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Experimental Study of the effect of parameters of design in the performance of a solar air heater with the Fresnel concentration

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Abstract—**In this theoretical and experimental study, we are interested in the effect of the parameters of design on the thermal performances particular the temperature of exit during the flow of a blade of air in the absorbent useful conduit, which placed in the focal zone of a solar collector with effect of concentration linear of Fresnel. The rectangular useful conduit have a 2.32 [m] length is equipped with twisted rectangular artificial roughness's, which are made of steel galvanized thickness 0.5 [mm] and laid out in conjunction in order to ensure a disordered and turbulent flow.**

The linear solar concentrator of Fresnel consists of two elements, comprising each one 40 ordinary mirrors perfectly reflective, are fixed on metal sheets to ensure the rigidity of the system during the operations of adjustment and maintenance.

An experimental study is started, to highlight the evolution of the thermal performances of the solar collector considered, primarily the temperature of the fluid at the exit of the absorbent useful conduit, balance of thermal energy and the thermal efficiency.

***Keywords*— Fresnel, Solar collector with concentration, heat transfer, roughness artificial.**

I. INTRODUCTION

In order to optimized the performance and the heat transfer in a rectangular useful conduit used in Linear Fresnel Reflector. Panna L.S et al [1] have carried an experiment study about a linear Fresnel reflection, with trapezoidal cavity absorbers, was provided with two types of pipe absorber respectively used round and rectangular. In this experimental study, they were studied experimentally at different concentration ratio, the results obtained presence that the thermal efficiency of round pipe was found higher, almost 8% as compared to rectangular pipe absorber.

N, Velázquez et al [2], through a numerical simulation of a Linear Fresnel reflection use as a solar cooling, have tried to understand the effect of design parameters on the thermal performance of the refrigeration system. The analysis of the heat balance in absorber has developed by a theoretical model, then validated by experimental results.

In this theoretical and experimental study, we investigate the effect of design parameters on the thermal performance of a Linear Fresnel solar reflection, including the evolution of thermal efficiency and the outlet temperature of air flowing in the useful rectangular conduit (absorber). The absorber was equipped with artificial roughness twisted made from galvanized steel with thickness 0.4 [mm], jointly set on the direction of flow by rivets to ensure perfect heat distribution and extract enough of calories from the absorber plate.

An experimental study was started, where we were interested in the thermal performances of the solar collector considered, primarily with the temperature of the heat transfer fluid at the exit of the useful conduit, as well as the thermal efficiency.

Fig.1: Linear Fresnel reflector solar concentrator with rectangular useful conduit absorber.

II. EXPERIMENTAL PROTOTYPE:

The experimental device appears in Fig.1; build in the mechanical laboratory of engineering of the University the

Biskra, whose dimensions geometrical and constituent elements were given in Table 1.

TABLE I
DIMENSIONS OF LINEAR FRESNEL REFLECTOR

Designation	Dimension [m]
Length of useful conduit absorber [L]	2.32
Width of the concentrator [Lar]	1.18
Number of mirrors [n]	80
Width of e each mirrors	0.1
Focal distance [f]	1
Width of the absorber duct [l]	0.23
Thickness of useful conduit absorber [e]	0.05
Height of artificial roughness [a]	0.045
Space between artificial roughness [b]	0.01
Thickness of the walls out of wood isolate	0.012

The experimental study presented in this work is carried in the open air near of the technological hall, of the mechanical department of engineering into the University of Mohamed Kaiderof Biskra.

The test data were measured at an average interval of 15 min, the experimental protocol need to use the following devices of measurement:

- Electronic pyranometerKipp& standard Zonen CM 11 with 1% accuracy for the measurement of the solar radiation adaptable on a horizontal and inclined surface.
- Digital k-type thermocouples with an accuracy of 0.01 ∘ C with range the (0 to 400 °C), for the measurement of the temperature of the air at the inlet and the outlet of the useful conduit absorber.
- Pressure transducer accuracy (Kimo CP301) with ±1 Pa and 0.5% of reading.
- Kimo-type anemometer with hot wire (VT300) with ±3% of reading and ±10 m 3 for the flow rate measurement and ±3% of reading and ±0.1 m/s accuracy were used.

III. THEORETICAL ANALYSIS:

This study is devoted to understand the heat exchange taking place in the useful conduit absorber with a rectangular section of dimension 2.32 × 0.23 [m]. The isolate wood has thickness 0.05 [m], the absorber plate is painted with black coating Fig.2.

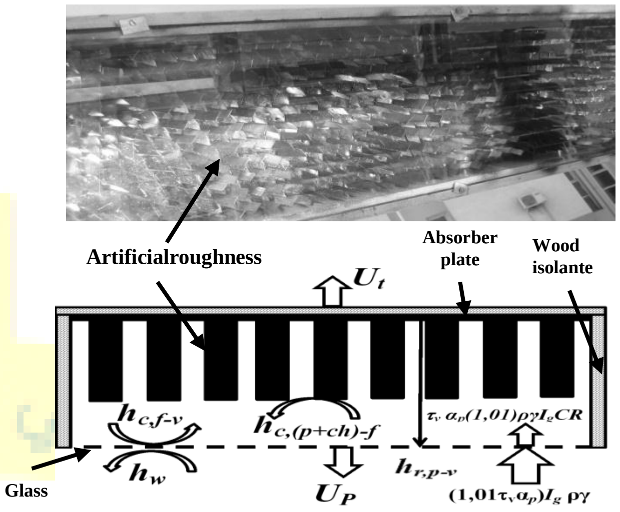

Fig.2: Schema of heat exchange in the useful conduit absorber.

The solar irradiation coming to the useful conduit absorber is estimated by following relation [3, 4]:

$$S_r = (1.01\tau_v\alpha_p)I_g\rho\gamma \quad (1)$$

By adopting the method by section and application of the electric analogy in the useful conduit absorber, Fig.2, 3, the equations of energy conservation give to us:

$$(h_{r,p-v} + h_{c,f-v} + U_{av})T_v - (h_{r,p-v})T_p - h_{c,f-v}T_f = \alpha_v\rho\gamma I_g \mathrm{CR} + U_{av}T_a \quad (2)$$

$$-(h_{r,p-v})T_v + (h_{c,(p+ch)-f} + h_{r,p-v} + U_{arr})T_p - h_{c,(p+ch)-f}T_f = \tau_v\alpha_p(1{,}01)\rho\gamma\, I_g CR + U_{arr}T_a \quad (3)$$

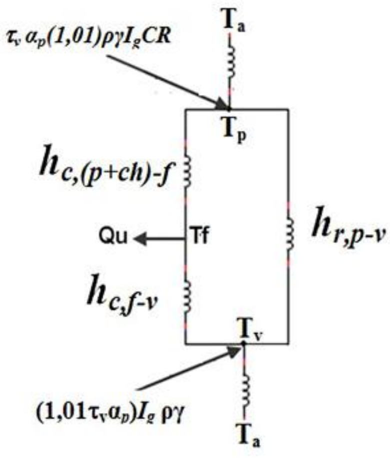

Fig.3: Electric diagram is equivalent relative to a section of the useful conduit absorber.

$$h_{c,f-v}T_v + h_{c,(p+ch)-f}T_p - \left(h_{c,f-v} + h_{c,(p+ch)-f} + 2\dot{m}\,Cp/_{A}\right)T_f = \left[2\dot{m}\,Cp/_{A}\right]\times T_{fe} \quad (4)$$

After mathematical simplifications the expression of the energy output through the air will be written as follows [3]:

$$Q_u = AF_R\left[(\tau_V\alpha_P(1{,}01)\rho\,\gamma\,I_g CR) - U_L\left(T_{fe} - T_a\right)\right] \quad (5)$$

The outlet temperature[3] of air from the useful conduit absorberwas expressed by

$$T_{fs} - T_a - \frac{\tau_V\alpha_p(1{,}01)\rho\,\gamma\,I_g CR}{U_L} \Bigg/ T_{fe} - T_a - \frac{\tau_V\alpha_p(1{,}01)\rho\,\gamma\,I_g CR}{U_L} = exp\left(-\frac{l\,F'U_L\,L}{\dot{m}Cp}\right) \quad (6)$$

The heat transfer coefficient $h_{(p+ch)-f}$resulted in the useful conduit absorber between the absorber plate and the heat transfer fluid is usually expressed by [3]:

$$h_{(p+ch)-f} = [Nu\times\lambda_f]/D_H \quad (7)$$

In this case of working conditions, we were adopted three correlations of the Nusselt number based by the flow regime, andthe type of artificial roughness equipped in the useful conduit absorber Fig.3.

- For 1900 ≤ Re ≤2300, we were used the correlation of employment Saini R.P and Saini J.S [4, 5]:

$$Nu = 4\times10^{-4}\times Re^{1{,}22}\left[\frac{a_{ch}}{D_H}\right]^{0{,}625}\left[\frac{s}{10.a_{ch}}\right]^{2{,}22}\left[\frac{b_{ch}}{10.a_{ch}}\right]^{2{,}26} exp\left[-1{,}25.\,ln\left(\frac{s}{10.a_{ch}}\right)^2\right] exp\left[-0{,}824.\,ln\left(\frac{b_{ch}}{10.a_{ch}}\right)^2\right] \quad (8)$$

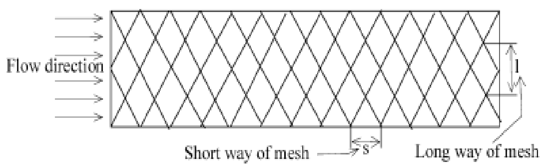

Fig.3: Artificial roughness used by Saini R.P and Saini J.S [5].

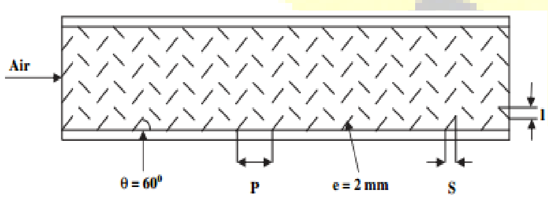

Fig.4: Artificial roughness used by Karmare and Tikekar[6]

- For 2300 ≤ Re <4000, The correlation of Kays [3] seems more convenient:

$$Nu = 0{,}0158\,Re^{0{,}8}\text{for } {}^{L}/_{D_H} > 10 \quad (9)$$

- For 4000 ≤ Re ≤ 17000, we were adopted the correlation of Karmare and Tikekar (2007) [6], whose arrangement of artificial roughness is near to the model considered Fig.4.

$$Nu = 2{,}4\times Re^{1{,}3}\left[{}^{a_{ch}}/_{D_H}\right]^{0{,}42}\left[{}^{l}/_{s}\right]^{-0{,}146}\left[{}^{p}/_{a_{ch}}\right]^{-0{,}27} \quad (10)$$

With, Re the Reynolds number expressed by [3]:

$$Re = V_f\,{}^{D_H}/_{\nu_f} \quad (11)$$

Where D_H corresponds to the hydraulic diameter are equivalent of the useful conduit absorber:

$$D_H = \frac{4.A_f}{P_m} = 2.\left(l.e - n_{ch}a_{ch}b_{ch}\right)\Big/\left(l + e + n_{ch}(a_{ch})\right) \quad (12)$$

The thermal efficiency of the Linear Fresnel reflector is indicated by the ratio of the useful output conveyed by the heat transfer fluid, with the solar energy concentrated by the mirrors of the Fresnel concentrator:

$$\eta_{th} = Q_u\Big/\left(\tau_v\alpha_p(1{,}01)\rho\gamma I_g CR\right)\times S_{Cap} \quad (13)$$

IV. Results:

The results resulting from the experiments made it possible to follow the evolution of the thermal performances of the Fresnel concentrator, in particularly the outlet temperature the air heater compared to that measured at the inlet of the useful conduit absorber, as well as the thermal efficiency, are then compared with those obtained by the numerical simulation, where we were adopted the model of Perrinde BRICHAMBEAUT theoretically to consider the solar radiation in the site of Biskra.

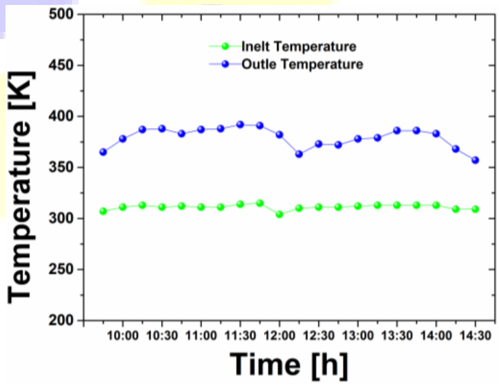

Fig.5: Evolution of the temperature at the inlet and outlet of the useful conduit absorber with an air mass flow of 0.019 [kg /s]

The evolution of the temperatures according to the time of the air heater at the inlet and the outlet of the useful conduit absorbent presented in Fig.5, show significant values which varying between 90 until a maximum of 120 °C obtained with approximately at 11h: 30, with a clear decrease with the approach of midday under the effect of a persistent cloudy passage.

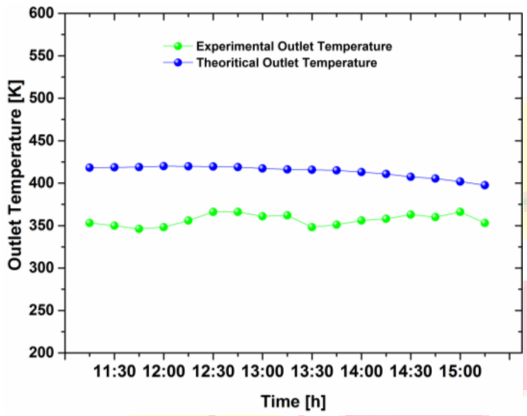

Fig.6: Evolution of the outlet temperatures of the air heater for a theoretical and experimental with an air mass flow of 0.019 [Kg /s]

The shape of the outlet temperatures curves of both theoretical and experimental air heater versus of time Fig.6, as well as thermal efficiency, Fig.6, shows a correlation between the measured values and those obtained by numerical simulation.

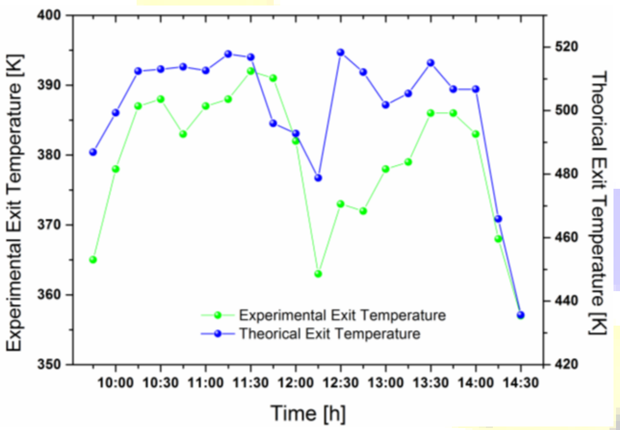

Fig.7Plot of the air exit temperature according to the time.

The Fig.7presents the evolution of the air exittemperatures both experimental and theoretical according to time, her we werenote that a great reduction in air exit temperature especially approximately tomidday and this is due to the presence of the clouds.

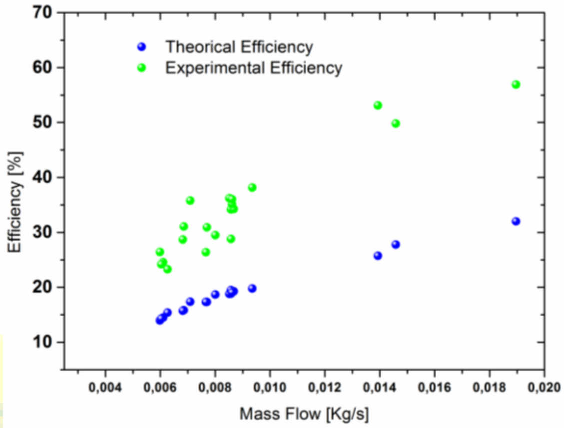

Fig.8: plot of the thermal efficiency according to the mass flow of air.

The Fig.8show the curves of evolution of the thermal efficiency according to mass flow of air, we recorded an increase in efficiency with the increase in the mass flow of air injected into the absorber, we also notices that the experimental efficiency remains always higher than the theoretical yield, this is due to temperatures higher in the ideal model (theoretical resultants) compared to the experimental model, i.e. the losses heat with environment are more important.

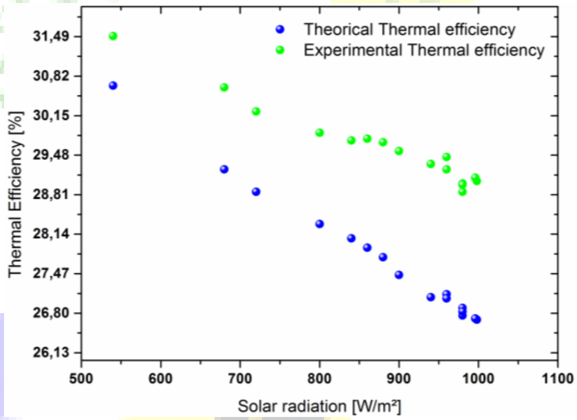

Fig.9: plot of the thermal efficiency according to the solar irradiation.

The Fig.8 presents the variations of the thermal efficiency according to the measured total solar radiation;we can see that the thermal efficiency decreases with the increase in the solar radiation. That it is for the theoretical yield with the experimental good, this reduction is due to the thermal losses which increase with the increase in the inlet and outlet temperatures and of air, in addition in the Fig.7, and we recorded the same evolutions of the outlets temperatureof air.

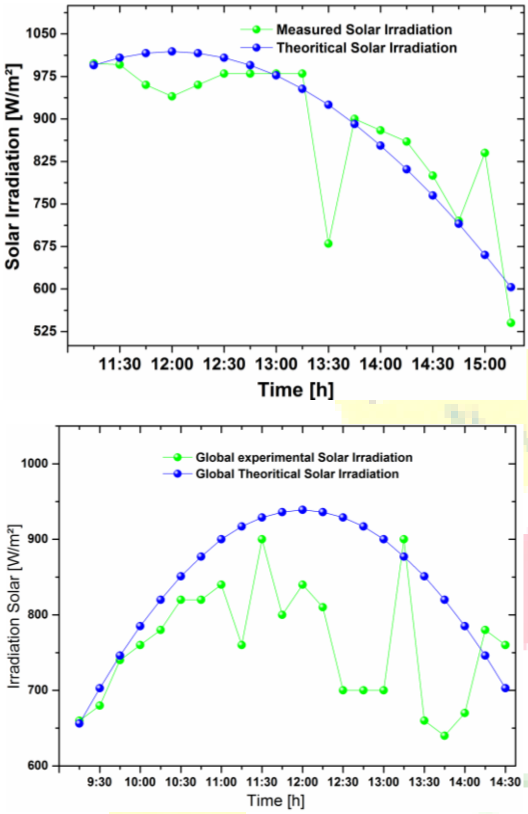

Fig.10: Evolution of the solar irradiation overall theoretical and experimental according to the time.

The difference between the theoretical and experimental values is mainly due to optical losses by misdirection elements that support the concentrator mirrors and errors induced by the model of Perrin de BRICHAMBEAUT which proved particularly inadaptable sky conditions covered Fig.10.

V. Conclusions

Considering the range of temperatures obtained between 70 and 120 ° C for the outlet of the considered useful conduit absorber of the Linear Fresnel concentrator, using a mass flow of 0.019 [kg/s] and a thermal efficiency of about 18%, the heating applications like the heat storage and drying are encouraging and more convenient.

The analysis of the difference between the experimental results compared with those obtained by numerical simulation is mainly due to errors induced by the semi-empirical model of Perrin BRICHAMBEAUT unsuitable climatic with overcast.

Acknowledgment

In order to optimized this dispositive an depth study are in

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NOMENCLATURE

A	Area activates of the Linear Fresnel reflection [m²]
a_{ch}	Height of the artificial roughness [m]
A_f	Bypass area of the air [m²]
b_{ch}	Space between artificial roughness [m]
Cp	Specific heat of air [J/kg-K]
CR	Solar concentrated energy available
D_H	Hydraulic diameter [m]
e	Thickness of the useful conduit absorber [m]
F'	Local coefficient of transfer effectiveness of air-plate absorber
$h_{c,(p+ch)-f}$	Coefficient of transfer by convection (plate absorbent, artificial roughness and air) [W/m²-K]
$h_{c,f-v}$	Coefficient of transfer by convection between air and glass [W/m²K]
$h_{r,p-v}$	Coefficient of irradiative transfer glass-plate absorber [W/m²-K]
h_w	Coefficient of transfer by convection caused by wind [W/m²-K]
l	Width of the useful conduit absorber [m]
L	Length of the useful conduit absorber [m]
I_g	Direct solar irradiation [W/m²]
$\dot{m}$	Mass air flow [kg/s]
P	Artificial roughness pitch [m]
Qu	Useful energy recovered [W/m²]
s	Short way of mesh [m]
S_{cap}	Open area under the shaded area [m]
Sr	Solar irradiation [W/m²]
T_a	Ambienttemperature [°C]
T_f	Average temperature of the moving air [°C]
T_{fe}	Air temperature at the inlet of the useful conduit absorber [°C]
T_{fs}	Air temperature at the outlet of the useful conduit absorber [°C]
T_p	The plate absorber temperature [°C]
T_v	Glass temperature [°C]
U_{arr}	Back heat loss coefficient [W/m²-K]
U_{av}	Forward heat loss coefficient [W/m²-K]
U_L	Overall heat loss coefficient [W/m²-K]
ρ	Reflective surface
τ_v	Optical coefficient of transmission of the glazing 0.84
α_v	Absorption coefficient of the glass 0.06
α_p	Absorption coefficient of the absorption plate0.95
γ	Invoice due to the false orientation of concentrator
λ_f	Conductivity of the Air [W/m-K]
η_{th}	Thermal efficiency [%]