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Numerical Study of Heat Transfer Performance of Homogenous Nanofluids under Natural Convection

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***Abstract*—The present study aims to identify heat transfer and flow characteristic due to buoyancy forces in a heated enclosure using nanofluid and their behaviors under natural convective heat transfer condition. In the present work nanofluids with water based containing Al_2O_3 nanoparticle numerically investigated. Numerical works are done on the use of the stable nanofluids under natural convective heat transfer conditions. Process of heating is done in two different ways: in first process the heater mounted to the down wall and in second way it mounted to the left vertical wall with a finite length, also heated and cooled walls keep in a constant temperature. Our numerical simulation has been undertaken incorporating a homogenous solid-liquid mixture. In particular this study deals with Al_2O_3 nanofluids with Newtonian behavior. Simulation have been carried out in the ranges Ra=10^3-10^6. Our volumetric fraction of nanoparticles was 1.3%. It was shown the Nusselt-Rayleigh number relation and then nanofluid Nu-Ra number diagrams based on found is plotted. Results showed an increasing in Nusselt-Rayleigh number at nanofluids diagrams as compared to Nusselt-Rayleigh relations of pure water. Increase in the average Nusselt number plays a significant role in heat transfer applications. Due to our numerical investigations vertical cavities with nanofluid were better than horizontal cavities. Also the cavities, which we used nanofluid, had better efficiency in natural convection numerical modeling for both horizontal and vertical fluid layer.**

***Index Terms*— Natural convection, Nanoparticle, Nanofluid, Homogenous.**

I. Introduction

Physics of natural convection has long been a subject of experimental, theoretical, and computational studies, the challenge lies in the complexity of natural convection flows, with various patterns emerging from seemingly chaotic dynamics. Also for natural convection heat transfer characteristics of nanofluids, relatively few research efforts have been undertaken. In this field Putra et al. [1], Wen et al. [2], Chang et al. [3] experimentally investigated some uncertainties about using nanofluids in practical applications. Khanafer et al.[4], Wen and Ding [5], Wang et al. [6], Rong et al. [7], Abu-Nada et al. [8] and Chen et al. [9], numerically investigated the influence of using nanofluids on heat transfer performance of conventional base fluids, separately. They investigated the influence of various uncertainties of using nanofluids instead of conventional base fluids.

There is no doubt that adding nanoparticles will increase the thermal conductivity of the base fluid. And it seems that, it could be one of the major reasons responsible for the heat transfer enhancement of nanofluids. So improving of the thermal conductivity is the key idea to improve the heat transfer characteristics of conventional fluids. Since nanoparticles have a larger thermal conductivity than a base fluid, suspending nanoparticles into the base fluid is expected to improve the thermal conductivity of that fluid. The enhancement of thermal conductivity of conventional fluids by the suspension of solid particles, such as millimeter- or micrometer-sized particles, has been well known for more than 100 years.

Many researchers have reported experimental studies on the thermal conductivity of nanofluids. The results from all of the available experimental studies indicated that nanofluids containing a small amount of nanoparticles have substantially higher thermal conductivity than those of base fluids. Al_2O_3 and CuO are the most well-known nanoparticles used by many researchers in their works. Even when the size of the particles and type of base fluids are different, all the experimental results showed the enhancement of the thermal conductivity.

Lee et al. [10] measured the thermal conductivity of nanofluids. The number-weighted particle diameter and the area weighted particle diameter used were 18.6 and 23.6nm for CuO, and 24.4 and 38.4nm for Al_2O_3, respectively. These particles were used with two different base fluids: water and ethylene glycol to get four combinations of nanofluids. The nanofluids showed substantially higher thermal conductivities than those of the same liquids without the nanoparticles. The thermal conductivity of suspended CuO in ethylene glycol showed an enhancement of more than 20% at 4% volume fraction of nanoparticles. The thermal conductivity ratios increased almost linearly with an increase in volume fraction. The experimental results revealed that the thermal conductivity of nanofluids was dependent on the thermal conductivity of both the particles and the base fluids.

Wang et al. [6] used the steady-state parallel-plate technique to measure the thermal conductivity of nanofluids containing Al_2O_3 and CuO nanoparticles. The particles were dispersed in water, ethylene glycol, vacuum pump oil and engine oil. Experimental data showed that the thermal

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conductivity of all nanofluids were higher than those of their base fluids. The thermal conductivity of the nanofluids increased with increasing volume fraction of the nanoparticles. For a specific volume fraction, the increase of thermal conductivity was different for each base fluid.

John Philip et al. [11] experimentally investigated nanofluid suspensions behavior. They have been found that the nanoparticles are singlephase with monoclinic structure if nanofluids have well treated. They reported that the sedimentation in CuO nanofluid is observed when the nanoparticles concentration was above 1 vol. %. Below 1 vol. %, the CuO nanofluids were quite stable for more than after 3 week. To produce a stable nanofluid, either the particles size should be small enough to be suspended by Brownian motion or the particles must be protected against aggregation by electric charge or other protective coatings. The stability of nanoparticles in the fluid has a major impact on the effective thermal conductivity of the fluid.

We want to numerically investigate the differences between Nu-Ra correlations in homogenous model of nanofluids as compared with pure water as a base fluid. This work aims principally to numerically investigate stable nanofluids that can be used under different conditions in a stable condition. We numerically investigated the heat transfer behavior of nanofluids (water-Al_2O_3) in a two-dimensional horizontal and vertical enclosure. As the size of Al_2O_3 nanoparticle is very small, even at this volume fraction, their suspensions in water are stable. The above short review shows that heat transfer behavior of nanofluids is very complicated and application of nanofluids for heat transfer intensification should not be decided only by the effective thermal conductivity. We also must be considered that, Nanofluids prepared through dispersing dry nanoparticles generally have stability problem. It was found that by using some special treatments on nanofluid suspensions, we can improve the stability problem of the nanofluids suspensions. These nano-properties of nanoparticles, host liquids, surfactant/dispersant used, as well as the extent of agglomeration of nanoparticles and the instability of nanofluids could result in similar problems as encountered in micron or millimeter sized particulate dispersions, and leading to sedimentation and even clogging of the system . Also it must be noticed that, for a same type of nanofluid, results presented by different authors exhibit considerable dispersion. Possible reason for this could be particle size/shape effects, temperature effects, particle interaction effects, etc.

II. MATHEMATICAL FORMULATION

A. Problem description

A schematic of the two-dimensional system is shown in Fig. 1. The cavity is differentially heated, both walls are isothermal at T_h and T_c respectively (T_h>T_c) and two other walls are adiabatic.

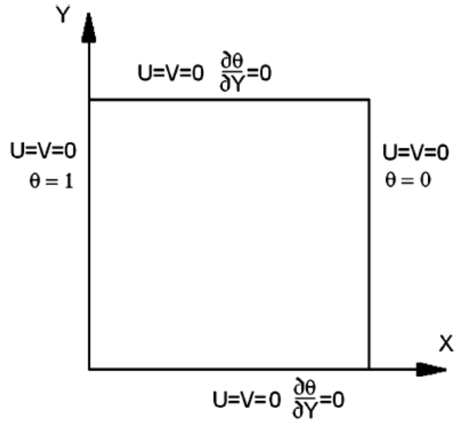


Fig.1. Schematic of the square cavity with the coordinate system and boundary Conditions

The modeling of the convection problem is made with the buoyancy term in the average momentum equation. The fluid properties are assumed to be constant except the density in the body force term which was determined according to the Boussinesq approximation [12]. For many natural convection flows, you can get faster convergence with the Boussinesq Model than you can get by setting up the problem with fluid density as a function of Temperature. This model treats density as a constant value in all solved equations, except for the buoyancy term in the momentum equation:

$$(\rho-\rho_0)g \approx -\rho_0\,\beta(T-T_0)g \tag{1}$$

This equation obtained by using the Boussinesq [12]:

$$\rho=\rho_0(1-\beta\Delta T) \tag{2}$$

The Boussinesq approximation [12] is valid when:

$$\beta(\mathrm{T}-\mathrm{T}_0) \ll 1 \tag{3}$$

A steady state situation is achieved when the temperatures of the plate and the ambient fluid are held constant (nature of the problem for both cases). Let’s look at non-dimensional equations which are used to solve natural convection flows:

Continuity equation:

$$\frac{\partial \mathrm{U}}{\partial \mathrm{X}}+\frac{\partial \mathrm{V}}{\partial \mathrm{Y}}=0 \tag{4}$$

Momentum equation:

$$\mathrm{U}\frac{\partial \mathrm{U}}{\partial \mathrm{X}}+\mathrm{V}\frac{\partial \mathrm{U}}{\partial \mathrm{Y}}=-\frac{\partial \mathrm{P}}{\partial \mathrm{X}}+\mathrm{Pr}\nabla^2\mathrm{U} \tag{5}$$

$$\mathrm{U}\frac{\partial \mathrm{V}}{\partial \mathrm{X}}+\mathrm{V}\frac{\partial \mathrm{V}}{\partial \mathrm{Y}}=-\frac{\partial \mathrm{P}}{\partial \mathrm{X}}+\mathrm{Pr}\nabla^2\mathrm{U}+\mathrm{RaPr}\theta \tag{6}$$

Energy equation:

$$\mathrm{U}\frac{\partial \theta}{\partial \mathrm{X}}+\mathrm{V}\frac{\partial \theta}{\partial \mathrm{Y}}=\nabla^2\theta \tag{7}$$

Where the non-dimensional parameters are given as bellow:

$$\mathrm{X}={}^{\mathrm{x}}/_{\mathrm{L}}\ ,\ \mathrm{Y}={}^{\mathrm{y}}/_{\mathrm{L}}\ ,\ \mathrm{U}={}^{\mathrm{uL}}/_{\alpha}\ ,\ \mathrm{V}={}^{\mathrm{vL}}/_{\alpha}$$

$$\theta = {(T - T_c)}/{(T_h - T_c)}$$

Natural convective heat transfer may be simulated by the terms of the Grashof and Prandtl numbers. The Grashof number indicates that the ratio of the buoyancy force to the viscous force acting on the fluid and plays the same role in the natural convection that the Reynolds number plays in forced convection heat transfer. The Prandtl number represents that the ratio of the momentum and thermal diffusivities. These numbers are defined as:

$$Gr = \frac{g\beta\Delta T L^3}{\nu^2} \tag{8}$$

$$Pr = \frac{\nu}{\alpha} \tag{9}$$

Where L is a characteristic dimension of the region, for example, the depth of the gap, β, υ and α are the fluid volumetric and thermal diffusivity, respectively, and g is the acceleration due to gravity.

The overall heat transfer phenomenon may be characterized by the Nusselt and Rayleigh numbers which are defined by:

$$Nu = \frac{hL}{k} \tag{11}$$

$$Ra = \frac{g\beta\Delta T L^3}{\nu\alpha} \tag{12}$$

Also correlation between local and average Nusselt number is defined by:

$$\overline{Nu} = \frac{1}{L}\int_{s=0}^{s=L} Nu(s)ds \tag{13}$$

B. BOUNDARY CONDITIONS

The boundary condition of the system shown are U=V=0 on all solid surfaces. The no-slip condition is valid at all boundaries as there is no cross flow. Thus,

On heated wall U=V=0, θ=1 (14)

On cooled wall U=V=0, θ=0 (15)

On the adiabatic walls: U=V=0, $\frac{\partial\theta}{\partial s} = 0$ (16)

III. Grid Sensitivity Testing

In order to ensure grid-independence solutions, several uniform grids have been tested. Results have been compared for selection of the best independent tested grid out of several other grids. Different meshes varying from 40×40 to 135×135 have been tested. An 80×80 grid was used for all cases presented in this paper. This choice was deemed conservative and does not cost much in terms of computational times.

IV. Validation

The real physical model with ideal situation of the enclosure, which represents two-dimensional rectangular object with differentially heated sides and adiabatic walls, has been defined in order to predict good enough results. Solution of the defined mathematical model with respect the nature of the equations has been done with numerical control volume method. We used SIMPLE procedure, which contains routines for solving variables field. Results of our numerical experiment have been compared with other results, and it gives a reliable agreements. In fig. 2 we compare natural convective heat transfer characteristics resulted from our numerical model in a rectangular cavity with experimental data of Chang et al. [3], Rossby [13] and Holland's et al. [14]. It must be noted that our numerical results, which compared with their experimental results were in very well agreement with their results. On the other hand because our numerical results were near to the experimental results, and Nu-Ra relations difference were almost negligible, also with the almost the same tendency, so we can conclude that we are able to use numerical methods results to obtaining Nu-Ra relation for a stable nanofluid.

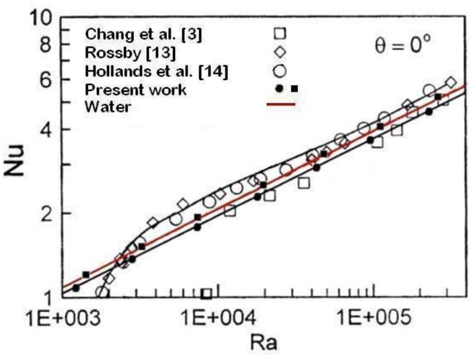


Fig.2. Nusselt number results for horizontal fluid layer (validation)

V. Results and Discussion

It must be considered that, by using a very small concentration of particles, also because the particles are small and weight less, nanofluids will completely maintained the Newtonian behavior of the fluid and nanofluids have been stable over months. However nanoparticles have a high tendency to re-agglomerate in dispersions due to the nature of attractive London van der Waals force among particles, which must be considered. So in nanofluids production procedure, some surfactants and/or dispersants were often used to enhance barriers between nanoparticles, therefore to stabilize the nanofluids. However cares must be taken as these surfactants/ dispersants could have significant influence on the flow and heat transfer, and some also may fail under high temperatures.

In this study, the volume fraction of Al_2O_3 nanoparticle suspension in water was 1.3%. The primary particle size which we analyzed was less than 10 nm. Nanofluids production processing could significantly influence their behaviors. So the obtained trends by different researchers in the natural convection are not obvious. As a complementary work in this study, we used homogenous properties of nanofluids to simulate the natural convection of nanofluids in a two-dimensional enclosure. Due to the extreme particle size, it seems reasonable to assume that such a mixture tend to behave more like singlephase fluids and therefore, we may assume that the motion slip between the phases would be negligible. We used colloid suspensions properties, which are stable against coagulation and solid particle do not settle with the time because of their Brownian motion. We want to analyze Nu-Ra relations of Al_2O_3-water nanofluid inside an enclosure in homogenous solid-particle dispersion.

In fig. 3 the natural convection heat transfer characteristics, which resulted from using nanofluids, is compared with obtained pure water results in a vertical rectangular cavity. The nanofluids were assumed to be in singlephase, in thermal equilibrium and without velocity slip between base fluid and solid particles.

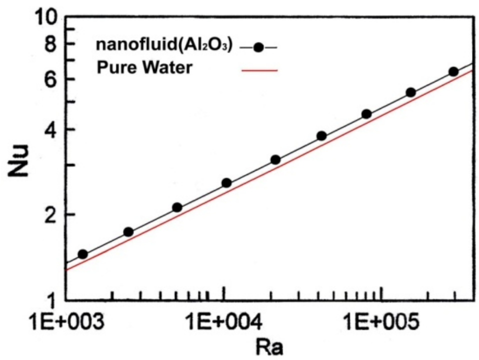


Fig.3. Comparison Nusselt number results for nanofluid and pure water

Figs. 4 and 5 show the comparison of the mean Nu-Ra relations for horizontal and vertical fluid layer for nanofluid and pure water, respectively. According to these results vertical cavities, which is heated and cooled form left and right walls were better than horizontal cavities, which is heated and cooled from down and top walls. However, the analyses of the results show that heat transfer performance of vertical cavities in natural convection, for both nanofluid and pure water, were higher than horizontal cavities.

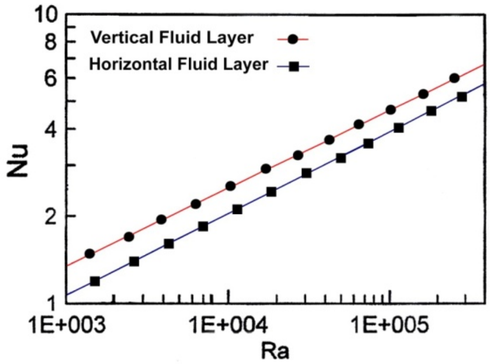


Fig.4. Comparison Nusselt number results for horizontal and vertical fluid layer (for nanofluid)

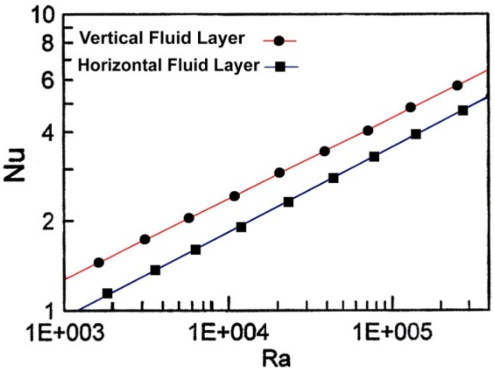
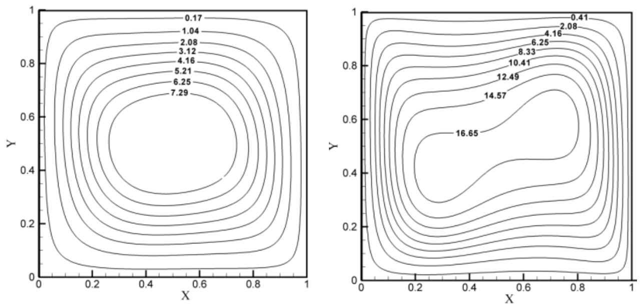


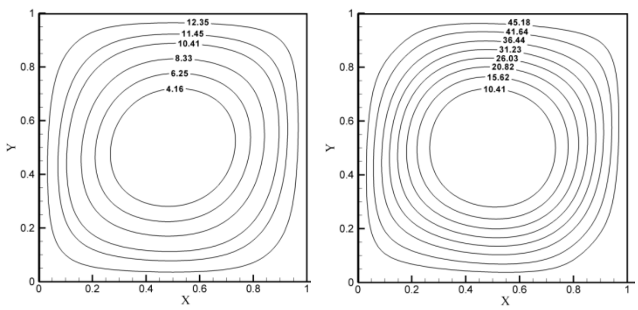
Fig.5. Comparison Nusselt number results for horizontal and vertical fluid layer (pure water)

In figs. 6 and 7 the effects of using nanofluids on the stream function counters at Ra=10^4 and Ra=10^5 for vertical and horizontal cavities are shown respectively. Streamline counters is the best tool to analyze and understand the flow properties in two-dimensional convective transport processes. These figures are shown that, in Ra=10^5, the central vortex tends to become elliptic. Also based on the physical properties, the steamlines give clockwise circulation pattern. At Ra=10^5, deformation tendency for streamline counters is more than Ra=10^4 deformation. Fig 6b shows that for Ra=10^5 in vertical fluid layer, the primary vortex starts dividing into two vortex.



a b

Fig.6. Counters of stream function for vertical layer of a)Ra=10^4 b) Ra=10^5

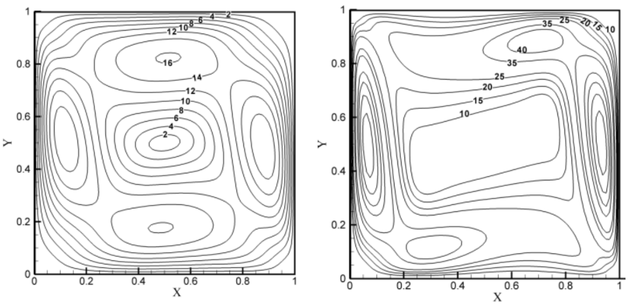


a b

Fig.7. Counters of stream function for horizontal layer of a)Ra=10^4 b) Ra=10^5

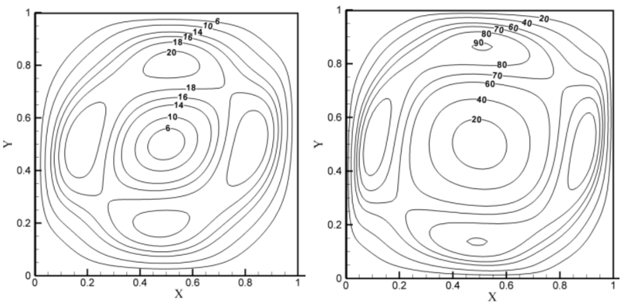
In figs. 8 and 9 the effects of using nanofluids on the velocity magnitude counters at Ra=10^4 and Ra=10^5 for horizontal and vertical fluid layer is shown respectively. Velocity magnitude counters are important to analyze fluid motion. Velocity magnitude counters represent velocity magnitude gradients for natural convective heat transfer. Also it is observed that velocity magnitude is less dense at the center points. For Ra=10^5 the fluid moves faster as the

Rayleigh number increase and velocity magnitude contours become more denser closer to hot and cold walls.



a b

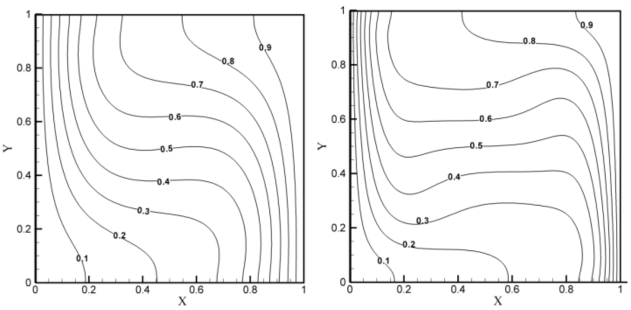
Fig.8. of velocity magnitude for vertical layer of Ra=10^5
a)Ra=10^4 b) Ra=10^5



a b

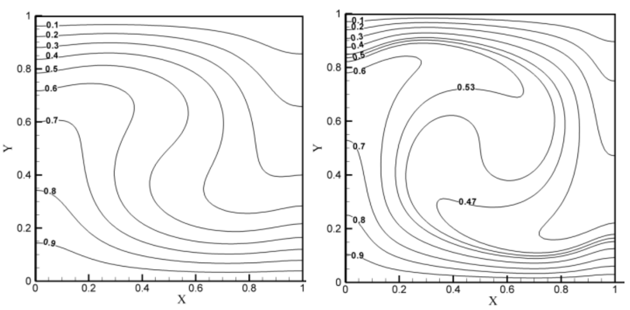
Fig.9. Counters of velocity magnitude for horizontal layer of
a)Ra=10^4 b) Ra=10^5

In fig 10 and 11 as the Rayleigh number is increased to 10^5, due to stronger circulation appear for both horizontal and vertical fluid layer, isotherms lines tend to deform more. These results caused more deformation in temperature counters. For higher Rayleigh numbers, natural convection become more vigorous and the fluid moves faster and the isotherms become more packed next to adiabatic walls. This is due to vortexes becoming more vigorous and results in a very small temperature difference.



a b

Fig.10. Counters of temperature fields for vertical layer of
a)Ra=10^4 b) Ra=10^5



a b

Fig.11. Counters of temperature fields for horizontal layer of
a)Ra=10^4 b) Ra=10^5

VI. Conclusion

A numerical study of the heat transfer enhancement capabilities of nanofluids in natural convection has been presented. Results have clearly shown that the use of stable nanofluids can significantly increase heat transfer capabilities in both horizontal and vertical cavities. The major findings contained in this paper are as follows: by using nanoparticles on the base fluid, the ratio of the Nusselt number of nanofluids to the base fluid is increased. By increasing Rayleigh number, deformation tendency for streamline counters in vertical fluid layer is more than horizontal fluid layer. But by increasing Rayleigh number, deformation tendency for isotherms in horizontal fluid layer is more than vertical fluid layer. Cavities which we used nanofluid with vertical fluid layer had higher heat transfer than cavities with horizontal fluid layer in natural convection. Also our obtaining numerical results is shown that in vertical cavities for both nanofluid and pure water, heat transfer coefficient and mean Nusselt number were higher in comparison with horizontal cavities.

However, it has been clearly shown by the available results that the heat transfer behavior of nanofluids is very complex and many other important factors influence on the heat transfer performance of the nanofluids in natural convective heat transfer, which should be identified further in future works. So further theoretical and experimental research investigations are needed to understand the heat transfer characteristics of nanofluids and identify new and unique applications for these fields.

Nomenclature

g acceleration of gravity (m/s^2)
Gr Grashof number = gbDTL^3/m^2
h coefficient of heat transfer (W/m^2 K)
k thermal conductivity of water (W/m_C)
L characteristic length (m)
Nu Nusselt number = hL/k
Pr Prandtl number = υ/α
Ra Rayleigh number = GrPr
T_w wall surface temperature (˚C)
T_{max} maximum temperature (˚C)
U, V Non-dimensional velocities

Greek symbols

α thermal diffusivity (m^2/s)
β thermal expansion coefficient of water
μ viscosity (N s/m^2)
ν kinematic viscosity (m^2/s)
ρ density (kg/m^3)
Ψ Stream function
θ Non-dimensional temperature

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