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Fluid Flow and Convective Heat Transfer in a Water Chemical Condenser

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Abstract: In this paper, a detailed investigation of water ($\mathrm{Pr} = 7.0$) convection in a chemical condenser is carried out. Two openings are located along one side of the cavity. The Navier-Stokes equations are solved in the frame of a control volume method using the SIMPLEC algorithm to implement adequate coupling of pressure and velocity. Special emphasis is given to the influence of the Reynolds number, the tilt of the channel and the Rayleigh number on the convective heat transfer. Results are presented and discussed allowing the control parameters to span relatively wide intervals: Rayleigh number ($10^4 \leq Ra \leq 5 \times 10^5$), channel inclination ($0^\circ \leq \varphi \leq 90^\circ$) and Reynolds number ($10 \leq \mathrm{Re} \leq 1000$). On the basis of these results, a new correlation of the Nusselt number is elaborated.

Keywords: Chemical condenser; tilted channel; convection; numerical study; heat transfer

Nomenclature

H	Channel width, m
L	Channel height, m
W	Obstacle width, m
h	Obstacle height, m
Re	Reynolds number, $\mathrm{Re} = H.U_\circ/v$
Ra	Rayleigh number, $Ra = g.\beta.\Delta T.H^3/\alpha.v$
Pr	Prandtl number, $\mathrm{Pr} = v/\alpha$
Ri	Richardson number $Ri = \dfrac{Ra}{\mathrm{Pr}.\mathrm{Re}^2}$
Gr	Grashof number, $Gr = \dfrac{Ra}{\mathrm{Pr}}$
P_0	Pressure of fluid, Pa
P	Dimensionless pressure
T	Temperature of fluid, K

T_H	Temperature on the source heat, K
T_c	Temperature of the cold surface, K
θ	Dimensionless temperature of fluid, $\theta = (T - T_C)/(T_H - T_C)$
u, v	Velocities in x and y directions, *m/s*
U, V	Dimensionless velocities in x and y directions, $U, V = (u, v)/U_0$
U_0	Average jet velocity at the entrance, m/s
x, y	Cartesian coordinates dimensionless
X, Y	Cartesian coordinates,$X, Y = (x, y)/H$
g	Gravitational acceleration, m/s^2
Nu	Average Nusselt number, (Eq. (5))

Greek symbols

α	Thermal diffusivity $m^2.s^{-1}$
β	Volumetric coefficient of thermal expansion K^{-1}
λ	Thermal conductivity of fluid, $W.m^{-1}.K$
ν	Kinematic viscosity of fluid $m^2.s^{-1}$
ρ	Fluid density $kg.m^{-3}$
φ	Inclination angle of the channel
Ψ	Dimensionless stream function
t	Time, *s*
τ	Dimensionless time

Subscripts

C	Cold
H	Hot
max	Maximum
min	Minimum
f	Fluid

1 Introduction

Mixed convection usually induced in cavities or channels containing heating elements on one of its walls or both walls has been intensively studied by dint of its theoretical and practical importance. Several studies of heat transfer and fluid flow have been conducted on mixed convection in cavities in order to give a comprehensive visualization of the flow and the temperature distribution within the studied configurations. This study will find applications in many industrial fields such as transport devices, solar collectors, heat regenerators, electronic cooling devices and many others (Icoz et al. [1]).

Akiyama et al. [2] have conducted an experiment to determine the onset of longitudinal columnar vortices due to buoyant forces for fully developed laminar forced convection between two infinite horizontal plates, each wall subjected to the identical uniform axial temperature gradient. They found out that the limiting case with vanishing axial temperature gradient and heating from below is known to have a critical Rayleigh number of 1708.

The influence of the buoyancy force on laminar forced convection in the entrance region between horizontal parallel plates has been studied by Naito et al. [3] when either one or both of the walls of the duct are maintained at equal constant temperatures or when one wall is held at a constant temperature, and the opposite wall is insulated. Among these conclusions, the values of the local Nusselt number are consistently smaller than those of a horizontal channel as inclination angles increase.

The natural convection in the air has been studied by Oronzio Manca et al. [4] In this study, the relations between process parameters and the composite correlations of average Nusselt number in terms of channel Rayleigh number were proposed for inclined channels with uniform symmetric or asymmetric heat flux.

Later, the optimal distribution and sizes of discrete heat sources in a vertical open channel cooled by natural convection have been examined by Da Silva et al. [5]. In this work, two classes of geometries are considered: (i) heat sources with fixed size and fixed heat flux, and (ii) single heat source with variable size and fixed total heat current. The authors found that, for configuration (i), the optimal location changes as the Rayleigh number increases, and the last heat source tends to migrate toward the exit plane, which results in a non-uniform distribution of heat sources on the wall. For configuration (ii), it has been shown that as the flow intensity increases, the optimal heat source size approaches the height of the wall.

In the same subject, Puangsombut et al. [6] have developed an empirical correlation for heat transfer in an inclined open-ended rectangular channel heated from the top and equipped with a radiant barrier (RB) on the lower plate. It was concluded that the use of the reflective foil known as RB increased the induced heat convection and the airflow rate by about. The corresponding decrease of heat gain through the lower plate of the channel is high ranging from about.

After, a numerical investigation of mixed convection in air subject to an interaction between a buoyancy flow and the flow induced by a moving plate in a vertical channel study by Assunta el al. [7] The effects of the channel aspect ratio, Rayleigh and Reynolds numbers were investigated. It turns out that the larger the channel aspect ratio, the stronger the effects of the moving plate. Increasing Reynolds number significantly decreases the dimensionless temperature of the channel walls. The mixed convection is also studied in an inclined rectangular channel with three discrete heat sources placed on the bottom surface by Guimarães et al. [8]. The Reynolds and Grashof numbers and the channel inclination are respectively: $1 \leq \mathrm{Re} \leq 1000, 10^3 \leq Gr \leq 10^5$, and $0° \leq \gamma \leq 90°$.The inclination has a stronger influence on the flow and heat transfer for low Reynolds numbers. In general, cases that show the lowest temperature distributions on the modules are consistent with the inclination angles of 45°and 90°.

In this paper, we focus on the chemical condensers, device which allows a vapor to pass through a confined space and be cooled by a surrounding cooler fluid (water) in order to change its phase. A water condenser (Liebig condenser) has the tube enclosed in a larger tube, so that cooling water is able to pass in the space between the two tubes. The efficiency is improved by slanting the apparatus, so that the vapor to be condensed flows downwards through the inner tube whilst the cooling water flows upward through the outer tube, thus creating a counter current flow. We will be interested to the cooling water flowing between the two tubes to find the best tilt and the minimum flow for the better cooling of chemical condenser.

The main objective of this study is to analyze the effect of cooling water flow, Reynolds number, and the inclination of the chemical condenser on the heat transfer between the cooling water the hot gas, Rayleigh number.

2 Physical Problem and Governing Equations

The geometry of the problem herein investigated is depicted in Fig. 1. The system is made of a chemical condenser, three configurations are studied in this work. In the first case, horizontal chemical condenser (case 1), the second configuration, inclined chemical condenser (case 2), for the third configuration, vertical chemical condenser (case 3). The flow is considered laminar, incompressible and the Boussinesq approximation has been applied. The dimensionless governing equations can be written as:

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \tag{1}$$

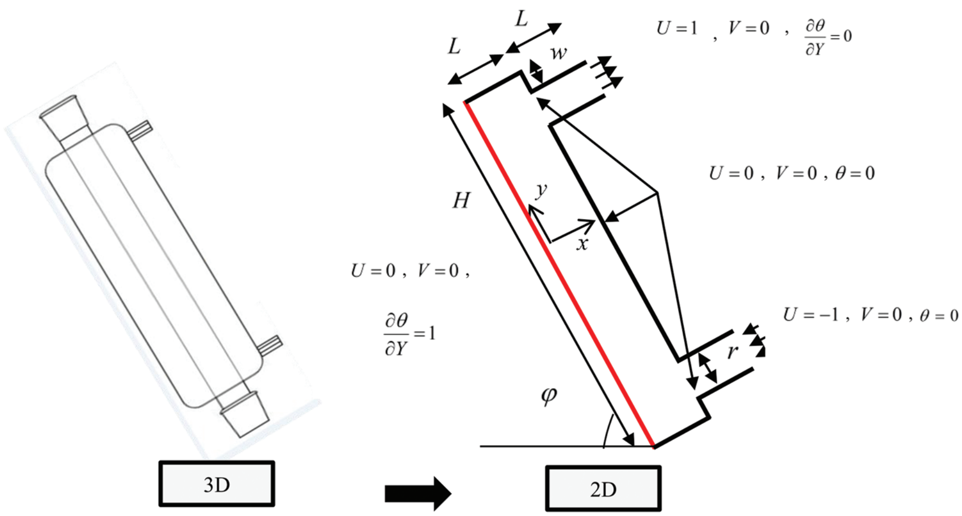


Figure 1: Studied configuration

$$\frac{\partial U}{\partial \tau}+U\frac{\partial U}{\partial X}+V\frac{\partial U}{\partial Y}=-\frac{\partial P}{\partial X}+\frac{1}{\mathrm{Re}}\left(\frac{\partial^2 U}{\partial X^2}+\frac{\partial^2 U}{\partial Y^2}\right)-\sin(\varphi)\frac{Ra}{\mathrm{Pr}\,.\mathrm{Re}^2}\theta \tag{2}$$

$$\frac{\partial V}{\partial \tau}+U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\frac{1}{\mathrm{Re}}\left(\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right)+\cos(\varphi)\frac{Ra}{\mathrm{Pr}\,.\mathrm{Re}^2}\theta \tag{3}$$

$$\frac{\partial \theta}{\partial \tau}+U\frac{\partial \theta}{\partial X}+V\frac{\partial \theta}{\partial Y}=\frac{1}{\mathrm{Pr}\,.\mathrm{Re}}\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right) \tag{4}$$

Referring to Fig. 1, the dimensionless variables are:
$X=\frac{x}{H}$, $Y=\frac{y}{H}$, $U=\frac{u}{U_0}$, $V=\frac{v}{U_0}$, $\theta=\frac{T-T_c}{T_H-T_c}$, $P=\frac{p}{\rho\,U_0^2}$, $\tau=\frac{t.\,U_0}{H}$ $Ra=\frac{g\beta\ \Delta\mathrm{T}\ H^3}{\alpha v}$ with $\Delta T=(T_H-T_c)$, $\mathrm{Re}=H.U_\circ/v$, $\mathrm{Pr}=\frac{v}{\alpha}$, $Gr=\frac{Ra}{\mathrm{Pr}}$, $Ri=\frac{Gr}{\mathrm{Re}^2}$

The imposed boundary conditions, in terms of temperature and velocity, are similar to those of the mixed convection flow in a vertical channel [Kriraa [9], El Alami [10], Sabour [11]]:

Initially, at $\tau=0$, $U=V=\theta=0$. The dimensionless boundary conditions for our study are presented at $\tau\geq 0$.

At $X=1$ and $0.3\leq Y\leq 0.8$; $U=-1$, $V=0$, $\theta=0$ (inlet)

At $X=1$ and $4.2\leq Y\leq 4.7$;$U=1$, $V=0$,$\frac{\partial\theta}{\partial Y}=0$ (outlet)

At $X=0.5$ and $0\leq Y\leq 0.3$ Or $0.8\leq Y\leq 4.2$ Or $4.7\leq Y\leq 5$; $U=0$, $V=0$, $\theta=0$

At $0\leq X\leq 0.5$ and $Y=0$ Or $Y=L$; $U=0$, $V=0$, $\theta=0$

At $X=0$ and $0\leq Y\leq L$; $U=0$, $V=0$, $\frac{\partial\theta}{\partial Y}=1$

The mean Nusselt number over one active wall of the channel is:

$$Nu=\frac{1}{H}\int_{channel\ wall}\frac{1}{\theta}dx \tag{5}$$

3 Numerical Method

The governing equations of the problem were solved, numerically, using the control volume method of Patankar [12]. The QUICK scheme developed by Leonard [13] was used for convective terms discretization. The final discretized forms of the Eqs. (1)–(4) were solved using the SIMPLEC algorithm of Van Doormaal et al. [14]. As a result of a grid independence study, the grid size of 240×41 was found to model accurately the flow fields described in the corresponding results. Time steps considered ranging between 10^{-5} and 10^{-4}. The accuracy of the numerical model was verified, by confronting our finding with that reported by De Vahl Devis [15] and Le Queré [16], for natural convection in the differential heated cavity (Tab. 1). Additional validation of the numerical code was performed by comparing with Hamouche et al. [17] results in a mixed convection air cooling of protruding heat sources mounted in a horizontal channel (Tab. 2 and Fig. 2). Also, we have confronted our result to those proposed by Desrayaud et al. [18] in a vertical channel with two ribs symmetrically placed on the channel walls, (Tab. 3) and we have found a good agreement in flow terms (M). We notice that a comparison of Nusselt number in a convergent channel was made by us in 2014 (kriraa et al. [9]).

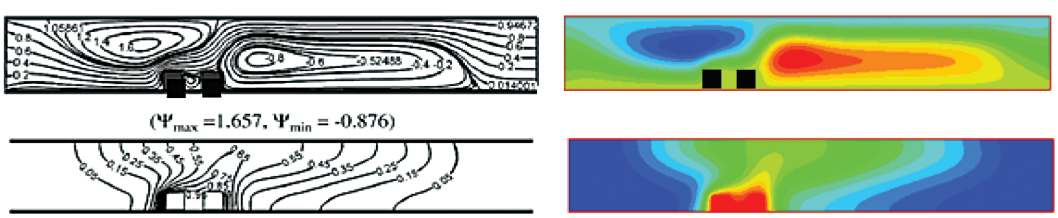
We can say here that the comparisons in all the above cases are found to be in excellent agreement. This favorable comparison lends confidence in the numerical results to be reported in the next section.

Table 1: Comparison of streamlines Ψ_{max} of [15] and [16] with our results in rectangular cavity

Ra	De Val Davis [15]	Le Queré et al. [16]	Current study	Maximum relative Error (%)
10^4	5.098	————	5.035	1.2 %
10^5	9.667	————	9.725	0.6 %
10^6	17.113	16.811	17.152	2 %
10^7	————	30.170	30.077	0.3 %

Table 2: Comparison of streamlines $\Psi_{max}$of Hamouche et al. [17] with our results

Re	Current results	Hamouche et al. [17]	Maximum relative Error (%)
5	1.675	1.675	0.00 %
10	1.056	1.028	2.72 %
30	0.999	1.000	0.10 %



(a) (b)

Figure 2: Comparison of flow structure and isotherms (a) Hamouche's results [17] with (b) Our results

Table 3: Comparison of our results and those of Derayaud et al. [18]

Ra = 10^5(A = 5)	Derayaud et al. [18]	Present study	Maximum deviation
Ψ_{max}	151.51	152.85	0.9%
M	148.27	151.72	2.2%

4 Results and Discussion

A detailed numerical study has been carried out on convection heat transfer enhancement. The study deals with the effects of the Reynolds number Re, number of Rayleigh Ra and inclination φ. In this study, dimensions are taken to be: $H = 5$; $L = 0.5$; $r = 0.5$, $w = 0.5$, Prandtl number $\mathrm{Pr} = 7.0$, Reynolds number $10 \le \mathrm{Re} \le 1000$, Rayleigh number is kept in the order of ($10^4 \le Ra \le 5 \times 10^5$) and channel inclination angle $0° \le \phi \le 90°$.

4.1 Effect of the Reynolds Number

We present the streamlines and isotherms in Fig. 3. At $Ra = 10^5$ (fixe value), for three values of Reynolds numbers ($\mathrm{Re} = 10,\ 500,\ 1000$) and different values of inclination ($\varphi = 0°,\ 45°,\ 90°$).

For $\varphi = 0°$, $\mathrm{Re} = 10$ (Fig. 3a) low Reynolds, the isotherms, in the up, shows that the buoyancy effects are prominent, especially at the half-parity of the exit.

The buoyancy effects decrease slightly with Reynolds, unto the critical Reynolds number, following the buoyancy effects are negligible; this is due essentially to the increase of Richardson number, Such that: $Ri \succ 1$ or $\mathrm{Re} = \sqrt{\frac{Ra}{\mathrm{Pr}}} \prec 120$ corresponds to the free convection, $Ri = 1$ or $\mathrm{Re} = 120$ corresponds to the mixed convection and $Ri \prec 1$ or $\mathrm{Re} \succ 120$ corresponds to the forced convection, so for $\mathrm{Re} = 10$ (Fig. 3a) the free convection is prominent, contrariwise for $\mathrm{Re} = 500$ (Fig. 3b) and for $\mathrm{Re} = 1000$ (Fig. 3c) the forced convection is prominent.

The streamlines corresponding, in the bottom, shows the same thing, when the Reynolds number is small the isotherms stretch out and occupy a large place in the channel. However, when the Reynolds number is higher, the isotherms become approximately horizontal with a notable decrease in the temperature distribution into the channel. Consequently, the increase of the Reynolds number reduces considerably the buoyancy effects, and provides a good evacuation of the heated water outside the channel.

For study the effect of the inclination, we inclined the channel for $\varphi = 45°$, (Fig. 3d–3f), the isotherms, in the right, and the streamlines, in the left, are similar to the previous case, so the effect of inclination is limited for all numbers of Reynolds.

In (Fig. 3j–3i), we present the solutions to the problem when we increased the tilt angle until $\varphi = 90°$. We obtained a widely different flow structure of the two previous cases. The isotherms in the right and the streamlines in the left show that, the appearance of a recirculation cell in the channel for the whole game of Reynolds (Fig. 3j) (Fig. 3h) and (Fig. 3i). The loss of the recirculation cell that has given way to a thermally stratified zone and extended around the channel axis (Fig. 3h) and (Fig. 3i). This thermal stratification is due to the development of flow kind boundary layers separated. The isotherms testify, they Also, this type of flow.

So for high numbers of Reynolds, the solution is independent of inclination.

To examine the effect of the Reynolds number on the heat removal rate, we display the variations of the average Nusselt number at and for different values of Reynolds.

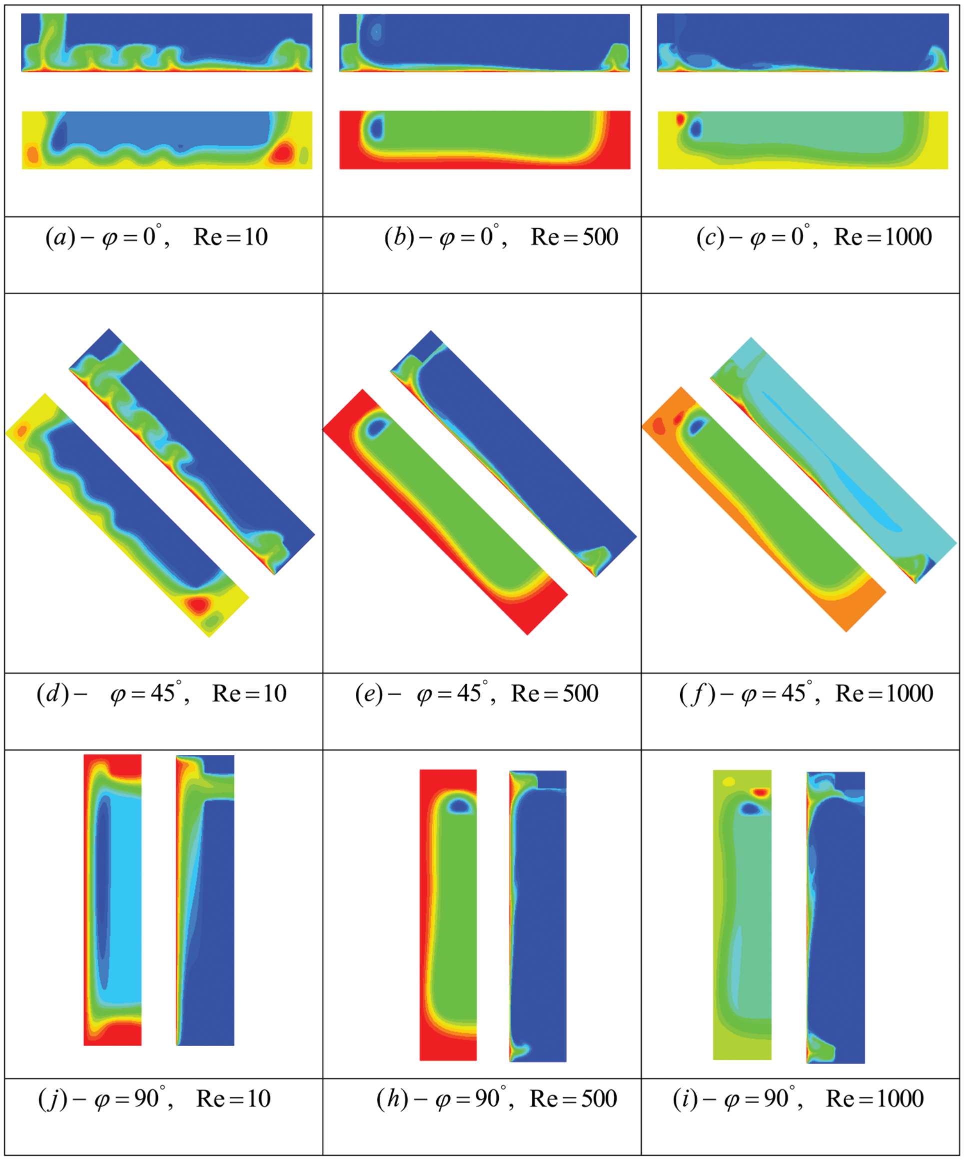


Figure 3: Flow structure and isotherms for $\varphi = 0°$ (a,b,c), $\varphi = 45°$ (d,e,f) , $\varphi = 90°$ (j,h,i) with Re = 10 (a,d,j), Re = 500 (b,e,h) and Re = 1000 (c,f,i)

In this geometry. We performed a heat transfer study in terms of Nusselt number in the channel with Reynolds number for different values of the inclination of the channel, and we study the variation of the Nusselt number with the inclination of the channel for different values of Reynolds.

In the Fig. 4, we display the variations of Nusselt number with Reynolds number for several inclinations of $Ra = 10^5$. The number of Nusselt increases according to the number of Reynolds, which has the effect of increasing the heat exchange on the active walls of channel (inside the condenser) for all inclinations. The important remark we see is that in the low Reynolds. In the small inclination $\varphi = 0°$, the Nusselt number for Re = 10 is greater than the Nusselt number for Re = 20. The inclination of the channel does not have a large effect on the Nusselt number, especially, for the large numbers of Reynolds.

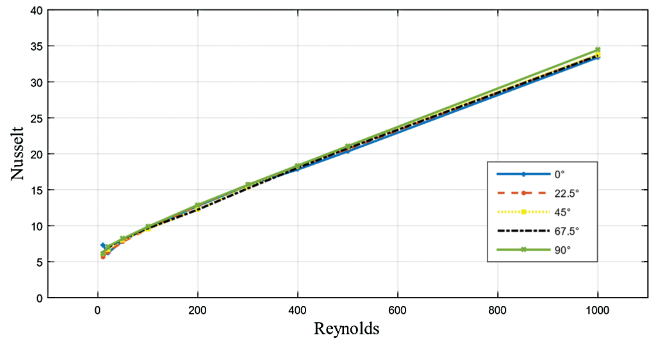


Figure 4: Nusselt variation with Reynolds, for different values of inclination φ

For more detail, we presented the Nusselt number with inclination for several Reynolds number. Note that, the inclination has no effect on the Nusselt number for the entire range of Reynolds number chosen (Fig. 5). The exception for low Reynolds Re $= 10$, the Nusselt number in inclination $\varphi = 0°$ is greater to the Nusselt number in inclination $\varphi = 20°$;this can be due to the appearance of circulating cells in the vicinity of the active wall, in the case of low values of Reynolds number and inclination. Although, the Nusselt number decreases with inclination because there is a disappear of the circulating cells. Finally, the bad cooling beyond $\varphi = 20°$, shows that the Nusselt number does not depend on inclination.

The effect of the tilt on the heat exchange through the channel is limited. Also, we notice that he changes of the Nusselt with Reynolds for different inclination values are identical and may be represented by the same curve. Therefore, we can find a linear equation to represent the 45 number of data points with coefficient of determination $r^2 = 0.996$. The correlation propose dis expressed as: $Nu = 0.028 \times \mathrm{Re} + 6.679$.

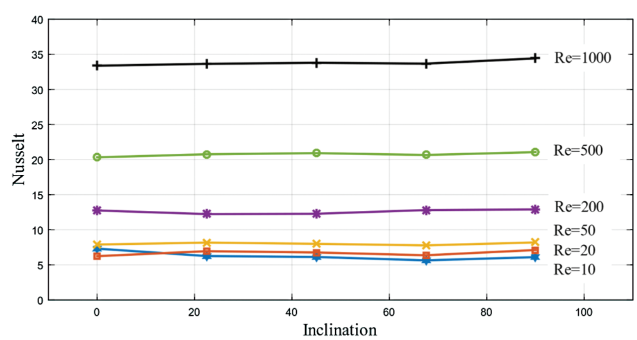


Figure 5: Nusselt variation with inclination, for different values of Reynolds

4.2 Effect of Rayleigh Number

For the heating effect on the heat transfer to different value of Reynolds, we present the variations of Nusselt number with Reyleigh number for several Reynolds for inclination $\varphi = 0°$ (Fig. 6). The Nusselt number increases with Rayleigh number for low Reynolds numbers Re ≤ 120 that is the say in the rang to naturel convection, Re ≈ 120 mixed convection, but for large Reynolds numbers Re ≥ 120 rang of forced convection, the effect of Rayleigh number has no effect. So, the Reynolds number has a positive effect in low Rayleigh and no effect in large Rayleigh.

4.3 Temperature Profiles

To focus on the minimum flow to be used to cold the condenser heated at a constant temperature, we traced the temperature profile in the outlet for several Reynolds in the horizontal channel $\varphi = 0°$ (Fig. 7).

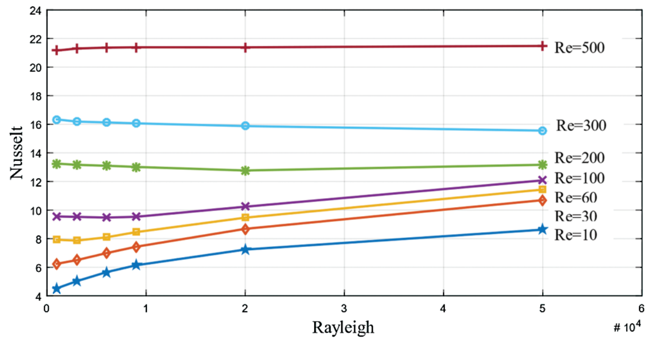


Figure 6: Nusselt variation with Rayleigh number, for different values of Reynolds

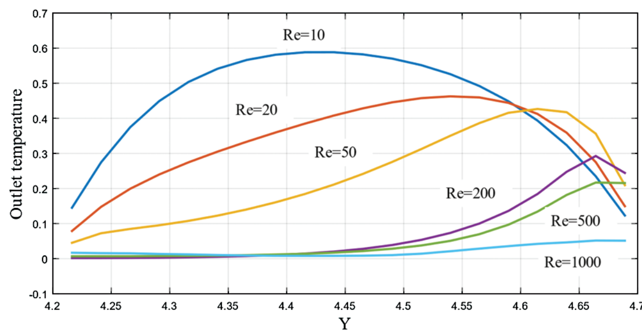


Figure 7: Variation of temperature in the outlet zone in horizontal channel ($\varphi = 0°$)

Generally, the temperature at the output to a maximum value, this value depends on the Reynolds number, it to a maximum value for low Reynolds $\theta_{\max} \leq 0.6$. Means that, in $\mathrm{Re} = 10$ ($\theta_{\max} = 0.58$ for example Fig. 8) if the wall temperature at $T_c = 100\ °C$ and the inlet water with $T_c = 20\ °C$ will quit the condenser with a temperature of $T = 66.4\ °C$. But in the case of $\mathrm{Re} = 500$ the temperature maximal is $\theta_{\max} = 0.21$ (Fig. 8) equal $T = 36.8\ °C$. In can conclude that, the average flow ($\mathrm{Re} \geq 120$) suffice to cold the condenser then minimize energy.

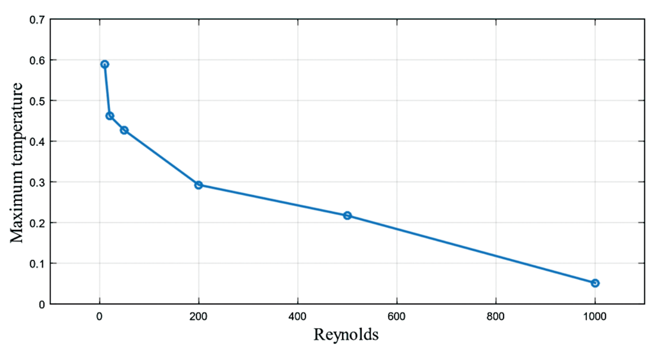


Figure 8: Variation of maximum temperature in the outlet zone *vs.* Reynolds number in horizontal channel ($\varphi = 0°$)

5 Conclusion

The cooling of the channels by the water, which simulates the chemical condenser, has been numerically investigated. The finite volume method has been used to solve the governing equations. The effects of the

Reynolds number, the inclination of the channel and the Rayleigh number on the flow structure and the thermal field have been examined. The main results can be summarized as follows:

- The effect of the slops of the chemical condenser appears just for the low Reynolds.
- The surface heat transfer rate increases with the Reynolds number.
- A new correlation has been proposed to calculate the average Nusselt number as a function of Reynolds number.
- The Nusselt number depends on the Rayleigh number only for the large values of the Reynolds number (forced convection).
- To minimize the energy, it is preferable to use average flow rates (Re > 120) to cool the chemical condenser.
- For low values of Reynolds (free convection), the cooling of condenser depends inversely proportional to tilt and directly proportional to the heating of the fluid. Unlike the case of great values of Reynolds (forced convection) where the cooling of condenser is independents of the both (tilt and heating of the fluid).

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Conflicts of Interest: The authors declare that they have no conflicts of interest to report regarding the present study.

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