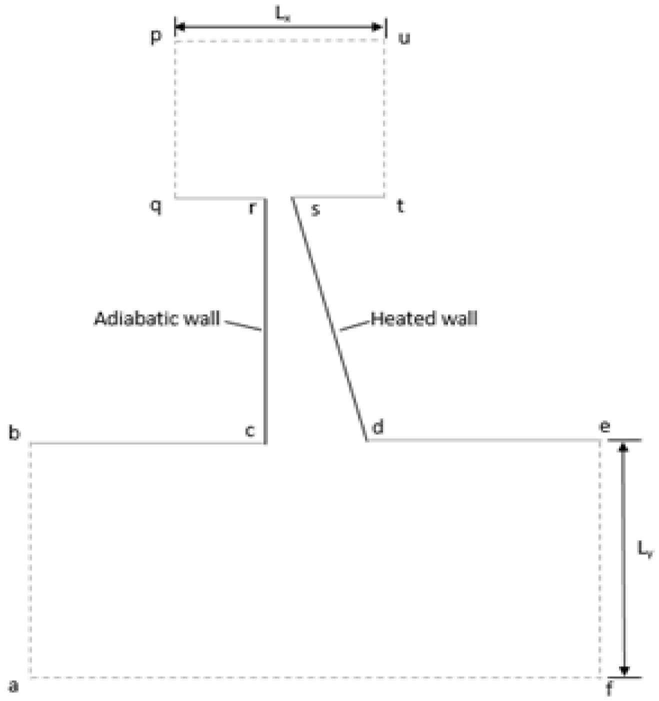

Figure 2. Computational domain of convergent channel

The important non-dimensional numbers for this problem are: aspect ratio-AR, Grashof number-Gr and the modified Rayleigh number-Ra*.

<u>For extended domains:</u>

The boundary conditions for vertical sides and horizontal sides of inlet and outlet extended domains are given in equation (14) and (15) respectively.

ab, ef, pq and ut: $\frac{\partial U}{\partial X} = 0,\ \frac{\partial V}{\partial X} = 0\ and\ \theta = 0$ (14)

af and pu: $\frac{\partial U}{\partial Y} = 0,\ \frac{\partial V}{\partial Y} = 0\ and\ \theta = 0$ (15)

<u>For vertical plate:</u>

Adiabatic boundary condition is imposed on the vertical plate of the convergent channel which can be given as,

cr: $\mathrm{U} = 0,\ \mathrm{V} = 0\mathrm{and}\frac{\partial \theta}{\partial Y} = 0$ (16)

<u>For tilted plate</u>*:*

Since the uniform heat flux is applied over the tilted plate of the convergent channel, the boundary condition imposed over it can be given as,

ds: $\mathrm{U} = 0,\ \mathrm{V} = 0$ and $q_w = -k_f \frac{\partial T}{\partial x}$ (17)

The reservoir at the minimum spacing of channel have horizontal (Lx) and vertical (Ly) lengths set equal to twenty times the minimum spacing distance between the walls while the for the reservoir at the maximum channel spacing are forty times the minimum spacing distance between the walls.

2.4. Solution Procedure

This numerical simulation has been carried out by means of the commercial FVM solver, FLUENT [34]. The segregated solution method is chosen to solve governing equations, which were linearized implicitly with respect to the equation's dependent variables. The second-order upwind scheme is chosen for the unsteady energy and momentum equations. The Semi Implicit Method for Pressure-Linked Equations (SIMPLE) scheme is chosen to couple pressure and velocity. Similar considerations were made for surface-to-surface radiation model (S2S) which assumes all surfaces to be diffuse and grey. A computation starts with zero value of velocity components and with temperature and pressure value equal to the ambient one. The convergence criteria of 1×10^{-6} for the residuals of the velocity components and of 1×10^{-8} for the residuals of the energy are assumed.

A grid independence study was carried out in order to obtain grid independent solutions. A non-uniform grid is used throughout the computational domain. Very fine grid is used near the walls to resolve boundary layers efficiently. This study was accomplished by monitoring the maximum temperature of a convergent channel with Smin=10 mm, δ = 4° at Ra_{Smin} = 537.525, ε_h = 0.9, ε_a = 0.9 and with Smin=40 mm, δ = 4° at Ra_{Smin} = 5.50×10^5, ε_h = 0.9, ε_a = 0.9. The maximum channel temperature was evaluated for three different grids with the following number of nodes in the channel: 18×100, 36×200 and 54×300. The details of the results of the grid independence study are given in Table: 1. When the total number of grid points in the channel increased from 36×200 to 54×300 a change of only 0.0056% for Smin = 10 mm, δ = 4° at Ra_{Smin} = 537.525, ε = 0.9 and 0.0283% For Smin = 40 mm, δ = 4° at Ra_{Smin} = 5.50×10^5, ε = 0.9 was observed on the maximum channel temperature. Hence, the numerical simulations were carried out using 36×200 nodes.

Table 1. Result of grid independence study

For S_{min} = 10 mm, δ = 4° at Ra_{Smin} = 537.525, ε_h = 0.9 and ε_a = 0.9		
No. of nodal points in the channel (m×n)	T_{max}(K)	Percentage change (abs)
18×100	334.307	-
36×200	334.378	0.0212
54×300	334.397	0.0056
For S_{min} = 40 mm, δ = 4° at Ra_{Smin} = 5.50×10^5, ε_h = 0.9 and ε_a = 0.9		
No. of nodal points in the channel (m×n)	T_{max}(K)	Percentage change (abs)
18×100	332.465	-
36×200	332.932	0.1403
54×300	333.026	0.0283

2.5. Validation of numerical procedure

The validation of the results for the asymmetrical heated convergent channel is accomplished by comparing the results obtained by the present study for the symmetrical heated convergent channel with those of Nicola Bianco et Al. [33]. Results of the average wall temperature profile along the heated wall for minimum spacing at channel inlet, bmin = 10 mm, emissivity value of both the plates, ε = 0.9, convergence angle, δ = 10° and for two values of uniform heat flux q = 30 w/m^2 and q = 220 w/m^2 (Fig. 3) were compared. From the figure, it is clear that the results of the present study agree very well with those of Nicola Bianco et Al.

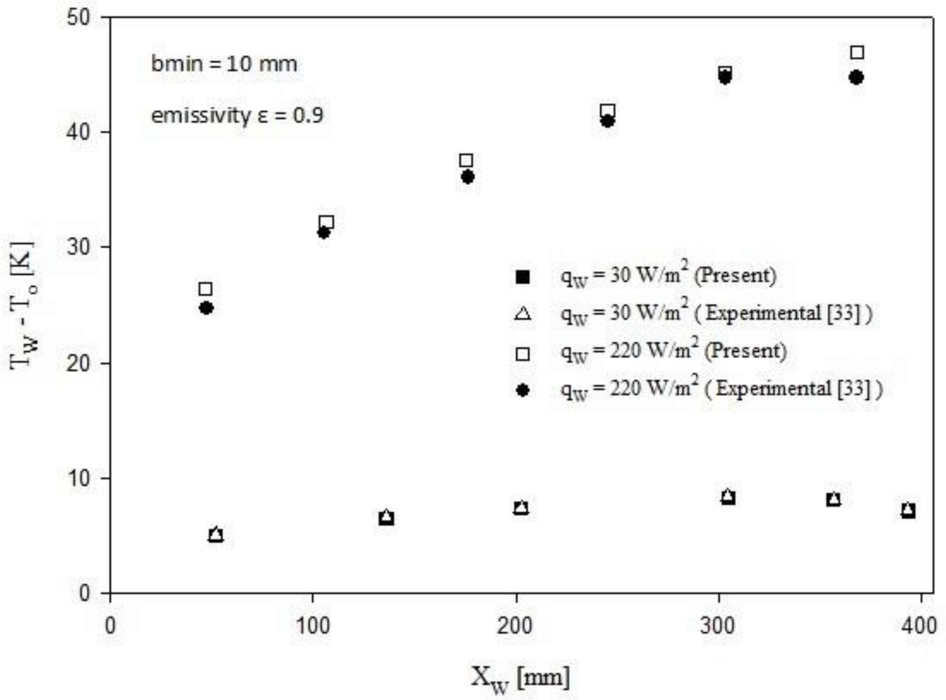

Figure 3. Wall temperature profile along the heated wall on contribution to Natural convection and Radiation in vertical convergent channel for bmin = 10 mm and emissivity value ε =0.9

III. RESULT AND DISCUSSION

Numerical analysis has been carried out with channel aspect ratio, L/Smin = 5, 10, 15 and 20 with the length of the heated plate constant, L = 20×10^{-3}m; convergence angle, δ = 2°, 4°, 6°, 8° and 10°; uniform heat flux, q_w = 100, 200, 300, 400 and 500W/m^2; emissivity of the adiabatic plate, ε_a = 0.1, 0.3, 0.5, 0.7 and 0.9 and for emissivity of the heated plate, ε_h = 0.1, 0.3, 0.5, 0.7 and 0.9.

3.1. Wall temperature profile

Wall temperature profiles as a function of the coordinate Y_w along the tilted wall of the channel, for aspect ratio, Ar = 5 and 20, δ = 4° and 10° and q_w = 100 and 500 W/m^{2}considering same emissivity of both the plates, ε = 0.10 and 0.90 are presented in the following to investigate the radiative effects in the channel. This is advantageous to remark the fluid dynamic and thermal behavior of natural convection in asymmetrically heated vertical convergent channel.

Fig. 4a reports the wall temperature profiles along the heated wall for aspect ratio = 5 and for convergence angle, δ = 4°. Results show that the wall temperature is arbitrarily having peaks and troughs along the length of the wall. As the heat flux decreases and the value of the wall emissivity increases, the difference between the maximum temperature and minimum temperature decreases. Same kind of results are found for aspect ratio = 5 and convergence angle, δ = 10° which is shown in Fig. 4b. The curve is found to be smooth in latter case compared to the former one, having only one peak near the inlet of the channel. For both the cases the maximum temperature and the difference between maximum and minimum temperature are very high for q_w = 500W/m^2, ε = 0.1 and they decrease with the decrease in the heat flux and increase in the wall emissivity. The reason of the arbitrariness of the wall temperature along the length of the wall is the oscillation of flow which is induced because of the flow reversal. In both of the cases the flow is found to be steady at the exit of the channel because of the no flow reversal. Flow reversal plays a significant role in the characteristics of heat transfer in vertical convergent channels having low aspect ratios. Because of the temperature difference, the minimum pressure is created inside the convergent channel which draws the air to move upstream along the thermally insulated plate from the downstream region and it makes the reversed flow unstable. This makes the interface between mainstream and reversed flow distorted and wavy. The reason for the transition of flow from laminar to turbulent is the vortices which are generated because of the strong mixing of the mainstream and the reversed flow. Due to the higher heat flux, the higher buoyancy force is generated along the wall which draws the reversed flow towards the top of the channel. The mainstream diminishes the effect of the reversed flow periodically. The highest point and the lowest point of the reversed flow move upstream and downstream respectively with increasing the heat flux i.e. buoyancy parameter of the system. Thus, higher the heat flux, higher will be the travel of the reversed flow. The reversed flow layers becomes highly unstable with increasing heat flux.

Fig. 5a and Fig. 5b reports the wall temperature profile along the heated wall for aspect ratio = 20, δ = 4° and for aspect ratio = 20, δ = 10° respectively. The results show the temperature distribution curves along the heated wall are very smooth without any peaks and troughs. It is worth noticing that the temperature along the heated wall increases and then becomes almost constant as the Y_w coordinate increases in both of the cases. The results also show that for low value of heat flux the difference between the temperature values for ε = 0.1 and ε = 0.9 are low because of the low channel spacing value that leads to a negligible radiation heat transfer between the plates and a small view factor between the plates and the surroundings. The effect of flow reversal is found to be negligible, and thus, the flow is found to be steady in the cases of heat transfer in the convergent channels having large aspect ratios. At the inlet of the channel the temperature is less because of the lower mass flow rate which is caused due to the weaker buoyancy force. The difference between the temperature values for the considered wall emissivity values at each location are larger for δ = 10° than for δ = 4° because of the larger view factor toward the surroundings.

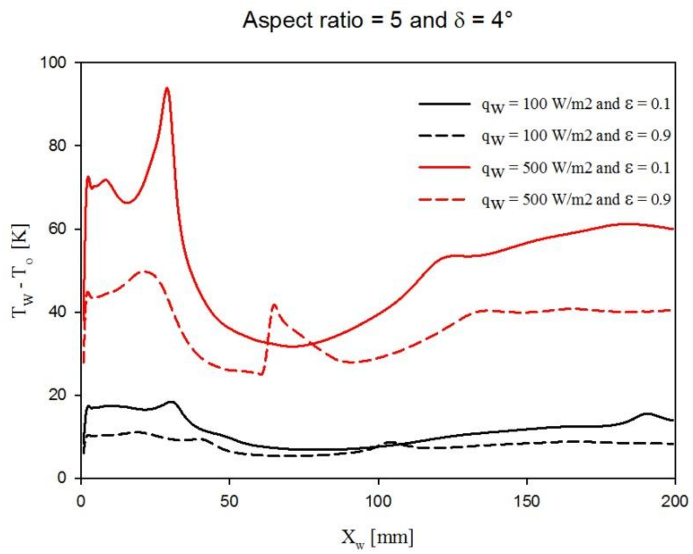

Figure 4(a). Wall temperature profiles along the heated wall for Ar = 5 and δ = 4°

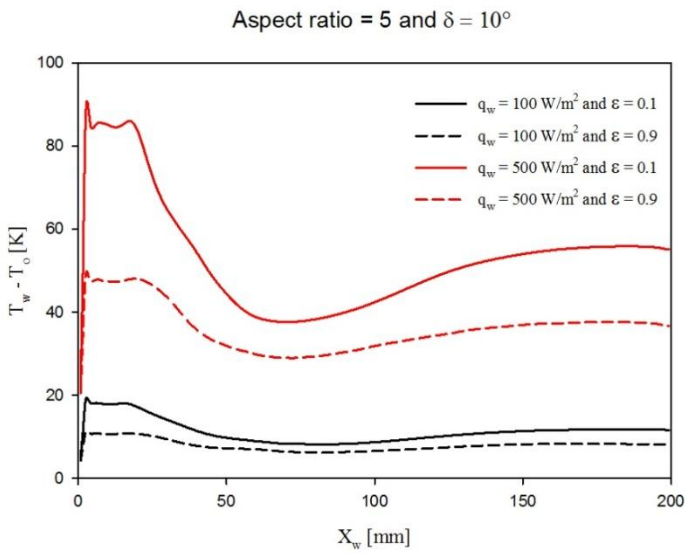

Figure 4(b). Wall temperature profiles along the heated wall for Ar = 5 and δ=10°

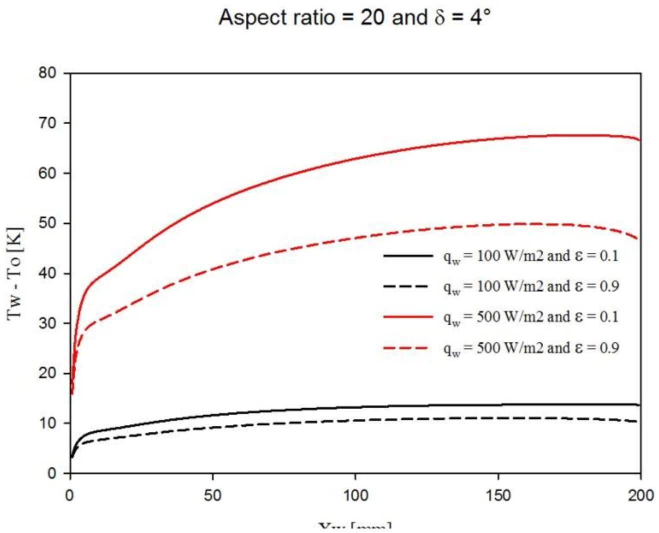

Figure 5(a). Wall temperature profiles along the heated wall for Ar = 20 and δ = 4°

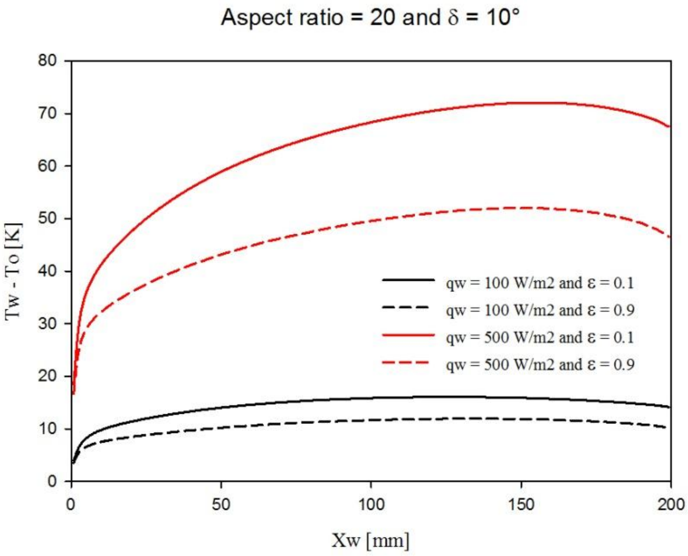

Figure 5(b). Wall temperature profiles along the heated wall for Ar = 20 and δ = 10°

3.2. Effect of modified Rayleigh number

Fig. 6 shows the streamline pattern for Ra_{Smin} = 5733.6, 17200.8 and 28668. The air is driven into the channel due to the low pressure created by the temperature difference between the heated plate and the adiabatic plate. As the modified Rayleigh number increases, the buoyancy force increases and hence the mass flow rate of the air passing through the channel increases. Since, higher the mass flow rate, higher the maximum temperature of the channel, and thus, higher the modified Rayleigh number, higher the maximum temperature of the channel. The temperature distribution in the convergent channel with aspect ratio = 10 and convergence angle, δ = 4° is shown in Fig. 7 for Ra_{Smin} = 5733.6, 17200.8 and 28668 considering both walls' emissivity = 0.5. At the inlet section, as fresh air comes into the contact with the heated plate convective heat transfer becomes large and hence at the inlet section the temperature of the heated wall is lower compared to the other parts. Apart from the convection, inlet and outlet sections are exposed to the open atmosphere and at these sections radiation heat transfer is high compared to interior parts. A radiation thermal boundary layer is formed parallel to the heated plate because of the radiation interaction. As the modified Rayleigh number increases, the effect of radiation increases and the thermal boundary layer becomes more pronounced. As expected, the non-dimensional maximum temperature increases as the modified Rayleigh number increases. Fig. 7 shows the variation of the non-dimensional maximum temperature with respect to the modified Rayleigh number while other parameters are at baseline values.

In this study the baseline parameters are taken as the reference parameters to develop the correlation. Hence, Ra_{Smin} = 17200.8, aspect ratio, Ar = 10, convergence angle, δ = 4°, emissivity of the heated wall, ε_h = 0.5 and emissivity of the adiabatic wall, ε_a = 0.5 are taken as reference values.

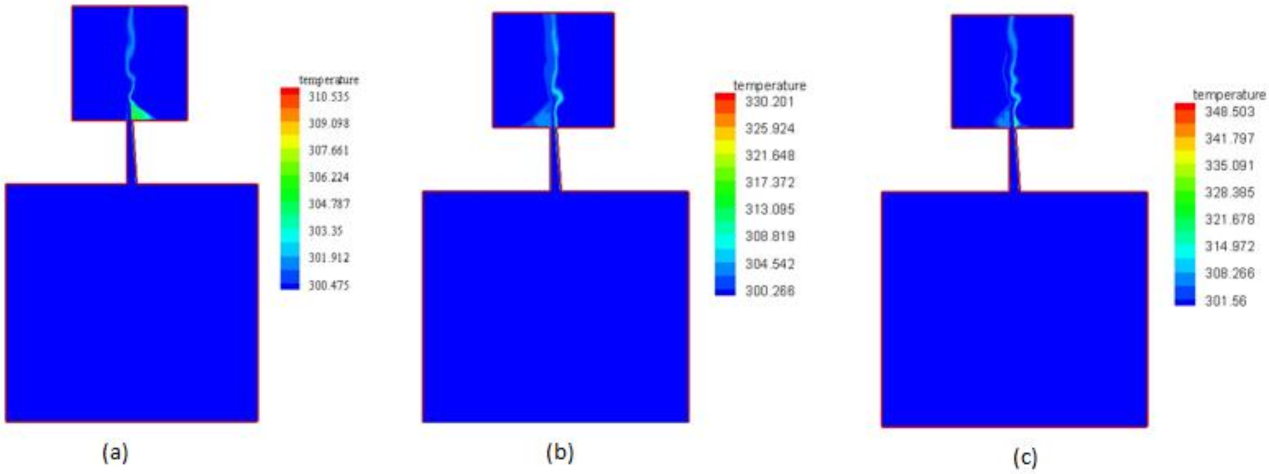

Figure 6. Streamline pattern for (a)Ra_{Smin} =5733.6, (b)Ra_{Smin} =17200.8 and (c)Ra_{Smin} =28668

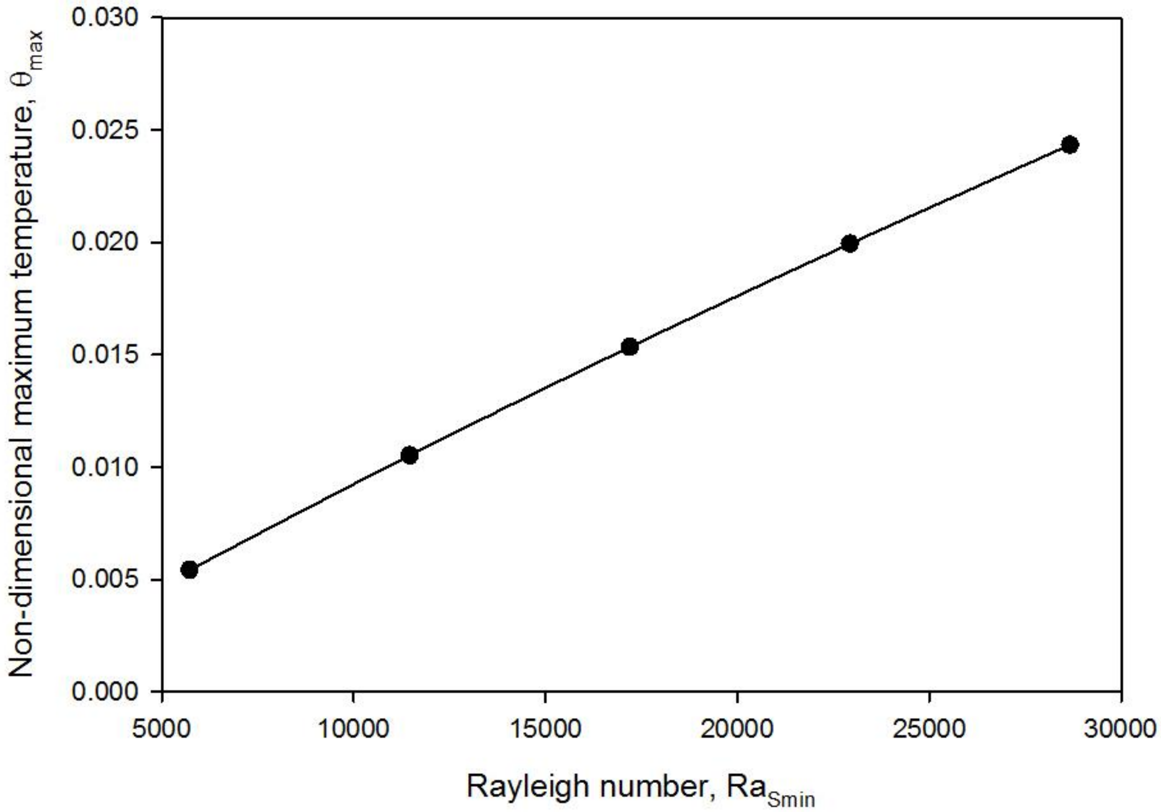

Figure 7. Variation of the non-dimensional maximum temperature with respect to the modified Rayleigh number

3.3. Effect of wall inclination

The effect of wall inclination on the heat transfer characteristics in the convergent channels has also been investigated. The higher the wall inclination, the narrower will be the channel at the exit. Fig. 8 shows the change in maximum temperature of the channel with respect to the angle of wall inclination. The results show that when the convergence angle is 2°, the maximum temperature of the channel is highest compared to the cases of higher convergence angles. The reason is the thermal radiation and view factor which are more pronounced in the cases of higher convergence angle and they lower the maximum temperature of the channel by weakening natural convection which is given in [35]. It can also be noticed that if the convergence angle is increased more than 6°, the maximum temperature of the channel also increases but in less amount and the reason is the higher mass flow rate of the air entering in the channel due to the higher pressure difference induced by higher convergence angle.

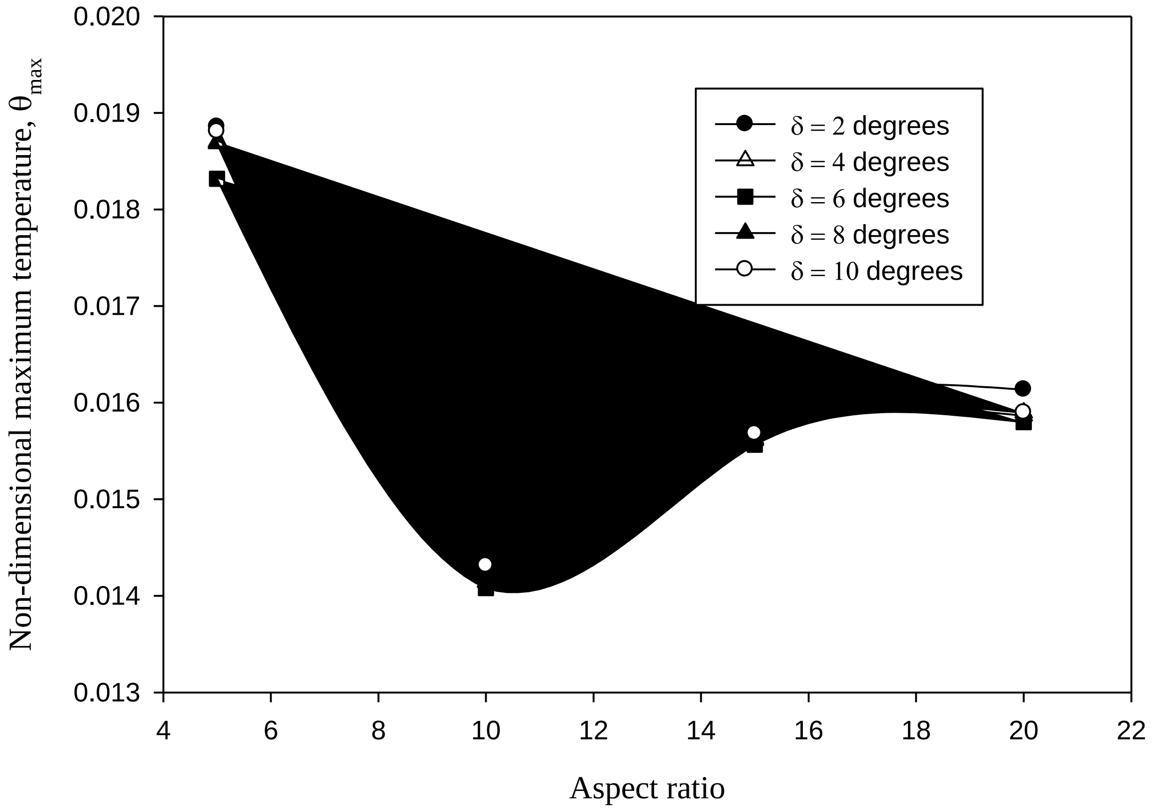

Figure 8. Effect of angle of inclination on non-dimensional maximum temperature

3.4. Effect of emissivity of heated plate (ε_h) and adiabatic plate (ε_a)

Fig. 9a and Fig. 9b shows the variation in non-dimensional maximum temperature in the convergent channel having aspect ratio = 10 and convergence angle = 4° with respect to the emissivity of the heated plate and adiabatic plate respectively for different values of $Ra_{S_{min}}$. The results show that with the increase of the wall emissivity the non-dimensional maximum temperature of the channel decreases because of an increase in radiation heat transfer which weakens the convective heat transfer according to [35]. As the radiation heat transfer from the heated plate or vertical adiabatic plate increases, the interaction between both the plates also increases. As the emissivity of the heated plate increases from 0.1 to 0.9 and the other values are kept at baseline values, the maximum temperature decreases from 338.7862 K to 329.8508 K. In the case of the vertical adiabatic plate, if the value of its emissivity increases from 0.1 to 0.9 the maximum temperature decreases from 334.9244 K to 331.1982 K. So, with the change of emissivity value of heated plate and vertical adiabatic plate from 0.1 to 0.9 the maximum temperature of the channel decreases about 9 K and 3.7 K respectively. From the above analysis it is clear that the effect of the emissivity of the heated wall is more significant than the emissivity of the adiabatic wall and it is also shown in the Fig. 10 where all other parameters are at baseline values.

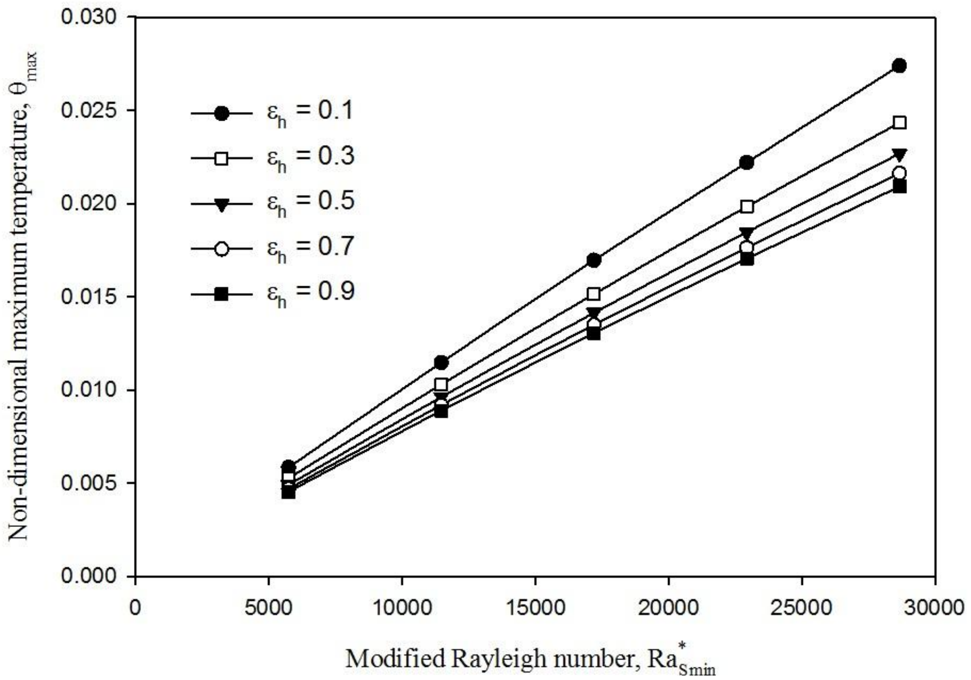

Figure 9(a). Variation in non-dimensional maximum temperature with respect to the emissivity of heated plate

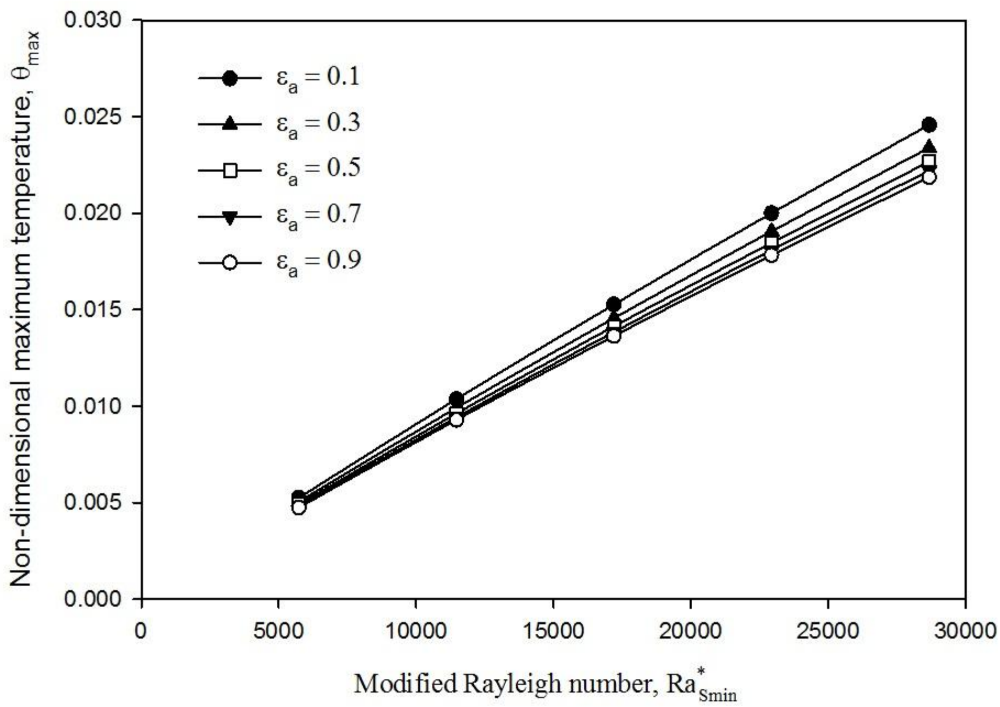

Figure 9(b). Variation in non-dimensional maximum temperature with respect to the emissivity of adiabatic plate for different values of Ra_{Smin}

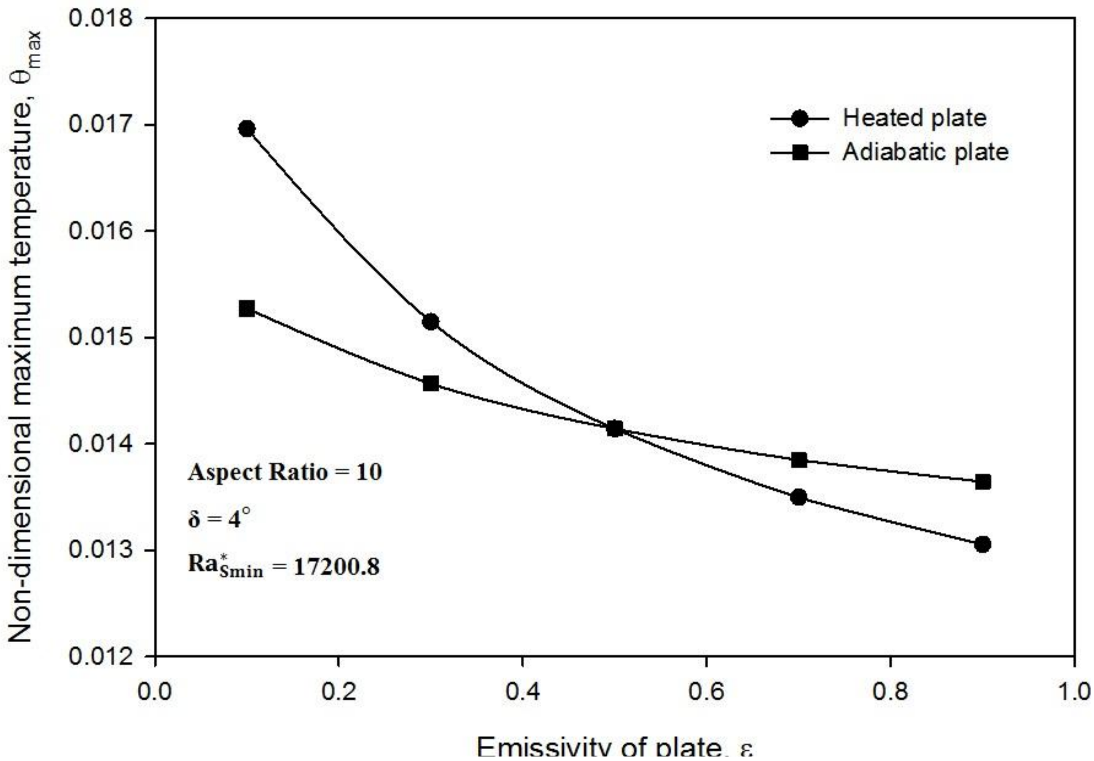

Figure 10. Effect of change in emissivity of heated plate and adiabatic plate over the non-dimensional maximum temperature

3.5. Effect of aspect ratio

Aspect ratio is the ratio of the length of the tilted wall to the minimum spacing between two plates at inlet. The effect of this aspect ratio on the heat transfer characteristics has also been investigated. Fig. 11 shows the effect of aspect ratio on the non-dimensional maximum temperature for various values of heat flux while all other parameters are at baseline value. The results show that for low aspect ratio the value of non-dimensional maximum temperature is more compared to higher aspect ratios. It can also be noticed that for low values of heat flux the deviations in non-dimensional maximum temperature with respect to aspect ratio is low but as the value of heat flux increases the deviation also increases. As the value of heat flux increases, the modified Rayleigh number and the buoyancy parameter increase which causes the higher mass flow rate of air to pass through the channel. As mentioned earlier, reason for the higher non-dimensional maximum temperature in the case of the low aspect ratios is the flaw reversal. So, in the case of low aspect ratio and higher heat flux the deviation in non-dimensional maximum temperature is high because of the combined effect of the both higher mass flow rate and flow reversal. It is also noticeable that after one value of the aspect ratio if its value is increased, non-dimensional maximum temperature increases and become almost constant. The reason is the absence of flow reversal in case of higher aspect ratios and the convective heat transfer which is higher than the radiative heat transfer.

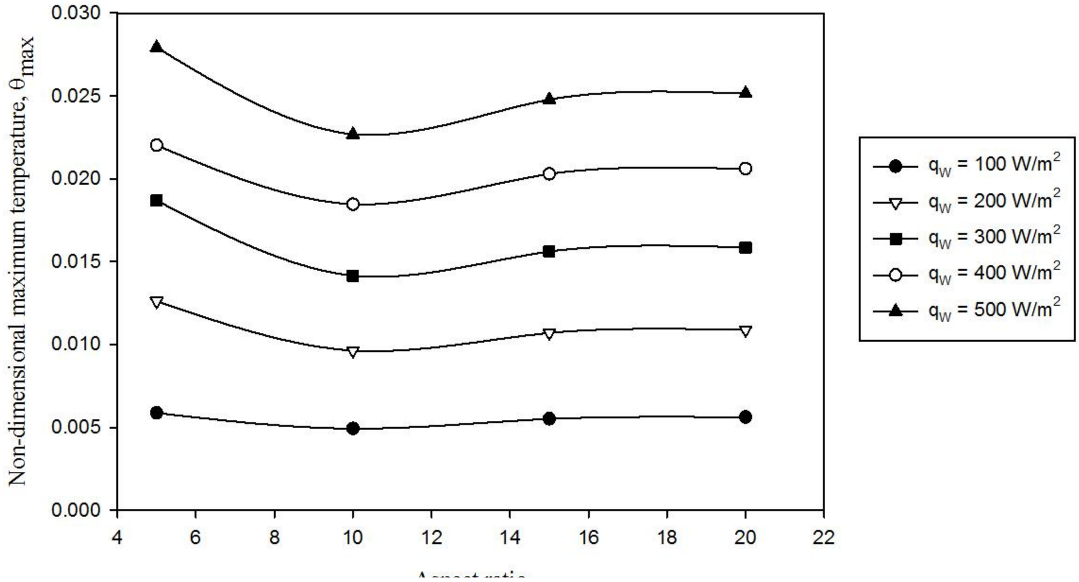

Figure 11. Effect of aspect ratio on the non-dimensional maximum temperature for various values of heat flux

IV. CONCLUSIONS

A numerical investigation on air natural convection in a convergent channel with asymmetric heating was accomplished, in order to analyze the effects of the radiative heat transfer. The vertical wall of the channel was thermally insulated and tilted wall was heated over at its entire surface with uniform heat flux. Results in terms of wall temperature profile along the tilted wall of the channel, as a function of heat flux, aspect ratio and wall inclination angle were given for two values of the wall emissivity, $\varepsilon = 0.10$ and 0.90. Flow visualization at three different modified Rayleigh numbers was carried out in order to show the flow pattern between the plates. Dimensionless maximum temperatures are correlated to the Rayleigh number, in the investigated range from 10^2 to 10^8 and $0° \leq \delta \leq 10°$.

Based on the parametric study, the following conclusions are arrived at:

1. The flow reversal plays a significant role in the characteristics of heat transfer between the plates of converging channel having low aspect ratio. Due to the strong mixing of the mainstream and the reserved flow, the transition of the flow from laminar to turbulent takes place. This is why the maximum temperature of the channels having low aspect ratios is higher compared to the maximum temperature of the channels having higher aspect ratios.
2. The wall temperature profile along the heated plate of channel having low aspect ratio is found arbitrary whereas in the case of the channel having high aspect ratio, the wall temperature profile along the heated plate is found smooth.
3. The non-dimensional maximum temperature increases non-linearly as the modified Rayleigh number increases. The reason is the increased mass flow rate driven in the channel at higher values of modified Rayleigh number.
4. The non-dimensional maximum temperature in the case of convergent channel having convergence angle, $\delta = 2°$ is found maximum compared to the channels with higher convergence angle because of the thermal radiation and view factor. The effect of the thermal radiation and view factor diminish with increase in mass flow rate driven between the plates of the channel.
5. As the emissivity of the heated plate or insulated plate increases the non-dimensional maximum temperature decreases because of the increase in the radiation heat transfer which weakens the convective heat transfer. The effect of emissivity of the heated plate is found more significant compared to the emissivity of the adiabatic plate.
6. While carrying out a thermal analysis of stack of circuit boards with electronic chips, the consideration of radiation heat transfer is absolutely essential to predict the non-dimensional maximum temperature accurately.

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