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NUMERICAL ANALYSIS OF THE NATURAL CONVECTIVE HEAT TRANSFER BETWEEN TWO INCLINED ISOTHERMAL WAVY SURFACES

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Abstract. *This article consists of a numerical analysis of the natural convective heat transfer from two inclined surfaces containing waves on both sides. The distance separation between the surfaces is relatively small in such a way that the flow from both surfaces influences the heat transfer rates from each surface. The waves considered in this work have a rectangular shape with constant height, having the effect of increasing the heat transfer rate of the heated plates to an adjacent fluid, in this case, ambient air. The main objective of this work is to determine the natural convective heat transfer rate from the inclined wavy surfaces to the ambient air. Effects on the natural convective heat transfer from the individual surfaces of each plate as well as from the total surface of each plate will be analyzed according to the following variables: inclination angle, dimensionless distance between the surfaces and the Rayleigh number. Results were obtained numerically using the standard* $k-\varepsilon$ *turbulence model including effects of buoyant forces with the aid of the commercial CFD solver ANSYS FLUENT©. Depending on the results obtained, geometric recommendations of the situation under analysis can be made, providing the greatest improvement in the natural convective heat transfer rate from the inclined wavy surfaces.*

Keywords: *natural convective heat transfer,* $k-\varepsilon$ *turbulence model, isothermal plates, inclined plates*

1. INTRODUCTION

The natural convective heat transfer is used in many practical situations to reduce the temperature and, because of this huge use, this area is getting considerable basic and applied interest. The present article will give a restricted view, related to external natural convective flows, that is, flow situations in which there are no constraining boundary surfaces significantly near to the surfaces being considered to have any considerable influence on the natural convective flow over these surfaces. The increase of the heat transfer rate in a given situation where the natural convective flows occurs is often difficult to accomplish. However, the use of wavy surfaces is one method of attempting to enhance natural convective heat transfer rates.

The enhancement of the heat transfer rate produced by using a wavy surface arises from the increase in the surface area exposed to the fluid to which the heat is being transferred and, in some cases, to the changes in the near surface flow produced by the presence of the surface waves. The total enhancement of the heat transfer rate will depend on the shape and relative size of the surface waves. Many wavy shapes have been considered in past studies but the most common shapes considered remain rectangular, triangular and sinusoidal waves. The enhancement of the heat transfer rate produced by using a wavy surface will also depend on the flow situation being considered, for example, flow over a plane surface or flow over a cylinder, and on the thermal boundary conditions at the surface. The two surface boundaries conditions most commonly considered being those in which there is a uniform temperature over the surface and those in which there is a uniform heat flux over the surface. Another factor that influences the natural convective heat transfer rate from a surface is its orientation, that is, is it horizontal or is it vertical, or is it inclined to the vertical and whether, when inclined, it is facing upward or downward (Oosthuizen, 2016).

While there have been some limited previous studies of natural convective heat transfer from thin, two-sided horizontal plates, these studies have mainly considered only the case where the plate is flat (non-wavy) and where the flow over the plate is laminar. Here, the conditions considered are such that laminar flow, transitional flow, and turbulent flow can occur over the plate. Numerical studies of heat transfer from a horizontal surface having rectangular waves and triangular waves for conditions under which laminar, transitional, and turbulent flow exist are described in (Oosthuizen, 2016a) and (Oosthuizen, 2016b). Other studies of natural convective heat transfer from horizontal wavy surfaces are described in (Prétot *et al*., 2000), (Prétot *et al*., 2003), (Siddiqa and Hossain, 2013) and (Siddiqa *et al*., 2015). In all of these studies, the natural convective heat transfer rate was obtained from a thin, one-sided, two-dimensional horizontal wavy plate having a uniform surface temperature. Studies of natural convective heat transfer from a thin, two-sided, two-dimensional one wavy plate having a uniform surface temperature are described in (Oliveira and Oosthuizen, 2018) and (Oliveira and Oosthuizen, 2019). In this way, there does not appear to be much available information on the natural convective heat transfer from a thin, two-sided, two-dimensional two wavy plates having a uniform surface temperature

The purpose of the present article is to undertake a numerical study of natural convective heat transfer from thin, two-sided, two-dimensional horizontal, two inclined plates having a uniform surface temperature. The surface shape is wavy, and attention has been given to the case where the surface waves have a rectangular shape with constant height. The importance of the present work is related to the increase of the natural convective heat transfer rate in situations where the implementation of a forced convection would be difficult to apply or even impossible. This may be the case, for example, when cooling a circuit board assembly or in cases where the thermal management of electronic components must be performed in more detail.

2. PHYSICAL SITUATION

The physical situation analyzed in this study is shown in Fig. 1:

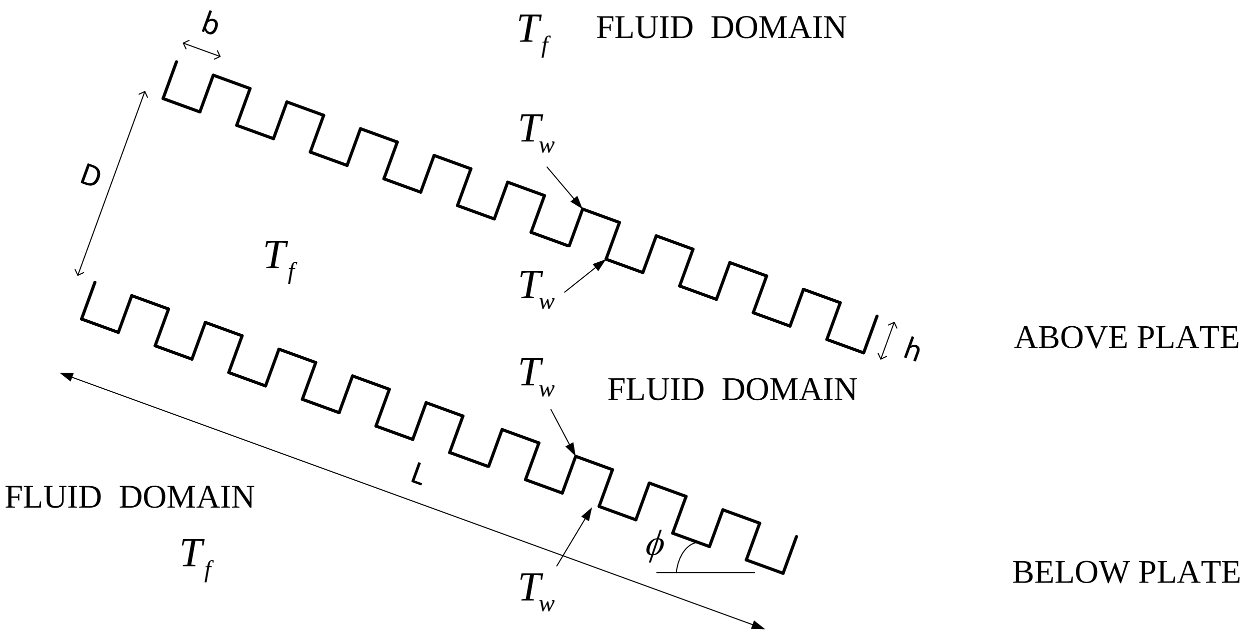

Figure 1. Two inclined wavy surfaces.

The situation under analysis consists of two thin, two-sided, two-dimensional inclined plates having a uniform surface temperature T_w. The surface shape of both plates is wavy and attention has been given to the case where the surface waves have a rectangular shape with constant height *h* and width *b*. The nineteen rectangular waves of both plates are equally spaced. The plates are in contact with a surrounding fluid at constant temperature T_f. For a heated surface, $T_w > T_f$ and both the top and bottom surfaces of both plates will exchange energy with the surrounding fluid by natural convection. Inclined surfaces have unit width *L* and unit depth *w*. The distance between plates is denoted by d. The purpose of this study is to calculate the heat transfer rate by natural convection between the heated surfaces and the surrounding fluid. The mean Nusselt numbers related to the natural convective heat transfer from the top and bottom surfaces of each plate can be calculated using the Newton law of cooling based on the distance between the surfaces and the definition of the mean Nusselt number based on the distance between the surfaces, that is:

$$\overline{\mathrm{Nu}}_{d\ \text{(above,top/bottom)}} = \frac{q_{\text{(above,top/bottom)}}d}{A(T_w - T_f)k} \quad (1)$$

$$\overline{\mathrm{Nu}}_{d\ \text{(below,top/bottom)}} = \frac{q_{\text{(below,top/bottom)}}d}{A(T_w - T_f)k} \quad (2)$$

where $\overline{\mathrm{Nu}}_d$ is the mean Nusselt number based on *d* and on the mean heat transfer rate, q is the mean heat transfer rate, *k* is the thermal conductivity of the fluid and *A* is the heat transfer area, calculated using *L*, *b*, *h* and the number of waves, according to Fig. 1.

3. SOLUTION PROCEDURE

In obtaining the numerical results discussed above, the mean flow has been assumed as steady. The Boussinesq approximation has been used, i.e., fluid properties have been assumed as constant (except for the density change with temperature in momentum equation). This gives rise to the buoyancy forces and the density change being assumed as proportional to the temperature change. Radiant heat transfer effects have been neglected. Allowance has been made for the possibility that turbulent flow can occur in the system. In order to deal with this the basic $k-\varepsilon$ turbulence model with standard wall functions and with full account being taken of buoyancy force effects has been used.

The mathematical model consists of an equation for the turbulent kinetic energy κ, Eq. (3), and a transport equation for the dissipation of turbulent kinetic energy ε, Eq. (4):

$$\frac{\partial(\rho\kappa)}{\partial t}+\mathrm{div}\,(\rho\kappa\mathrm{U})=\mathrm{div}\left[\frac{\mu_t}{\sigma_\kappa}\;\mathrm{grad}\;\kappa\right]+2\mu_t S_{ij}\cdot S_{ij}-\rho\varepsilon \tag{3}$$

$$\frac{\partial(\rho\varepsilon)}{\partial t}+\mathrm{div}\,(\rho\varepsilon\mathrm{U})=\mathrm{div}\left[\frac{\mu_t}{\sigma_\varepsilon}\;\mathrm{grad}\;\varepsilon\right]+C_{1\varepsilon}\frac{\varepsilon}{\kappa}2\mu_t S_{ij}\cdot S_{ij}-C_{2\varepsilon}\rho\frac{\varepsilon^2}{\kappa} \tag{4}$$

Equations (3) and (4) contains five adjustable constants, e.g., C_μ, σ_k, σ_ε, $C_{1\varepsilon}$ and $C_{2\varepsilon}$. The standard $k-\varepsilon$ turbulence model uses values for these constants obtained through comprehensive curve adjustments for a wide range of turbulent flows, i.e., $C_\mu=0.09$, $\sigma_k=1.00$, $\sigma_\varepsilon=1.30$, $C_{1\varepsilon}=1.44$ and $C_{2\varepsilon}=1.92$. U is the velocity vector, μ_t is the turbulent viscosity and S_{ij} is the deformation rate. The horizontal wavy surfaces has unit depth w and unit width *L* maintained at a uniform surface temperature $T_w=310$ K. The surrounding fluid is air at a temperature $T_f=290$ K at atmospheric pressure in all cases.

The governing equations subject to the boundary conditions have been solved numerically using the commercial CFD solver ANSYS FLUENT©. The numerical approach used here in order to determine when turbulence develops which involves solving the Reynolds averaged governing equations together with a turbulence model, in which the effects of buoyancy forces are taken into account, for all conditions considered and then monitoring the results obtained with increasing Rayleigh numbers to determine when significant turbulence effects develop. This approach has been used quite extensively in the study of forced convective flows, e.g., see (Schmidt and Patankar, 1991) and (Zheng *et al.*, 1998). The solutions presented in this work all basically have the following parameters:

1. The Rayleigh number, Ra_d, based on the reference length scale d of the heated surface and the difference between the temperature of the heated surface, T_w, and the temperature of the undisturbed fluid well away from the system, T_f, i.e.:

$$Ra_d=\frac{g\beta(T_w-T_f)d^3}{\nu\alpha} \tag{5}$$

2. The dimensionless width of the waves, $B=b/L$;
3. The dimensionless height of the waves, $H=h/L$;
4. The dimensionless distance between the plates, $D=d/L$;
5. The inclination angle of the plates, ϕ, and
6. The Prandtl number, Pr.

In Eq. (5), Ra_d is the Rayleigh number based on *d*, *g* is the gravitational acceleration, β is the bulk coefficient of thermal expansion, *d* is the distance between the plates, ν is the kinematic viscosity of the fluid and α is the thermal diffusivity of the fluid. Results have only been obtained for a Prandtl number of 0.71, i.e., effectively the value for air at 300 K. Before obtaining numerical results, a mesh independence study was carried out using the highest Rayleigh number value, i.e., 10^{12}, for a case where $B=H=0.0526232$, $D=0.3$ and $\phi=2^\circ$. All meshes were created with the aid of the GAMBIT© software. Results of the mesh independence test can be seen in Tab. 1:

Table 1. Mesh independence test results.

Number of elements	$\overline{\text{Nu}}_{d\ (\text{above,total})}$	$\overline{\text{Nu}}_{d\ (\text{below,total})}$
106.200	12.809	10.584
152.928	13.542	11.225
208.152	14.854	11.987
271.872	14.789	12.544
344.088	15.541	13.789
424.800	15.874	13.923
514.008	15.645	13.811

According to Tab. 1, for 424.800 elements, the mean Nusselt numbers remained approximately constant. This number of elements was then used in all numerical simulations. In all simulations, the mean Nusselt number integrated over the surface was monitored to ensure convergence and to verify that the simulation reached the steady state. The complete computational domain is 3.5m width and 4.0 high. The configuration of the ANSYS FLUENT© solver was based on the work of Oliveira and Oosthuizen (2018), Oliveira and Oosthuizen (2019), Oosthuizen (2016a) and Oosthuizen (2016b), having already been extensively tested and validated with results of numerical and experimental works by these authors. The convergence criteria used for all variables in numerical simulations was 10^{-5}. Figure 2 show an excerpt of the mesh used in numerical simulations:

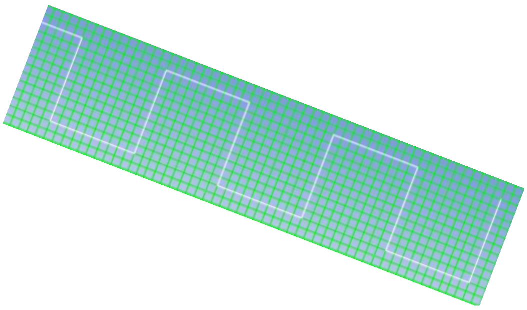

Figure 2. Excerpt of the mesh.

4. RESULTS AND DISCUSSION

Typical variations of the mean Nusselt number based on heat transfer rates averaged over the total surface area of the below plate and over the total surface area of the above plate, i.e., $\overline{\text{Nu}}_{d\ (\text{below,total})}$ and $\overline{\text{Nu}}_{d\ (\text{above,total})}$ with Rayleigh number for various values of the dimensionless distance between the plates, D, and for various plate inclination angle, ϕ, were obtained numerically. Values of the dimensionless distances used were 0.3, 0.5, 0.7 and 0.9. Besides, values of the plate inclination angle used were 2º, 4º, 6º, 8º and 10º. Both plates have unit depth w and unit width L maintained at an uniform surface temperature $T_w = 310\ \text{K}$. The surrounding fluid is air at a temperature $T_f = 290\ \text{K}$ at atmospheric pressure. Numerical simulations were performed for Rayleigh numbers varying between 10^6 to 10^{12}.

Figures 3 to 6 show the variation of the mean total Nusselt number for the above/below wavy plates with the Rayleigh number for $B = H = 0.052632$ and $\phi = 2^\text{o}$ in the case where the dimensionless distances between the plates are equal to 0.3, 0.5, 0.7 and 0.9.

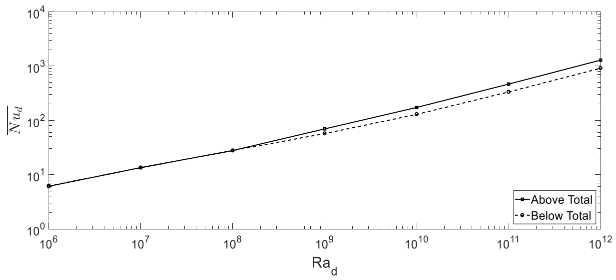

Figure 3. Variation of the mean total Nusselt number for the above/below wavy plates with the Rayleigh number for $B = H = 0.052632$, $D = 0.3$ and $\phi = 2^\text{o}$.

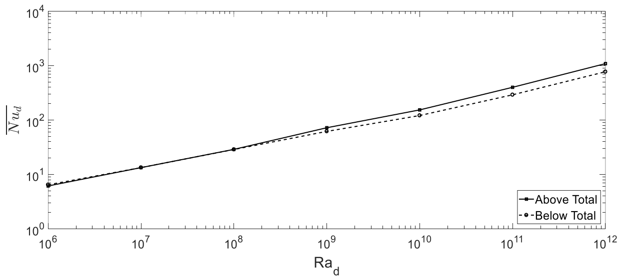

Figure 4. Variation of the mean total Nusselt number for the above/below wavy plates with the Rayleigh number for $B = H = 0.052632$, $D = 0.5$ and $\phi = 2^\text{o}$.

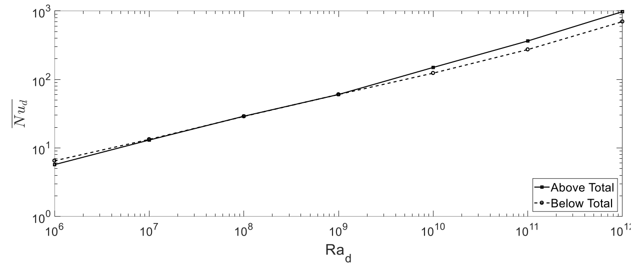

Figure 5. Variation of the mean total Nusselt number for the above/below wavy plates with the Rayleigh number for $B = H = 0.052632$, $D = 0.7$ and $\phi = 2^{o}$.

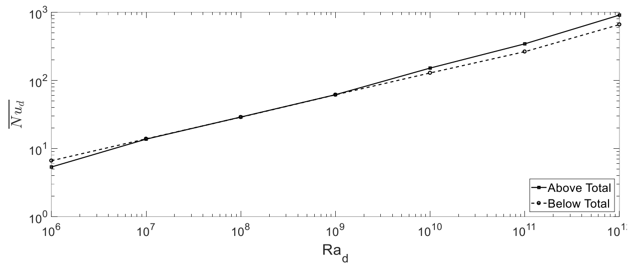

Figure 6. Variation of the mean total Nusselt number for the above/below wavy plates with the Rayleigh number for $B = H = 0.052632$, $D = 0.9$ and $\phi = 2^{o}$.

From the results shown in Figs 3 to 6, it can be seen that the mean above total Nusselt number and the mean below total Nusselt number increase with the increase of the Rayleigh number. Besides, for both plates, the mean above total Nusselt number and the mean below total Nusselt number are approximately the same for Rayleigh number equal to 10^6, 10^7 and 10^8 for $D = 0.3$ and $D = 0.5$, with the maximum percentage variation equal to 4.8%. However, for $D = 0.7$ and $D = 0.9$, it can be seen that the mean below total Nusselt number stay higher than the mean above total Nusselt number for Rayleigh number equal to 10^6 (12.1% and 19.1% respectively), before the congruence for Rayleigh numbers equals to 10^7 and 10^8 (with less than 1% of variation). Besides, it can be seen that the mean total Nusselt number is higher on the above plate than on the below plate for Rayleigh number varying from 10^8 to 10^{12} and for $D = 0.3$ and $D = 0.5$. Furthermore, this last characteristic can be seen for $D = 0.7$ e $D = 0.9$, but the difference between the mean total Nusselt number are mainly noted for Rayleigh number varying between 10^{10} to 10^{12}.

Figures 7 to 10 show the variation of the mean total Nusselt number for the above/below wavy plates with the dimensionless distance between the plates for $B = H = 0.052632$ and $\phi = 2^{o}$ in the case where the Rayleigh number are equal to 10^6, 10^8, 10^{10} and 10^{12}:

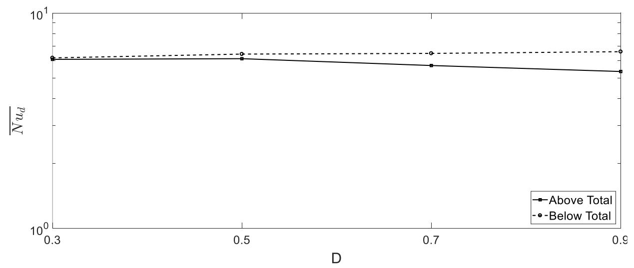

Figure 7. Variation of the mean total Nusselt number for the above/below wavy plates with the dimensionless distance between the plates for $B = H = 0.052632$, $Ra = 10^6$ and $\phi = 2^{o}$.

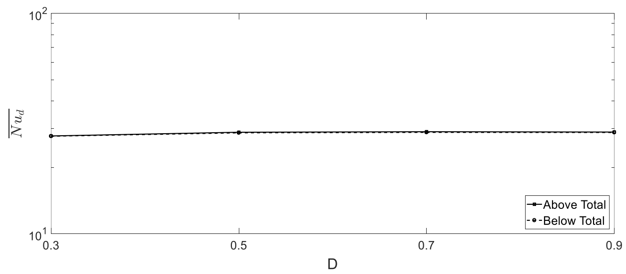

Figure 8. Variation of the mean total Nusselt number for the above/below wavy plates with the dimensionless distance between the plates for $B = H = 0.052632$, $Ra = 10^8$ and $\phi = 2^{o}$.

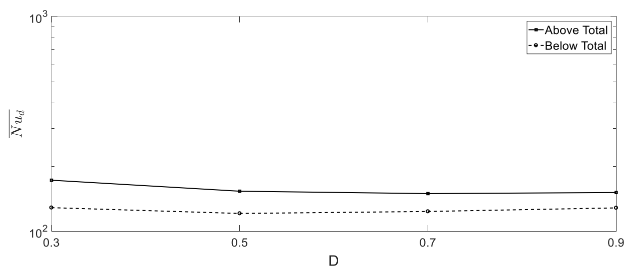

Figure 9. Variation of the mean total Nusselt number for the above/below wavy plates with the dimensionless distance between the plates for $B = H = 0.052632$, $Ra = 10^{10}$ and $\phi = 2^{o}$.

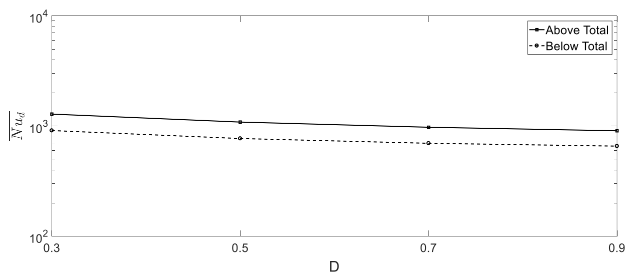

Figure 10. Variation of the mean total Nusselt number for the above/below wavy plates with the dimensionless distance between the plates for $B = H = 0.052632$, $Ra = 10^{12}$ and $\phi = 2^{o}$.

Regarding Figs. 7 to 10, it is noted that an increase of the Rayleigh increase the mean total Nusselt numbers for the top and the bottom plates. In Fig. 7, the mean below total Nusselt number stays almost constant for all *D* considered (the maximum percentage variation is 4.1%) and always higher than the mean above total Nusselt number, who demonstrated a decreasing behavior although the increase of *D*, with maximum percentage variation equal of 7.0%. From Figs. 8 to 10, it can be seen that the mean total Nusselt for the above plate is higher than the mean total Nusselt number for the below plate. Figure 8 shows that the values of the mean Nusselt number (mean total above and mean total below) stay almost congruent for all *D* considered. The maximum percentage variation stays at 4%, increasing the mean total Nusselt number for $D=0.5$ and $D=0.7$ and showing a small decrease for $D=0.9$. From Fig. 9, it can be seen that both values of mean total Nusselt number (below and above) decrease from $D=0.3$ to $D=0.5$ and afterwards the numbers stay almost congruent from $D=0.5$ to $D=0.9$. with the maximum percentage variation at 10.3% and 9.5% respectively for above and below plates. From Fig. 10, it can be seen decreasing values of the mean total Nusselt numbers with the increase of distance for both plates, with maximum percentage variations at 29.5% and 27.7%.

Figures 11 to 14 show the variation of the mean total Nusselt number for the above/below wavy plates with the Rayleigh number for $B=H=0.052632$ and $D=0.3$ in the cases where the plate inclination angle are equal to 2º, 4º, 6º, 8º and 10º:

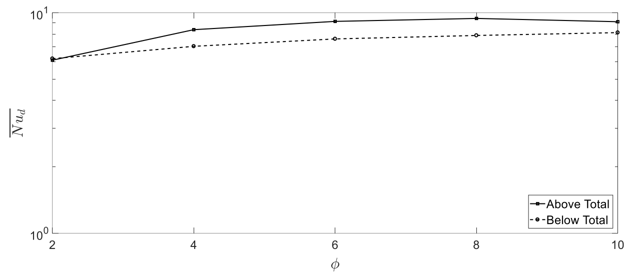

Figure 11. Variation of the mean total Nusselt number for the above/below wavy plates with the plate inclination angle for $B=H=0.052632$, $Ra=10^6$ and $D=0.3$.

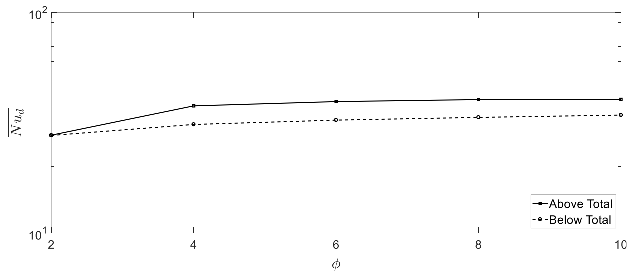

Figure 12. Variation of the mean total Nusselt number for the above/below wavy plates with the plate inclination angle for $B=H=0.052632$, $Ra=10^8$ and $D=0.3$.

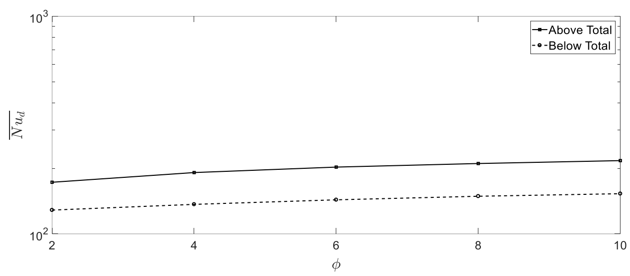

Figure 13. Variation of the mean total Nusselt number for the above/below wavy plates with the plate inclination angle for $B=H=0.052632$, $Ra=10^{10}$ and $D=0.3$.

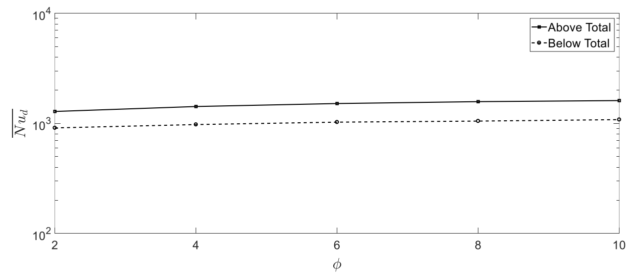

Figure 14. Variation of the mean total Nusselt number for the above/below wavy plates with the plate inclination angle for $B=H=0.052632$, $Ra=10^{12}$ and $D=0.3$.

Regarding Figs. 11 to 14, it is noted that an increase of the Rayleigh number causes an increase of the mean total Nusselt numbers. From figures 11 and 12, it can be seen higher values of mean total Nusselt number at above plate in comparison to the below plate for inclination angles 4º, 6º, 8º and 10º, with maximum percentage variation at 9%. However, at inclination angle of 2º, the mean total Nusselt numbers above and below show almost the same values, with percentage variation at 1.6%. From figures 13 and 14, it is shown higher values of the mean total Nusselt numbers for above plate in comparison to the below plate. Furthermore, both values of mean total Nusselt numbers increase with higher inclination angles, with maximum percentage variation at 25.6% and 18.9% from 2º to 10º, for above and below plates, respectively.

Figures 15 to 19 show the variation of the mean total Nusselt number for the above/below wavy plates with the Rayleigh number for $B=H=0.052632$ and $D=0.3$ in the cases where the plate inclination angle are equal to 2º, 4º, 6º, 8º and 10º:

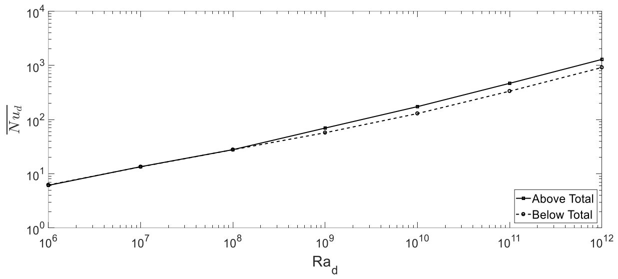

Figure 15. Variation of the mean total Nusselt number for the above/below wavy plates with the Rayleigh number for $B = H = 0.052632$, $D = 0.3$ and $\phi = 2^{\circ}$.

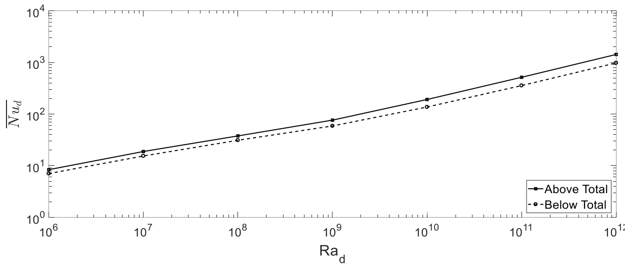

Figure 16. Variation of the mean total Nusselt number for the above/below wavy plates with the Rayleigh number for $B = H = 0.052632$, $D = 0.3$ and $\phi = 4^{\circ}$.

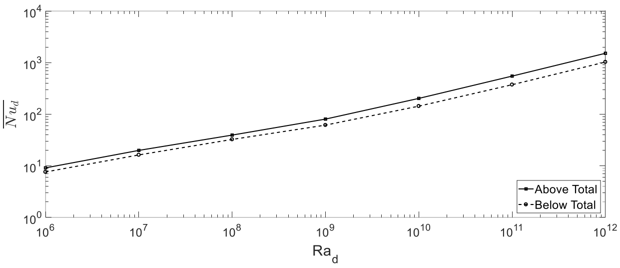

Figure 17. Variation of the mean total Nusselt number for the above/below wavy plates with the Rayleigh number for $B = H = 0.052632$, $D = 0.3$ and $\phi = 6^{\circ}$.

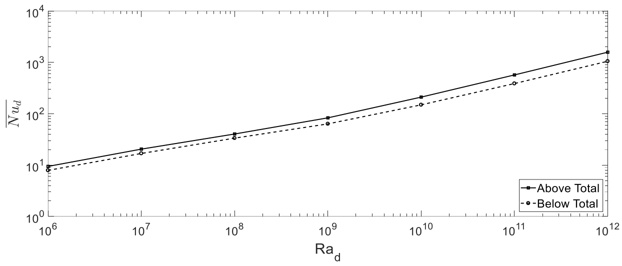

Figure 18. Variation of the mean total Nusselt number for the above/below wavy plates with the Rayleigh number for $B = H = 0.052632$, $D = 0.3$ and $\phi = 8^{\circ}$.

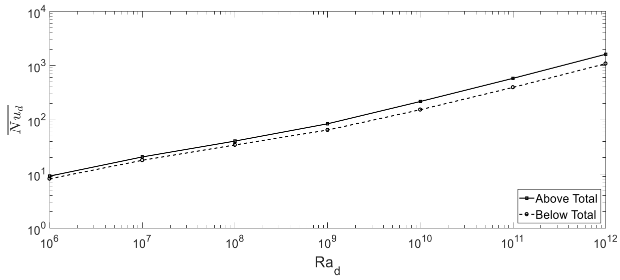

Figure 19. Variation of the mean total Nusselt number for the above/below wavy plates with the Rayleigh number for $B = H = 0.052632$, $D = 0.3$ and $\phi = 10^{\circ}$.

From Figs. 15 to 19, it can be seen that higher values of Rayleigh number cause an increase of the mean total Nusselt numbers. From Fig. 15, it is shown that the mean total above Nusselt number and the mean total below Nusselt number stay almost the same for Rayleigh number varying from 10^6 to 10^8, with maximum percentage variation at 1.6%. Afterwards, it can be seen higher mean total Nusselt numbers at above plate than the below plate, with maximum percentage variation varying from 18.0% to 29.1%. From Figs. 16 to 19, it can be seen that the mean total above Nusselt number stay higher than the mean total below Nusselt number. However, it was noticed that this difference increase with higher Rayleigh numbers, with maximum percentage variation at 33.4% for 8° at Rayleigh number 10^{12}.

5. CONCLUSIONS

The study of the natural convective heat transfer from two inclined surfaces containing waves on both sides has been undertaken. Some conclusions can be obtained from the numerical results present in this work:

- It was shown that the variation of D for $\phi = 2^{\circ}$ does not provide much difference of the mean total Nusselt number (above and below) for each Rayleigh number considered and for each D considered. However, it was noticed that higher Rayleigh numbers give higher mean total Nusselt numbers for the above plate in comparison to the below plate.

- It was shown that the mean total Nusselt (above and below) do not vary significantly for Rayleigh number equal of 10^6 and 10^8. However, for Rayleigh number equal of 10^{10} and 10^{12} it was shown a maximum percentage variation of 29.5% and 27.7%, respectively, for above and below plates. It was noticed higher mean total Nusselt numbers for $D = 0.3$.
- It was shown higher mean total Nusselt number for $\phi = 10^{o}$ in comparison with other plate inclination angles considered, except for Rayleigh number equal of 10^6.
- It was shown that for $D = 0.3$ and for the range of Rayleigh number considered a higher difference between the mean above/below total Nusselt number. It was noticed higher mean total Nusselt number for the above plate in comparison with the below plate.

6. REFERENCES

Oliveira, S.D.R. and Oosthuizen, P.H., 2018. “A numerical study of the simultaneous natural convective heat transfer from the top and bottom surfaces of a thin horizontal plane having a wavy surface”. In *26th Annual Conference of the Computational Fluid Dynamics Society of Canada* – CFD 2018, Winnipeg, Canada.

Oliveira, S.D.R. and Oosthuizen, P.H., 2019. “A numerical study of the simultaneous natural convective heat transfer from the top and bottom surfaces of a thin inclined plate having a wavy surface with constant height”. In *4th Thermal and Fluids Engineering Conference* – TFEC 2019, Las Vegas, USA.

Oosthuizen, P.H., 2016. “External natural convective heat transfer from bodies having a wavy surface for conditions under which laminar, transitional, and turbulent flow can exist”. *Advances in Heat Transfer,* Vol. 48, p. 261-317.

Oosthuizen, P.H., 2016a. “A numerical study of natural convective heat transfer from a horizontal isothermal surface with rectangular surface roughness elements”. In *Proceedings of the First Pacific Rim Thermal Engineering Conference* - PRTEC 2016, Hawaii’s Big Island, USA.

Oosthuizen, P.H., 2016b. “A numerical study of the effect of triangular waves on natural convective heat transfer from an upward facing heated horizontal isothermal surface”. In *ASME 2016 International Mechanical Engineering Congress and Exposition*, Phoenix, USA.

Prétot, S., Zeghmati, B. and Caminat, Ph., 2000. “Influence of surface roughness on natural convection above a horizontal plate”. *Advances in Engineering Software*, Vol. 30, p. 793-801.

Prétot, S., Miriel, J., Baily, Y. and Zeghmati, B., 2003. “Visualization and simulation of the natural convection flow above horizontal wavy plates”. *Numerical Heat Transfer, Part A: Applications*, Vol. 43, p. 307-325.

Siddiqa, S. and Hossain, M.A., 2013. “Natural convection flow over wavy horizontal surface”. *Advances in Mechanical Engineering*, Vol. 5, p. 1-7.

Schmidt, R. C. and Patankar, S. V., 1991. “Simulating boundary layer transition with low-Reynolds-Number *k*-*ε* turbulence models: Part 1- an evaluation of prediction characteristics”. *ASME Journal of Turbomachinery*, Vol. 113, p. 10-17.

Siddiqa, S., Hossain, M.A. and Gorla, R.S.R., 2015. “Natural convection flow of viscous fluid over triangular wavy horizontal surface”. *Computers & Fluids*, Vol. 5, p. 130-134.

Zheng, X., Liu, C., Liu, F. and C.-I. Yang., 1998. “Turbulent transition simulation using the *k*-*ω* model”. *International Journal for Numerical Methods in Engineering*, Vol. 42, p. 907-926.

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