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RESEARCH ARTICLE

STUDY OF THE TEMPERATURE DISTRIBUTION IN A PARABOLIC CYLINDRICAL CONCENTRATORUSING A NANOFLUID AS HEAT TRANSFER FLUID

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Abstract

We have numerically studied the heat transfers in a Cylindrical-parabolic concentrator using a nanofluid as heat transfer fluid for its use in Solar Thermodynamics. After an analysis of the work relating to solar concentrators and nanofluids, we have, after having described our physical system and posed working hypotheses, written the equations which govern the transfers in our collector. The latter as well as the associated boundary conditions were then dimensionless in order to generalize the problem and reveal the parameters that control the operation of the absorber. To solve our equations, we used the method of finite differences and the algebraic system obtained is solved thanks to the Thomas method combined with an iterative process of line-by-line relaxation type. The computer code that we developed made it possible to find the temperature distributions according to the spatial coordinates and at different times. The effects and influences of wind effect, axial and transverse thermal dispersion, absorber length, geometric shape factor on the average temperature distributions of the coolant is analyzed. At the end of the study, we were able to identify the most important physical and geometric parameters which give our system optimum operation for its use in Solar Thermodynamics

Introduction:-

Heat transfer is a process of great importance, it forms the basis of many industrial processes that are present in our daily lives. The intensification of these exchanges and the improvement of efficiency have now become a major problem in the industrial world, in regulatory bodies, but also in society as a whole, which is becoming aware of the progressive depletion of energy resources and who cares about the energy future. The improvement of heat transfer by convection is the main object of several works relating to the description of the phenomena managing convection, the effect of the nature of the systems in which it takes place (especially geometry), and the properties of fluids. involved (physico-chemical properties). Ideas for improvement have mainly affected the geometry of systems, and the physico-chemical nature of convective media, the work has only affected the macroscopic or sometimes microscopic order of the process. But with the appearance and rapid development of nanosciences and nanotechnologies during the second half of the 20th century, convection took a large part of this new richness, and took on another aspect of improvement: it is at the nanometric level of the matter of the convective medium that

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recent work has focused on. Nanofluids are then one of the fruits of such wealth. Having particular and interesting physico-chemical properties such as their high thermal conductivity, nanofluids offer a heat transfer coefficient that is unbeatable by other coolants [1][2][3]. With regard to applications in the field of heat transfer, studies carried out over the past ten years have shown that the addition of nanoparticles to a basic fluid could increase the heat transfer compared to the case of the pure fluid in significantly modifying the thermal conductivity of the carrier fluid [4][5][6]. Faced with energy and environmental issues, the technological challenge lies in the development of new processes for better energy management [7][8][9]. There is a real demand in the industrial world to develop new strategies to improve thermal behavior in solar collectors [10][11][12]. This work is part of this framework, and particularly concerns the problems related to the improvement of heat exchanges in the receiver tube of Cylindrical-parabolic collectors which are currently the most proven solar concentration techniques. This present study follows that of A. Mar [13] who used an overall thermal balance to find the average temperature of the heat transfer fluid outlet without taking into account the effects of thermal dispersion, the variation of the longitudinal velocity in the direction transversal among others [14]. To appreciate and quantify these effects we will, from the local equation of the quasi-stationary enthalpy, seek the influences of the effects of the control parameters on the spatio-temporal distribution of the temperature of the coolant.

Mathematical Approach:-

The physical system consists of a linear reflector which collects and concentrates the solar radiation on a collector which is a cylinder of radius R made of a homogeneous material, non-deformable and of conductivity λ in which circulates the heat transfer fluid placed along its focus line. The entire cylindrical-parabolic concentrator is mobile, it is oriented along a north-south axis and follows the path of the sun from east to west. At the initial instant, the reflector reflects on the cylinder a thermal flux density and at the same time, a fluid of temperature $T_a$and of inlet speed $\overrightarrow{V_e}$ circulates in the cylinder.

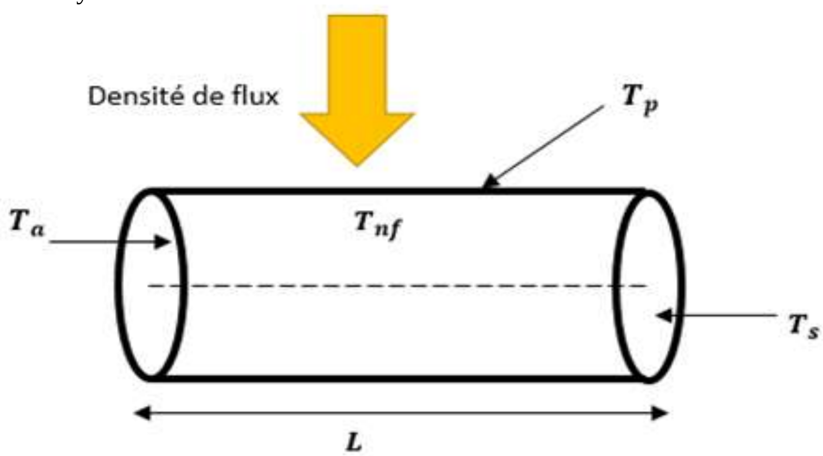


Figure 1:- Block diagram of the receiving tube.

To do this, we start from the equation of the energy balance, of the heat and of continuity on an element of volume at the level of the receiving tube. Using the pseudo-steady state, the flow is unidirectional and of the form:

$$\vec{V}(r) = \frac{2Q}{\pi R^4}(R^2 - r^2)\vec{e}_z \qquad (1)$$

The adimensionalization of the equations gives a better approach to the reality of physical phenomena; it allows the reduction of parameters, the comparison of effects and the generalization of the study. To reduce the heat equation to a dimensionless form, it is necessary to define reference quantities allowing us to form reduced quantities. We pose

$$\tilde{V} = \frac{V}{V_r}; \quad \tilde{T} = \frac{T-T_a}{\delta T_r}; \quad \tilde{z} = \frac{z}{L}; \quad \tilde{r} = \frac{r}{L}; \tilde{R} = \frac{R}{L}$$

With$\tilde{V}, \tilde{T}, \tilde{z}et\tilde{r}$are the reduced magnitudes.

By taking into account the quantities of reference, the equation of the heat in the adimensional form is written:

$$\tilde{V}\frac{\partial \tilde{T}}{\partial \tilde{z}} = \frac{1}{Pe}\left(\frac{\partial^2 \tilde{T}}{\partial \tilde{z}^2} + \frac{\partial^2 \tilde{T}}{\partial \tilde{r}^2}\right)(2)$$

avec $Pe = Re.Pr$: the thermal Péclet number characterizing the heat dissipation and defined by $Re = \frac{VL}{\upsilon}$ and $Pr = \frac{\upsilon}{\alpha}$ are the Reynolds and Prandtl numbers respectively.
If we choose: $V_r = \frac{2Q}{\pi R^2}$ alors $\vec{\tilde{V}}(r) = (1 - f^2.\tilde{r}^2)\vec{e}_z$ (3)
With $f = \frac{L}{R}$: the geometric form factor of the collector
Conditions to the limits:
$\tilde{z} = 0\ ;\ \tilde{T}(\tilde{z} = 0, \tilde{r}) = 0$ (4)
$\tilde{z} = 1\ ;\ \frac{\partial \tilde{T}}{\partial \tilde{z}} = 0$ (5)
$\tilde{r} = 0;\ \frac{\partial \tilde{T}}{\partial \tilde{r}}\Big|_{\tilde{r}=0} = 0$ (6)
$\tilde{r} = 1;\ \tilde{q}_L(\tilde{z}, t) = \frac{\partial \tilde{T}}{\partial r}\Big]_{\tilde{r}=\frac{1}{f}} + Bi.\tilde{T} + \frac{1}{N}.\left\{\left(\frac{T}{\delta T_r}\right)^4 - \left(\frac{T_a}{\delta T_r}\right)^4\right\}$ (7)

Numerical Method:-

The transfer equations that describe our problem are nonlinear and coupled partial differential equations. Due to their complexity, these equations are solved by numerical techniques. The finite difference method was used for the discretization of the final system of non-dimensional equations with the boundary conditions while a purely implicit scheme is adopted for the temporal discretization [15]. In order to find the spatio-temporal distributions of temperatures, the physical domain is approached by a discrete domain formed by a network of nodes on which the temperatures are calculated. The system of equations obtained will then be solved using Thomas's algorithm combined with an iterative process line by line. After having calculated the $T_{i,j}$ they must then be compared with the field of arbitrary temperatures $T_{i,j}^a$. To stop the iterative process, we require that the following criterion be respected:
$\sum_{i=1,j=1}^{i=i_m,j=j_m}\left\{\frac{\left|T_{i,j} - T_{i,j}^a\right|}{\left|T_{i,j}\right|}\right\} < \text{Critère}$ (8)
If this criterion is not met [16], the process is resumed by replacing the arbitrary temperature:
$(1 - \omega).T_{i,j} + \omega.T_{i,j}^a$ (9)

Results and Discussions:-

In this chapter, we will analyze the results from our calculation code. We will first give the calculation conditions and then we will comment on the selected results. These results come from a computer code written in Fortran 95 that we have developed. We place ourselves in the case where the losses by radiation are very low compared to the losses by convection due to the effect of the wind despite the fact that the parietal temperatures are very high. In other words, it is assumed that the coating on the external face of the absorber has a very low emissivity.

We will analyze the influences of certain geometric and physical quantities on the temperature distribution. To carry out our analyses, we will first comment on the results when L=5 m: Pe=300; $f = 200$; Cs=100; Bi=0.1. The results obtained for these values will serve as a reference when analyzing the effects of certain quantities.

The curves of figure 2 show us that when L≤3 m the variations of the average temperature are quasi-linear. This means that in this region, the axial and longitudinal dispersion effects are negligible compared to the transport effects. As one goes deeper into the pipe, the rate of variation decreases because of the losses by the effect of the wind which take on more and more importance and the contribution of thermal diffusion. On the other hand, we find that with these data, this absorber works perfectly in Solar Thermodynamics especially between 11 a.m. and 3 p.m.

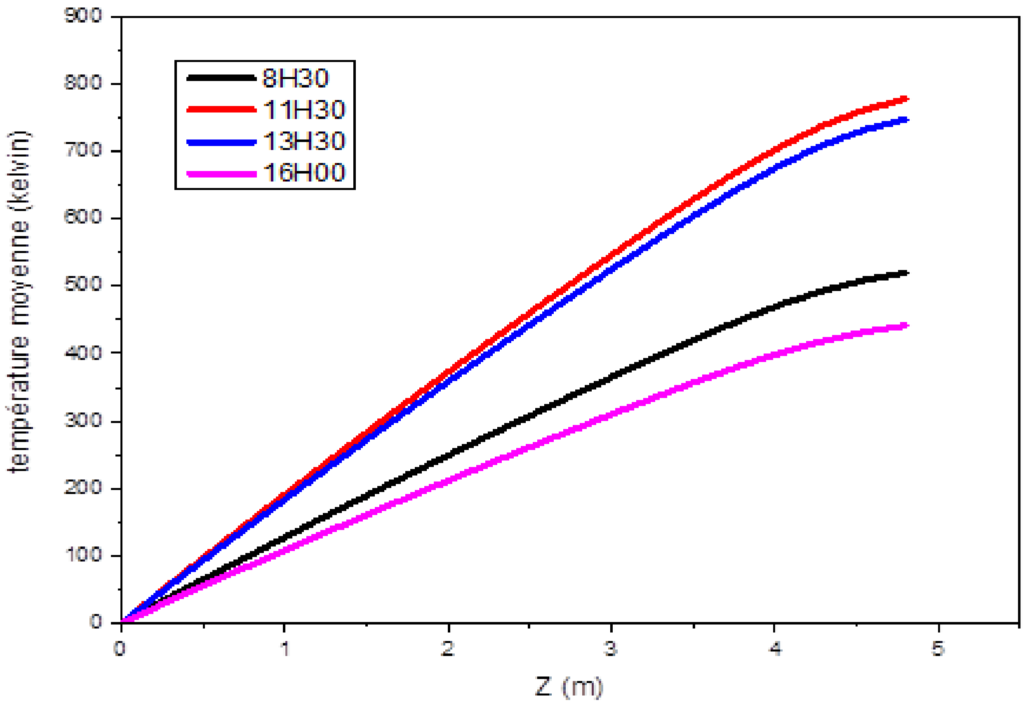


Figure 2:- Variation of the average temperature as a function of z for different times. L=5m; Number of Peclet=300; Form Factor=200; Power factor=100; Biot=0.1.

Influence of longitudinal dispersion on mean temperature

When we neglect the axial dissipation, the variations of the average temperature obtained are linear over the whole length of the absorber and are, for the same time, very far superior to those of figure 3. This shows that in the absorbers with high concentration the effects of axial diffusion should not be neglected because the temperature values are overestimated and therefore the dimensioning of the absorber can be distorted.

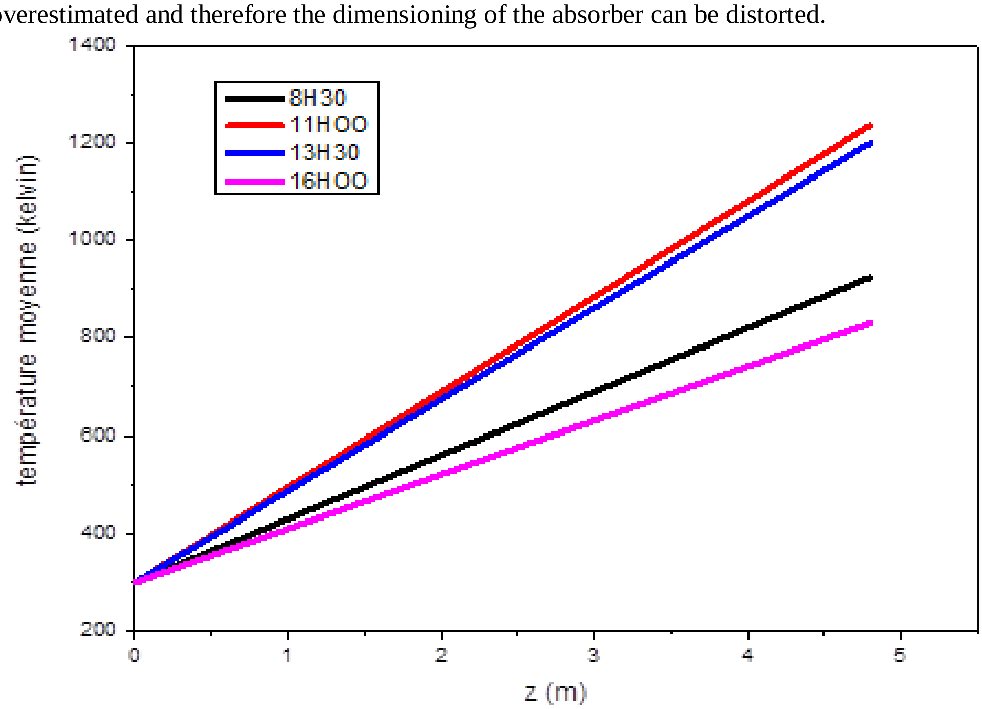
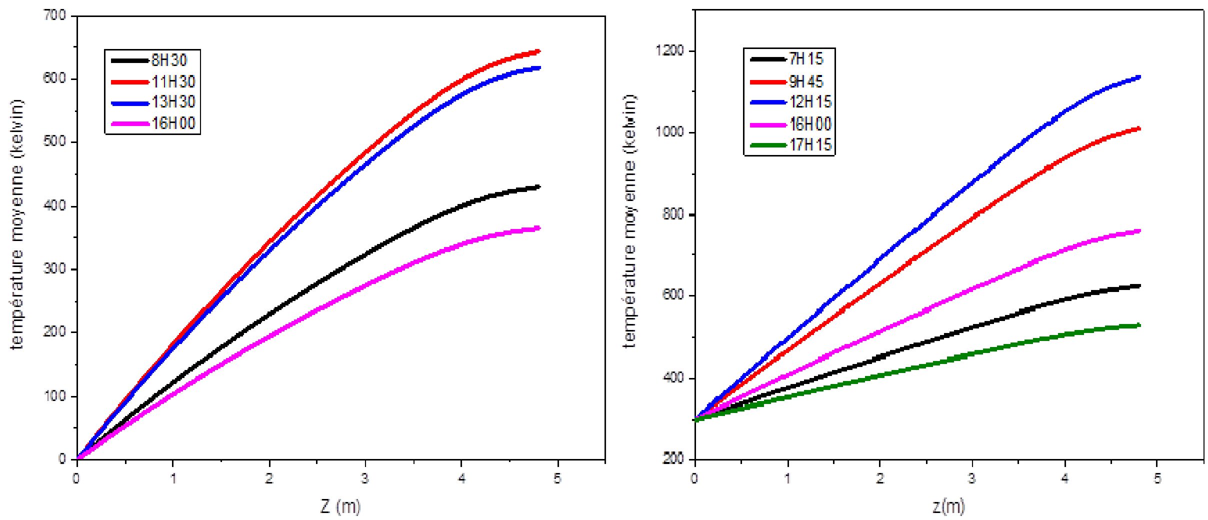


Figure 3:- Variation of the average temperature as a function of z for different times L=5m; Number of Peclet=300; Form Factor=200; Power factor=100. ;Biot=0.1.

Influence of wind losses on average temperature

We first notice that the Biot number curves (Figure 4) do not affect the temperature profiles. However, the values of the mean temperature are very sensitive to those of the Biot number. It should be noted that as soon as the Biot number is greater than 0.1 the effects of the wind become predominant over the effects of heat transport.

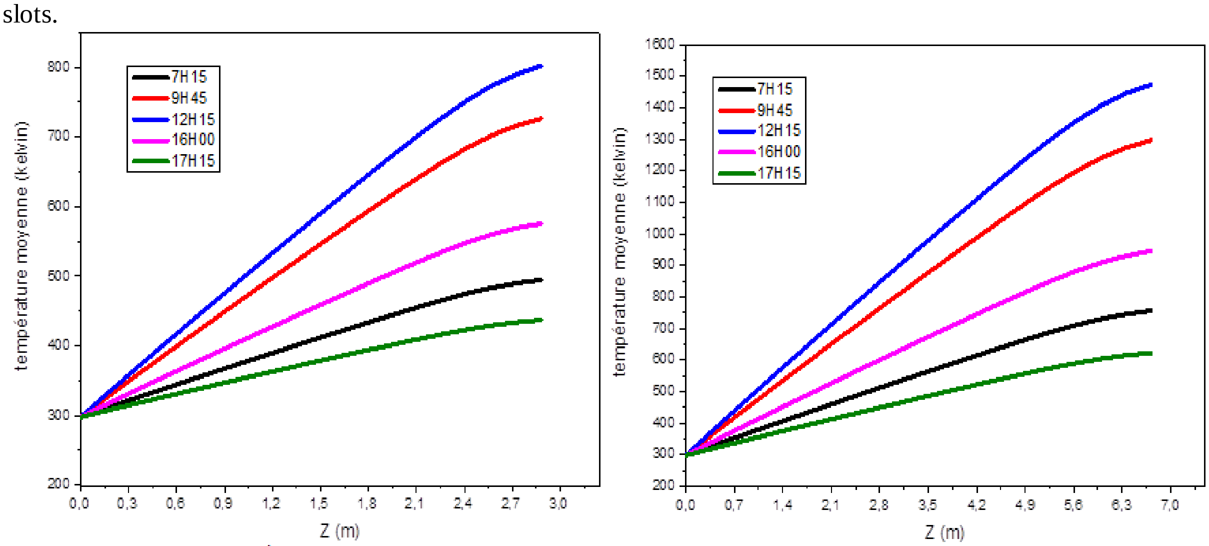


a) Nombre de Biot = 0,5 b) Nombre de Biot = 0,05

Figure 4:- Variation of the average temperature as a function of z for different times.

Influence of the length L of the absorber on the average temperature

When the length of the absorber (Figure 5) increases the average temperatures also increase. With the calculation parameters with which we conducted our simulations, the temperature increases are explained by the fact that the cumulative heat recovered by the heat transfer fluid during its transport is greater than the sum of the losses and the effects of diffusion. On the other hand, for absorber lengths of less than 3 m, this system should not be used for Solar Thermodynamic purposes because outlet temperatures above 650 K are only obtained for very limited time slots.

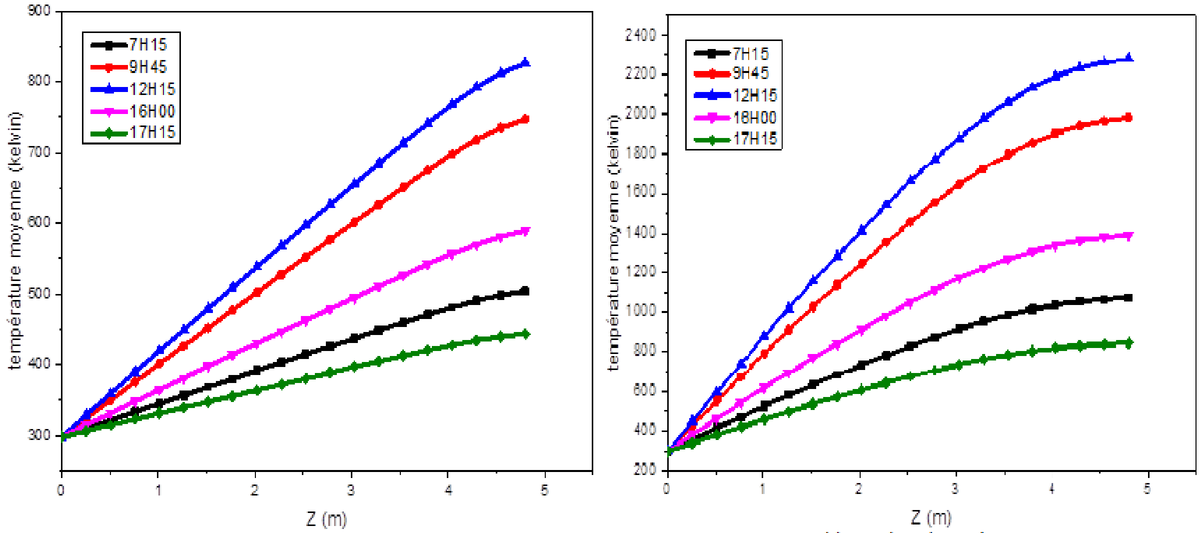


a) L = 3m b) L =7m

Figure 5:- Variation of the average temperature as a function of z for different times.

Influence of the Peclet number on the average outlet temperature

When the Peclet number (Figure 6) increases, the average temperatures decrease. Indeed, with high flow rates, the residence time of the fluid in the absorber decreases and therefore the fluid benefits less and less from the contribution of incident solar radiation which is the driving force behind the rise in temperature. In these types of collector, it is therefore more appropriate to work with moderate flow rates if one is more interested in the thermal potential than in the quantity of heat collected.



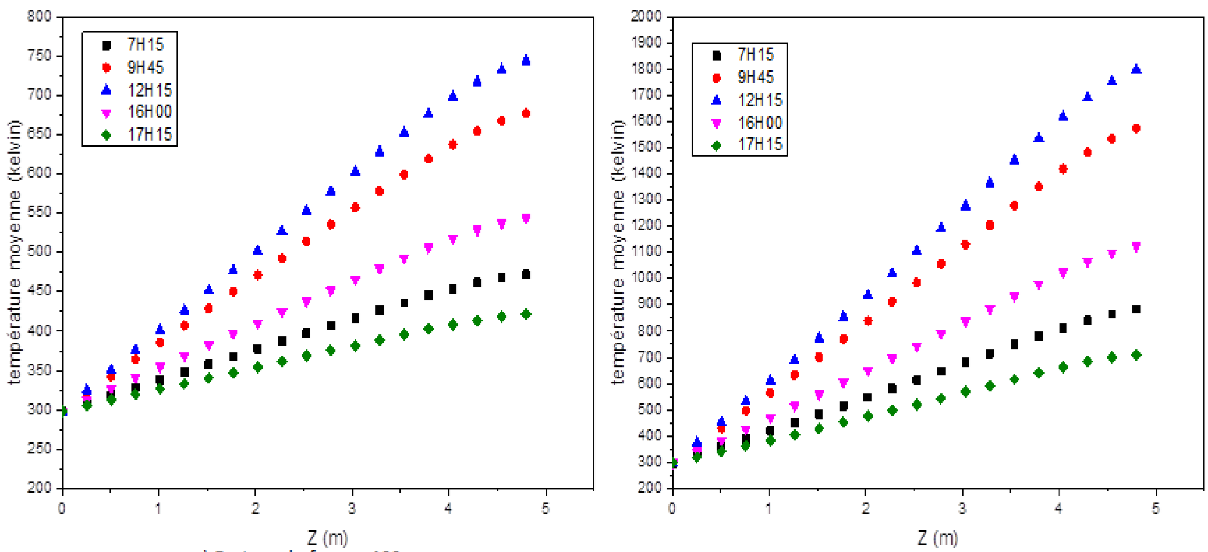
a) Nombre de Peclet = 500

b) Nombre de Peclet =100

Figure 6:- Variation of the average temperature as a function of z for different times.

Influence of the form factor on the average output temperature

To study the influence of the form factor f (figure 7), we set the length of the absorber to 5 m. In other words, we analyze the effect of the section of the absorber. When the section of the absorber decreases the average temperature of the fluid increases because the effects of radial diffusion are attenuated and therefore the temperature of the fluid tends towards the temperature of the lateral surface. It should also be noted that as soon as the shape factor f is less than or equal to 300, all the average temperatures at the outlet of the absorber are greater than 650 K between 7 a.m. and 5 p.m.



a) Facteur de forme=100

b) Facteur de forme=300

Figure 7:- Variation of the average temperature as a function of z for different times.

Conclusion:-

This study relates to the determination of the spatio-temporal distributions of the temperatures of a nanofluid in forced flow in a cylindrical absorber of horizontal axis whose lateral surface is subjected to a very high density heat flow for its use in Thermodynamics. Solar. In order to take into account, the effects of axial and longitudinal dispersion, we have numerically solved the local equation of the heat of a nanofluid in a cylinder with a horizontal axis under the hypothesis of longitudinal transport. After having adimensionalized the transport-diffusion equation obtained in order to reveal the control parameters, we approached it by discretizing it by finite differences. The resulting system of algebraic equations is then solved numerically using a computer code that we have developed. We have in this chapter analyzed the effects of certain control parameters such as those of the effect of the wind, the form factor, the longitudinal dispersion and the length of the absorber on the temperature distributions of the coolant. We conducted our analyzes by taking an absorber with the characteristics: L=5 m; Pe=300; f=200; Cs=100; Bi=0.1 as a reference model. We have also shown that to achieve the objective we have set, namely to obtain average outlet temperatures above 650 K, the system must be operated in well-defined time slots.

Nomenclature

Latin letter	
Bi: parameter of torus pole $[m]$	C_s: the solar constant
g:intensity of gravity	f: geometric form factor
L:absorber tube length[m]	N: Nusselt number
Q: volume flow rate	Pr: Prandtl number
R: absorber tube radius[m]	Re Reynolds Number
t: dimensionless time[s]	T: dimensionless temperature[K]
T_a : temperature at the initial moment $[K]$	
$\vec{V}$: dimensionless velocity components in the transformed plane	x, y, z: cartesian coordinates $[m]$
Greek symbols	
α : thermal diffusivity$[m^2s^{-1}]$	λ: thermal conductivity$[Wm^{-1}K^{-1}]$
ν: kinematical viscosity $\left[m^2s^{-1}\right]$	δT_r: difference of temperature [K]

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