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Studying the Performance of Back – Pass Plain Plate Solar Air Heating Un-Glazed Collector

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Abstract:

This paper presents a studying performance of Back – pass plate solar air heating un-glazed collector that is simply say the recent cladding of new building and industrial facade. The use of a solar energy as a source of air heating used for ventilation or air preheating will consequently reduced the energy consumption which is the aim of this paper.

This paper describes the main characteristics of this type of air collector and the heat balance between all its components to reach to an equation which shows the air temperature variation across the collector and to study the effects of its main parameters on the temperature rise between the inlet initial air temperature and its outlet air temperature and made the possibility to design such type of air heating system collectors.

The main parameter which plays the main role in temperature variation and rise in this type of collector is the solar radiation, it is calculated for the vertical walls facing south and then it is changed to an equation describe its variation through the daylight time for clear and shiny day, the equation is a half – sine model.

A useful equation was found to predict the temperature variation and the rise on the air collector and gives a good indication about the system performance to make it simply to understanding the work of this type of air collectors.

الخلاصة:

إن هذا البحث يطرح دراسة لأداء نوع من أنواع مجمعات الطاقة الشمسية والتي تدعى (المجمع الشمسي لتسخين الهواء المار من الجهة الخلفية للوح المعدني الغير مغطى بالزجاج), المستخدم حديثا في واجهات الأبنية والمعامل الحديثة. إن إستخدام الطاقة الشمسية كمصدرلتسخين الهواء المستخدم لأغراض التهوية أو التسخين المسبق للهواء سيؤدي بالنتيجة إلى تقليل وترشيد الطاقة المستهلكة والتي هي هدف هذا البحث.

في هذا البحث وصف لخصائص هذا النوع من مجمعات تسخين الهواء وعملية التوازن الحراري بين جميع مكوناته للوصول إلى معادلة تظهر تغير درجات الحرارة وأرتفاع درجات الحرارة ما بين درجة حرارة الهواء الداخل إلى المنظومة والخارج منها, بالإضافة إلى دراسة تأثير العوامل الرئيسية التي تدخل في هذه المعادلة على تغير درجات الحرارة وإرتفاعها وكذلك

إمكانية تصميم مثل هذا النوع من مجمعات تسخين الهواء المستخدم في أعمال التهوية. إن العامل الرئيسي الذي يلعب دورا رئيسيا في تغير درجات الحرارة وإرتفاعها في هذا النوع من المجمعات هو الإشعاع الشمسي,لقد تم حساب كمية الإشعاع الشمسي الساقط على الجدران العمودية المواجهة للجنوب لنهار مشرق وصافي, ومن ثم تحويل هذه الأرقام المجدولة إلى معادلة رياضية بتطبيق موديل الجيب النصفي (Half – Sine). لقد تم الوصول إلى صيغة معادلة رياضية لتخمين تغير وإرتفاع درجة الحرارة للهواء المار عبرمجمع تسخين الهواء والتي أعطت نتائج ومؤشرات جيدة حول أداء هذه المنظومة الحرارية وبسطت مفهوم وعمل هذا النوع من المجمعات المستخدمة لتسخين الهواء.

1. Introduction:

The Solar Air Heating (SAH) system is a proven system for heating or preheating ventilation air in various applications. An old design that was used in the 1980's and has recently appeared is the back pass unglazed collector. The difference between a back pass non perforated collector and the transpired (perforated) collector is the lack of holes. The lack of holes means that the incoming air must pass as close to the back side of the metal panel if it is to remove the heat from the back side (referred to as back pass collector) [ref. (1)].

Solar Air heating system installations are beginning to be used more for the cladding of exterior walls, a schematic drawing for such system is shown in figure (1) below.

The system is commonly and most widely used in cooler months to heat ventilation air for industrial & commercial buildings and then requires less energy input to heat ventilation air, but it has also been used in processes such as drying agricultural crops and for other wide variety of products where heated air is an important requirement.

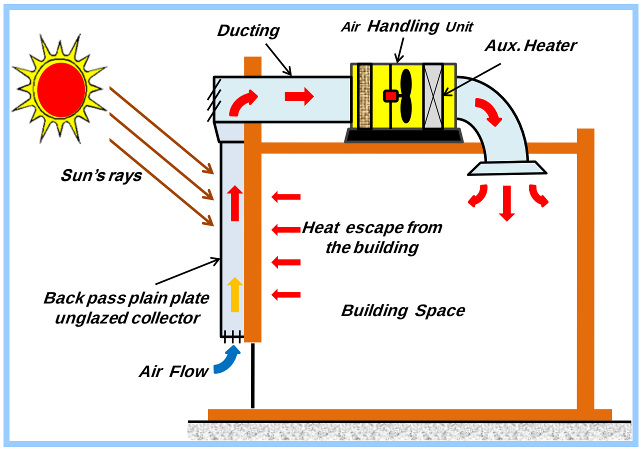

Fig. (1):Schematic drawing for Flat plate (Back pass unglazed collector) for industrial & commercial building.

The new construction for solar collector cladding allows the use of less expensive wall cladding material as a backing, the second most cost-effective is generally for retrofits when there are plans to repair an existing wall, improve the indoor air quality, or add more ventilation or makeup air to balance exhaust air [ref. 2].

2. The Description of Solar Air Heating Systems:

The Unglazed Plain plate collectors are sheets which are attached to the building's exterior walls to preheat ventilation air. A fan or Air handling unit is used to draw ventilation air (ambient air) from the bottom of the collector to the top [see fig. (1)]. During this time the collector plate which is heated by the incident sun's rays transfers its heat to the air flow passing in the cavity between the back side of the plate collector and building wall, and then the air flow is ducted into the building through the fan or the AHU.

At the same time the collector can capture the heat that would normally escape through the wall during winter or cold weather [see fig. (1)]. High-efficiencies are possible because the solar collector plate is only a few degrees warmer than the outdoor air. Therefore, there is little heat loss and most of the solar radiation is transferred to heat the air flow.

Bypass dampers can be located in the face of the canopy. These dampers allow ambient air to be fed directly into the building or process when no heating is required.

3. The Absorbers:

The metal panels (Absorbers) are a conventional metal facade. Panels are available in many colors including black and dark shades of brown, gray, red, blue and green. Ideally, the absorber plate should have high solar absorptance, extended surface area and be designed to promote turbulence to improve heat transfer.

For low temperature applications matte black paint is commonly used, with selective coatings reserved for high efficiency systems. The darker the better, as dark colors absorb more of the sun’s energy. Black is the best, followed by dark brown [ref. (3)].

The panel is 41.25” wide with 39.39” wide coverage (net width exposed to atmosphere), and are available in custom lengths (made according to required specifications). It is installed to give a continuous appearance along the entire wall. To add structural strength and rigidity, the material is processed through rollers to form corrugations.

The standard solar panel profiles are shown in fig. (2). The panels are made of either .032” thick aluminum or .027” thick pre-weathered zinc (resistant to bad weather) [ref. (3)].

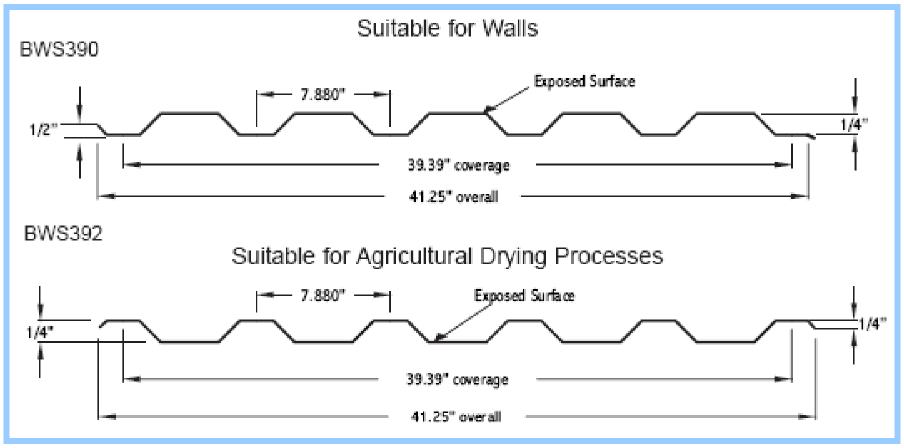

Figure (2): Schematic diagram of Unglazed Solar Panel used [ref. (3)].

4. Theoretical Analysis of the Unglazed Plain plate Collector:

To analyze the temperature rise and variation along the Unglazed plain plate collector, we should be first work to calculate the solar energy that can be collected on the plain plate and utilized as a heat transfer to the air flow passing in the cavity between the collector and the building's exterior wall, this energy is heating up the air flow temperature which in turns compared with the reference temperature for required processes to see if any additional heat should be added by the auxiliary heater or not.

To study this case figure (3 & 4) below is applied to represent a simplified schematic for unglazed plain plate collector model which has the dimensions of length (L_1), the depth of (L) and height of (x). Then to reach to a general formula for the collector temperature and how it is changed during the daylight time, the ***solar intensity*** which plays the major impact in this analysis should be known.

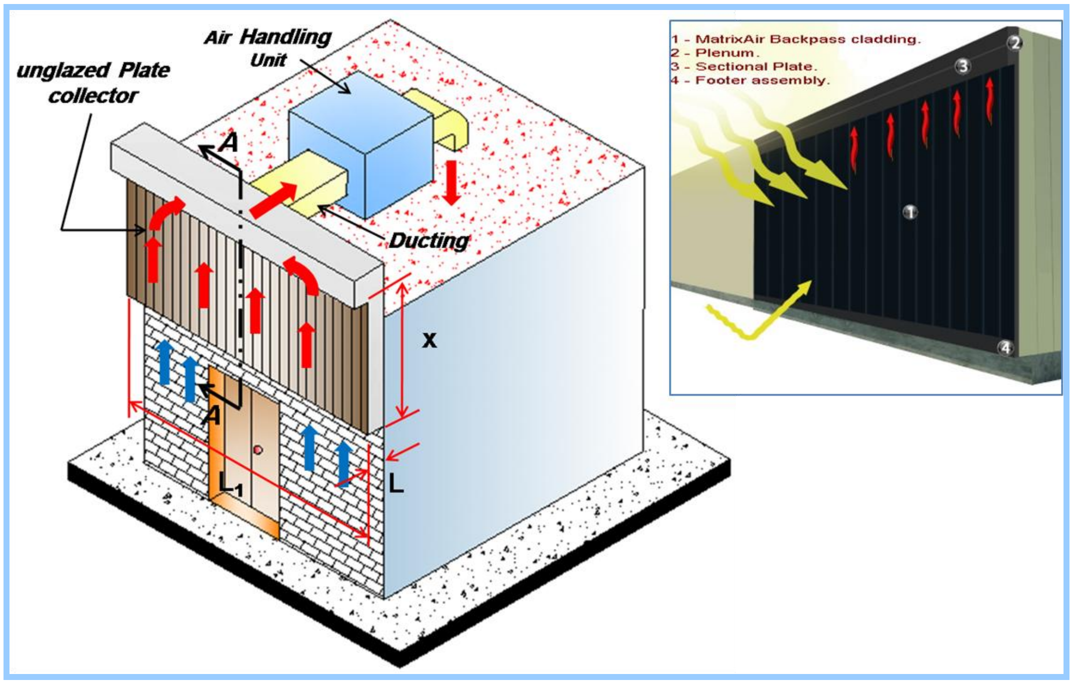

Figure (3): Schematic diagram of simplified Solar Air Heating System

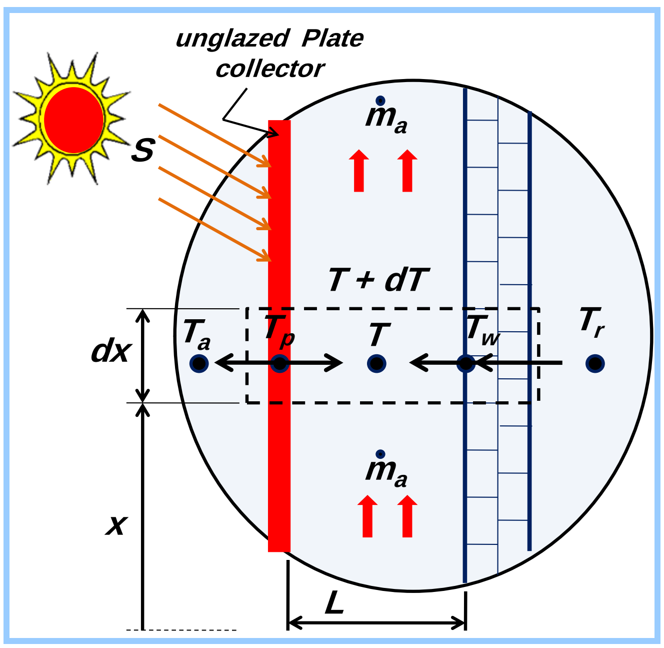

Figure (4): Schematic drawing of section A – A in fig. (3).

4.1. The Heat Balance for the Absorber plate:

The heat capacity of the absorber plate is assumed to be small, so it will be neglected during heating up process [the thickness is about 1 mm [ref. (3)], the absorber will be considered to have a uniform temperature within its thickness. The formulating equation for the air stream along the wall height (***x***) and time (***t***) will be derived below using simple energy balance. The absorber receives its heat from the incident solar flux during the daylight time, heats up and transfer part of it by convection to air flow passing through the cavity between the absorber and the building exterior wall, as well by radiation to the surface of exterior wall and losing heat to ambient. The heat balance equation [see fig. (4) above] for the unglazed plain plate collector (Absorber) per unit length along (L_1) is:

$$S.dx = h_{pa}\,.dx.\left(T_p - T_a\right) + h_{pf}\,.dx.\left(T_p - T\right) + \frac{\sigma.dx}{\left(\dfrac{1}{\varepsilon_p} + \dfrac{1}{\varepsilon_w} - 1\right)}.\left(T_p^4 - T_w^4\right)$$

This heat flow equation is linearised by extracting the temperature difference $\left(T_p - T_w\right)$ and thus the heat transfer coefficient for radiation ($\boldsymbol{h_r}$) is determined [ref. 4, 5]:

$$S\,.\,dx = h_{pa}\,.\,dx\,.\left(T_p - T_a\right) + h_{pf}\,.\,dx\,.\left(T_p - T\right) + h_r\,.\,dx\,.\left(T_p - T_w\right)$$

$$where: \qquad h_r = \left[\frac{\sigma.\left(T_p^2 + T_w^2\right).\left(T_p + T_w\right)}{\left(\dfrac{1}{\varepsilon_p} + \dfrac{1}{\varepsilon_w} - 1\right)}\right]$$

But, ($\boldsymbol{T_p^4 - T_w^4}$) can be approximated when the temperatures are close together by the below expression [ref. (6) page 159]:

$\left(T_p^4 - T_w^4\right) = 4T_{av}^3\,.\left(T_p - T_w\right)$, where: $T_{av} = \dfrac{T_p + T_w}{2}$, then ($\boldsymbol{h_r}$) becomes:

$$h_r = \frac{4\,.\,\sigma\,.\,T_{av}^3}{\left(\dfrac{1}{\varepsilon_p} + \dfrac{1}{\varepsilon_w} - 1\right)}$$

Then, by dividing the above equation by (***dx***), it becomes:

$$S = h_{pa}\,.\left(T_p - T_a\right) + h_{pf}\,.\left(T_p - T\right) + h_r\,.\left(T_p - T_w\right) \qquad \text{............} \quad (1)$$

Where:

- ***S*** = Incident solar flux absorbed in flat plate absorber (W/m^2)

- h_{pa} = Convective heat transfer coefficient for Absorber – ambient side (W / m^2. ^{o}C)
- T_p = Plate (Absorber) temp. (^{o}C)
- T_a = Ambient air temp. (^{o}C)
- h_{pf} = Convective heat transfer coefficient for Absorber – Air flow side (W / m^2. ^{o}C)
- T = The Air flow temp. (^{o}C)
- σ = Stefan – Boltzmann constant (equal to 5.6697 x 10^{-8} W/m^2 . ^{o}K)
- ε_p = Emissivity of the absorber plate
- ε_w = Emissivity of the building exterior wall.
- T_w = The building's exterior wall surface temp. (^{o}C)
- T_{av} = The average temperature of the absorber & exterior wall.
- h_r = The equivalent radiative heat transfer coefficient (W/m^2 . ^{o}C)

4.2. The Heat balance for the exterior wall surface:

The exterior wall surface receives heat from the absorber by radiation and from the building space through the wall which are transferred to the air flow moving inside the cavity. It appears that the absorber can capture the heat that would normally escape through the wall [fig. (4) above], the heat balance equation for the exterior wall is:

$$\left(\frac{\sigma \,.\, dx}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_w} - 1} \right) . \left(T_p^4 - T_w^4 \right) + U_w \,.\, dx \,.\, \left(T_r - T_w \right) = h_{wf} \,.\, dx \,.\, \left(T_w - T \right)$$

Substituting for $(T_p^4 - T_w^4) = 4\ T_{av}^3\ .\ (T_p - T_w)$ and dividing by (**dx**), the above equation becomes:

$$U_w \,.\, \left(T_r - T_w \right) + h_r \,.\, \left(T_p - T_w \right) = h_{wf} \,.\, \left(T_w - T \right) \qquad \ldots\ldots\ldots\ldots\ldots \quad (2)$$

Where:

- h_{wf} = Convective heat transfer coefficient for exterior wall – Air flow side (W / m^2. ^{o}C)
- T_r = Building space temperature (^{o}C)
- U_w = Wall heat transfer coefficient (W/m^2 . ^{o}C)

$$U_w = \left[\frac{1}{U_o} - \frac{1}{h_{wf}} \right]^{-1}$$

Where: U_o = Overall heat transfer coefficient of the wall (W /m^2 . ^{o}C)

4.3. The Heat Balance for the Air flow:

Referring to figure (4) above, the useful heat gains by the Air Flow moving in the cavity between the plate absorber and the exterior wall through the boundary (***dx***) can be represented by the following equation:

$$\dot{m}\,.C_p\,.\left(T + dT\right) = \dot{m}\,.\,C_p\,.T + h_{wf}\,.\,dx\,.\left(T_p - T\right) + h_{wf}\,.\,dx\,.\left(T_w - T\right)$$

The above equation becomes after rearranging and dividing by (dx) as following:

$$\dot{m}\,.\,C_p\,.\,\frac{dT}{dx} = h_{pf}\,.\left(T_p - T\right) + h_{wf}\,.\left(T_w - T\right) \qquad \text{.............} \;(3)$$

Where:

- $\dot{m}$ = Air mass flow rate (kg/sec)
- C_p = Air specific heat (W . sec /kg . ^{o}C)

From equation (2), we obtain the following equation for (T_w):

$$T_w = \frac{h_r\,.\,T_p + U_w\,.\,T_r + h_{wf}\,.\,T}{h_{wf} + h_r + U_w} \qquad \text{..............} \;(4)$$

Substituting equation (4) for (T_w) into equation (1), rearranging and solving for the solar radiation (***S***) incident on the absorber plate we will obtain:

$$S = K_1\,.T_p - K_2\,.T - K_3\,.T_r - h_{pa}\,.T_a \qquad \text{.............} \;(5)$$

<u>Where:</u>

- K_1 = Constant (W / m^2. ^{o}C)
- K_2 = Constant (W / m^2. ^{o}C)
- K_3 = Constant (W / m^2. ^{o}C)

$$K_1 = h_{pa} + h_{pf} + \frac{h_r\,.\left(h_{wf} + U_w\right)}{h_{wf} + h_r + U_w}, \quad K_2 = h_{pf} + \frac{h_r\,.\,h_{wf}}{h_{wf} + h_r + U_w}, \quad K_3 = \frac{h_r\,.\,U_w}{h_{wf} + h_r + U_w}$$

Solving equation (5) for (T_p), it will yield to:

$$T_p = \frac{S + K_2\,.\,T + h_{pa}\,.\,T_a + K_3\,.\,T_r}{K_1} \qquad \text{.......... ......} \;(6)$$

The term $U_w.(T_r - T_w)$ in the right side of equation (2) can be solved by substituting ($\boldsymbol{T_w}$) from equation (4) and then substituting for ($\boldsymbol{T_p}$) from equation (6), we will obtain:

$$U_w.(T_r - T_w) = K_5.T_r - K_4.T - K_3.\left(\frac{S + K_2.T + K_3.T_r + h_{pa}.T_a}{K_1}\right)$$

Where:

- $\boldsymbol{K_4}$ = Constant (W / m^2. ^{o}C)
- $\boldsymbol{K_5}$ = Constant (W / m^2. ^{o}C)

$$K_4 = \frac{h_{wf}.U_w}{h_{wf} + h_r + U_w}, \qquad K_5 = \frac{U_w.(h_{wf} + h_r)}{h_{wf} + h_r + U_w}$$

Simplifying the above equation, it will yields to:

$$U_w.(T_r - T_w) = \left(K_5 - \frac{K_3^2}{K_1}\right).T_r - \frac{K_3}{K_1}.S - \left(\frac{K_3.K_2}{K_1} + K_4\right).T - \frac{K_3.h_{pa}}{K_1}.T_a \qquad .... (7)$$

Substituting ($h_r.(T_p - T_w)$) from eq. (2) and (T_p) from equation (6) into eq. (1), it yields to:

$$S - h_{pa}.\left(\frac{S + K_2.T + K_3.T_r + h_{pa}.T_a}{K_1} - T_a\right) = h_{pf}.(T_p - T) + h_{wf}.(T_w - T) - U_w.(T_r - T_w)$$

Substituting [$U_w.(T_r - T_w)$] from equation (**7**) into the above equation, that will yield to:

$$h_{pf}.(T_p - T) + h_{wf}.(T_w - T) = \left(1 - \frac{h_{pa}}{K_1} - \frac{K_3}{K_1}\right).S - \left(\frac{h_{pa}.K_2}{K_1} + \frac{K_3.K_2}{K_1} + K_4\right).T$$
$$+ \left(K_5 - \frac{h_{pa}.K_3}{K_1} - \frac{K_3^2}{K_1}\right).T_r + \left(h_{pa} - \frac{h_{pa}^2}{K_1} - \frac{h_{pa}.K_3}{K_1}\right).T_a$$

But, we have from equation (3) that:

$$\dot{m}.C_p.\frac{dT}{dx} = h_{pf}.(T_p - T) + h_{wf}.(T_w - T)$$

Thus, equation (3) becomes:

$$\dot{m}.C_p.\frac{dT}{dx} = K_6.S - K_7.T + K_8.T_r + K_9.T_a \qquad .... (8)$$

Where:

- $\boldsymbol{K_6}$ = Constant (Dimensionless)
- $\boldsymbol{K_7}$ = Constant (W / m^2. ^{o}C)
- $\boldsymbol{K_8}$ = Constant (W / m^2. ^{o}C)

- K_9 = Constant (W / m^2. ^{o}C)

$$K_6 = 1 - \frac{h_{pa}}{K_1} - \frac{K_3}{K_1}, \qquad K_7 = \frac{h_{pa}\,.\,K_2}{K_1} + \frac{K_3\,.\,K_2}{K_1} + K_4$$

$$K_8 = K_5 - \frac{h_{pa}\,.\,K_3}{K_1} - \frac{K_3^2}{K_1}, \qquad K_9 = h_{pa} - \frac{h_{pa}^2}{K_1} - \frac{h_{pa}\,.\,K_3}{K_1}$$

But, ($\dot{m} = \rho\,.A\,.\,u$), where [A = the cavity cross – sectional area = Absorber length x Absorber width = 1 x L], thus [$\dot{m} = \rho\,.\,L\,.\,u$], where, [$\rho =$ Air density (kg / m^3] and [u = Air velocity inside the cavity (m / sec)], thus equation (8) becomes:

$$\frac{dT}{dx} = \frac{K_6}{\rho\,.\,L\,.\,u\,.C_p}\,.\,S - \frac{K_7}{\rho\,.\,L\,.\,u\,.C_p}\,.\,T + \frac{K_8}{\rho\,.\,L\,.\,u\,.C_p}\,.\,T_r + \frac{K_9}{\rho\,.\,L\,.\,u\,.\,C_p}\,.\,T_a$$

$$\frac{dT}{dx} = K_{10}\,.\,S - K_{11}\,.\,T + K_{12}\,.\,T_r + K_{13}\,.\,T_a$$

Where:

- K_{10} = Constant (m . ^{o}C / W)
- K_{11} = Constant (1 / m)
- K_{12} = Constant (1 / m)
- K_{13} = Constant (1 / m)

$$K_{10} = \frac{K_6}{\rho\,.\,L\,.\,u\,.\,C_p}, \quad K_{11} = \frac{K_7}{\rho\,.\,L\,.\,u\,.\,C_p}, \quad K_{12} = \frac{K_8}{\rho\,.\,L\,.\,u\,.\,C_p}, \quad K_{13} = \frac{K_9}{\rho\,.\,L\,.\,u\,.\,C_p}$$

For constant room or space temperature (T_r) and for simplicity, we will deal with (T_a) as the average ambient temperature during the daylight, and it is clear the solar flux (S) is a function of time (t) not function of (x), then we can write equation (8) with new parameter (θ) as follows:

$$\frac{dT}{dx} = K_{11}\,.\left(K_{14}\,.\,S - T + K_{15}\,.\,T_r + K_{16}\,.\,T_a \right) \qquad \text{..........}\ (9)$$

Letting, [$\theta = K_{14}\,.\,S - T + K_{15}\,.\,T_r + K_{16}\,.\,T_a$], so, [$\frac{d\theta}{dx} = -\frac{dT}{dx}$] and equation (9)

Becomes:

$$\frac{d\theta}{dx} + K_{11}\,.\,\theta = 0 \qquad \text{..............}\ (10)$$

Where:

- $K_{14} = \frac{K_{10}}{K_{11}}$ = Constant (Dimensionless)

- $\boldsymbol{K_{15}} = \dfrac{K_{12}}{K_{11}}$ = Constant (Dimensionless)
- $\boldsymbol{K_{16}} = \dfrac{K_{13}}{K_{11}}$ = Constant (Dimensionless)

Equation (10) is a First – Order differential equation with independent variables ($\boldsymbol{x}$), where ($\boldsymbol{S}$) in terms of weather conditions is a function of time ($\boldsymbol{t}$).

The solution of equation (10) is achieved by multiplying both sides of equation by the integrating factor ($e^{K_{11} . x}$) [ref. 7], then we will obtain:

$$\theta = C(x) . e^{-K_{11} . x}$$

To find the constant of integration [$\boldsymbol{C(x)}$], we should know the initial conditions which are { at ($\boldsymbol{x = 0}$), [$(\theta) = \theta_i$] }, substituting the initial condition in the above equation, we will have $(\theta_i = C)$, thus the solution of equation (10) becomes:

$$\theta = \theta_i \, . \, e^{-K_{11} . x}$$

Substituting, { $\theta = K_{14} . S - T + K_{15} . T_r + K_{16} . T_a$ and $\theta_i = K_{14} . S - T_i + K_{15} . T_r + K_{16} . T_a$ }, therefore the final solution for equation (10) becomes as the following:

$$T = T_i \, . \, e^{-K_{11} . x} + \left(1 - e^{-K_{11} . x}\right) . \left(K_{14} . S + K_{15} . T_r + K_{16} . T_a\right) \quad \ldots\ldots\ldots\ldots\ldots (11)$$

The incident solar flux absorbed by the plate ($\boldsymbol{S = \alpha . I_T}$) is the only parameter to be known in order to have complete solution for equation (11). Where:

- $\boldsymbol{\alpha}$ = the absorptivity of the absorber plate.
- $\boldsymbol{I_T}$ = the total solar flux incident on the absorber plate (W / m^2)

The total solar flux incident ($\boldsymbol{I_T}$) on the absorber facing the sun's tracking can measured directly and tabulated or it can be calculated and also tabulated. Because the measured data is not available, we calculated the total solar flux incident for a vertical wall facing to the south [see fig. (5) below] for ***Baghdad city*** using the following below formulas [ref. 8], we obtain the results for 21st of each month which are tabulated below in table No. (1).

The total flux can be calculated using the following formulas:

$$I_T = I_{b,n} . \cos\theta_i + \left[I_{d,h} . \left(\frac{1 + \cos\beta}{2}\right) + \rho . I_{t,h} . \left(\frac{1 + \cos\beta}{2}\right) \right]$$

$$I_{b,n} = I_o . \left(a_o + a_1 . e^{-k . \frac{1}{\cos\theta_z}} \right)$$

$$I_{d,h} = I_o \cdot \cos\theta_z \cdot \left[0.271 - 0.2939 \cdot \left(a_o + a_1 \cdot e^{-k \cdot \sec\theta_z} \right)\right]$$

$$a_o = 0.4237 - 0.00821\,(6 - E)^2$$

$$a_1 = 0.5055 + 0.00595\,(6.5 - E)^2$$

$$k = 0.2711 + 0.01858\,(2.5 - E)^2$$

$$I_o = 1367\left[1 + 0.034 \cdot \cos\left(\frac{360 . N}{365.25}\right)\right]$$

$$\theta_i = \cos\alpha \cdot \cos(\gamma - A)$$

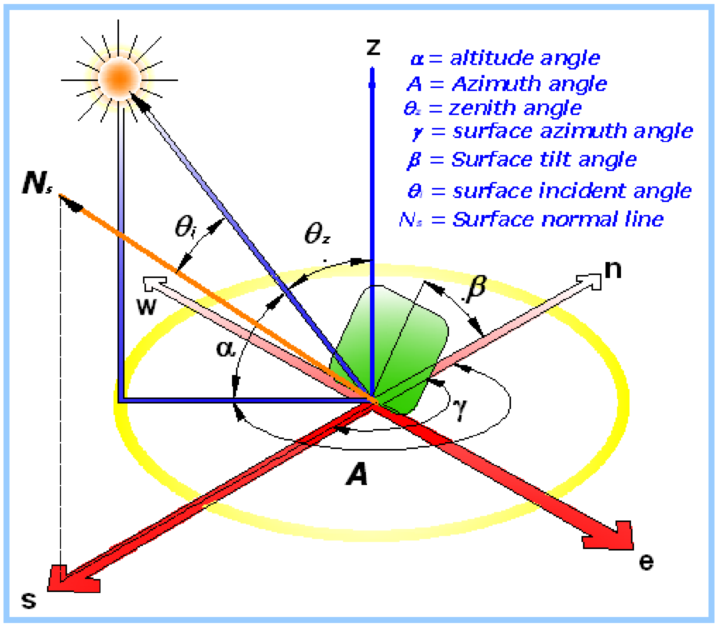

Figure (5): Schematic drawing of the Sun's position angles & surface angles.

Where**:**

- $I_{b,n}$ = Hourly beam radiation (W / m^2)
- $I_{d,h}$ = Hourly diffuse radiation (W / m^2)
- $I_{t,h}$ = $I_{b,n} + I_{d,h}$
- θ_i = Surface incident angle
- β = Surface tilt angle = 90^o for vertical walls
- ρ = The reflectance of the surrounding surface = 0.2 for a pale soil
- I_o = Solar radiation received outside the earth's atmosphere (W / m^2)
- θ_z = Solar zenith angle
- E = Local elevation in (km) = 0.035 km for Baghdad city
- N = The day number (starting at January 1)

- ***a_o, a_1 & k*** = Parameters described by the above equations
- α = The solar altitude angle
- γ = The surface azimuth angle
- ***A*** = The solar azimuth angle

The data in table (1) can be formulated to predict the solar flux radiation by using the simple Half-Sine model of clear and shiny day solar radiation [ref. (8)]. In this model the only input required is the time of sunrise, sunset and the peak solar flux radiation (I_{noon}) at noontime, the model state as follows:

$$I_T = I_{noon} \, . \, \sin\left[\frac{\pi * \left(t - t_{sunrise} \right)}{\left(t_{sunset} - t_{sunrise} \right)}\right]$$

Where:

- I_{noon} = The peak noontime Solar Irradiance (W / m^2)
- $t_{sunrise}$ = The Sunrise time (Hour)
- t_{sunset} = The Sunset time (Hour)
- t = Time (Hour)

Table (1): The Total solar flux incident on vertical wall facing south.

	Local Clock Time (LCT) (hour)																
	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
	Total solar flux incident on Vertical wall facing south																
21st of Jan.				105	253	470	629	727	766	745	664	523	321	105			
21st of Feb.				57	271	455	597	690	728	711	638	514	345	57			
21st of Mar.			0	96	250	400	521	599	629	607	536	421	275	119	0		
21st of Apr.		0	28	48	171	295	396	460	480	454	384	279	153	34	21	0	
21st of May		0	48	72	98	191	281	337	354	329	265	171	93	66	38	0	
21st of June		0	50	72	98	140	227	284	304	284	226	138	97	72	49	0	
21st of July		0	39	66	93	175	269	332	355	337	280	189	96	70	46	0	
21st of Aug.		0	22	58	166	292	395	462	485	462	396	293	168	58	23	0	
21st of Sep.		0	7	48	78	436	547	613	630	596	513	389	237	84	3	0	
21st of Oct.			62	164	365	531	649	716	726	682	583	435	245	76	62		
21st of Nov.				107	334	532	669	746	762	718	615	450	229	107			
21st of Dec.				123	262	485	642	736	766	735	641	482	259	123			

Letting the sunrise time ($t_{sunrise}$) to be the zero time, then ($\Delta t = t_{sunset} - t_{sunrise}$) represents the daylight time and the above equation for **(I_t)** becomes:

$$I_T = I_{noon} \cdot \sin\left(\frac{\pi * t}{\Delta t}\right) \quad \ldots\ldots\ldots\ldots\ldots (12)$$

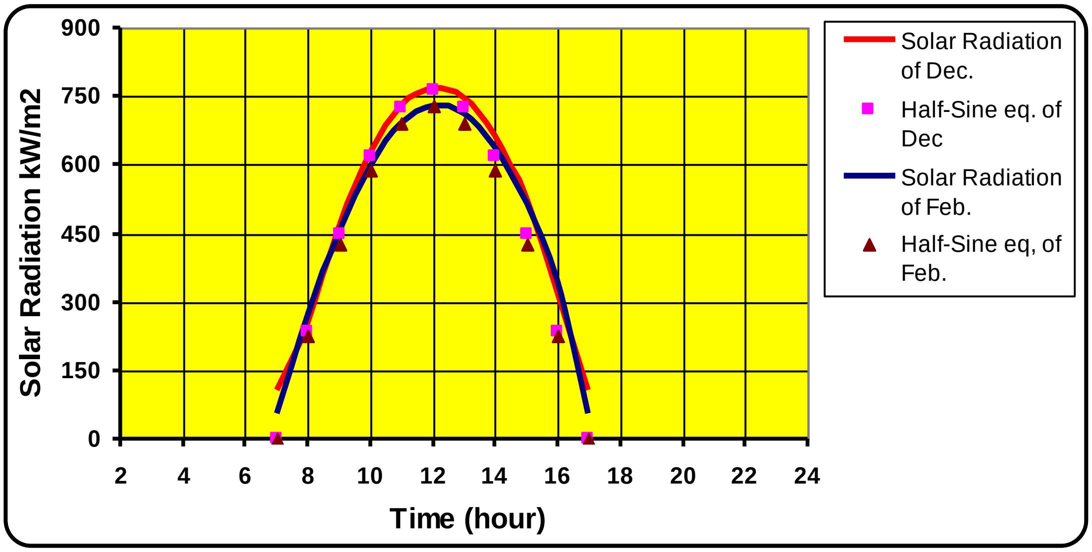

Fig. (6): The Estimated and Half-Sine eq. of Solar Radiation for 21st of Dec. & Feb.

The peak noontime Solar Irradiance **(I_{noon} (W / m^2))** for **21st** of some months will used to obtain (I_T) using equation (12) and shown in figure (**6**) above, which are drawn against the tabulated values in table (1) above, figure (6) shows a good coincident between the two readings which means that equation (12) can be used and substitute into equation (11) to become:

$$T = T_i \cdot e^{-K_{11} \cdot x} + \left(1 - e^{-K_{11} \cdot x}\right) \cdot \left(K_{14} \cdot \alpha \cdot I_{noon} \cdot \sin\left(\frac{\pi \cdot t}{\Delta t}\right) + K_{15} \cdot T_r + K_{16} \cdot T_a \right) \quad \ldots\ldots (13)$$

Theoretical application:

In this paragraph a sample of theoretical calculation will put into practice, to do that we should know all the design parameters mentioned below to find out the constants (***K_1, K_2, K_3,*** …..etc) used in equation (13):

Air flow rate: The amount of air required for ventilation depends on several factors such as: application, activity level, extent of cigarette smoking …. etc, the ventilation requirement increases with the occupancy. The quantities are laid down for 10 - 12 litre/sec. per person, and minimum rate of 10 litre/s per person for most non-domestic applications [ref. (9)], and 3 to 12 litres/sec. per person [ref. (10)]. A normal flow rate of preheated ventilation is (120 liter/sec per meter of collector length (L_1) [ref. (11)]. To facilitate the calculation, assume no. of persons is [(48 persons), it can be used any number of persons, it is just to show how to use eq. (13)] and required ventilation per person is (10 liter/sec), thus the total air flow is:

The ventilation required = 48 X 10 = 480 liter/sec.

Thus,

The collector length (L_1) = 480 / 120 = 4 meter

Air velocity: A certain air cavity between the absorber and the exterior wall is necessary to allow the heated air to travel up to fan intake, it should be in the range of (10 – 20 cm) and not to exceed (30 cm) with maximum air velocity of (3 m/sec) [ref. (3)]. Assume L = 0.1 m.

Air Temperature rise: The typical air temperature rise of the air flow for sunny days is normally in the range of (10 – 17 ^{o}C) over the ambient [ref. (3)]. Assume ($\Delta T = 10\ ^{o}C$) rise above ambient in order to get the average temperature of the air flow passing through the collector and then to find the air properties such as (density, specific heat, … etc.), which is equal to:

The average air temp. passing through the collector = T_{fi} + (ΔT / 2)

If the temperature rise is more (assume it 15 ^{o}C), it will affect the air properties and it may there is a little bit or no sense variation on the obtained results.

The panel color: The solar panel is a part of the building; the panel can be any dark color. The darker the better, as dark panel absorbs more of the sun's energy [the black is the best (solar absorptivity = 0.94) , followed by dark brown (solar absorptivity = 0.9)] [ref. (3)].

Heat transfer coefficient (h_{pa}): The convective heat transfer coefficient of absorber – ambient (h_{pa}) as a function of the wind velocity (V) can be expressed by the following formulas taking into account the surface conditions [ref. (12)]. So, in order to calculate (h_{pa}) and substitute it in equation (13), the wind velocity should be assumed. Assume (V = 1 m/sec), and for other value of wind velocity figure (11) shows the difference in the temp. rise for different velocities (V = 1 m/s & V = 2 m/s):

$$h = 5.47 + 3.95\,V \qquad (\text{ for smooth surfaces })$$

$$h = 6.16 + 4.19\,V \qquad (\text{ for rough surfaces })$$

Exterior wall construction: For simplicity, we will consider that the wall construction is of unique material which is brick of (***25 cm thickness***). The most building materials have a value of ($\varepsilon_\lambda = 0.9$) in the range of thermal radiation; they are described as thermally grey. Examples are brick and tiles, concrete, glass, white paper, most paints, slate [ref. (13)].

Average Ambient Temperature: The average ambient temperature during the daylight is calculated depending on the Meteorological Database for Baghdad city (***Year 2000***) for 21st of each month and tabulated in table (**2**) below:

Table (2): The Avg. daylight ambient temp. for 21st of each month.

Month	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
(T_a) Avg. ambient temp. (^oC)	11.83	15.00	23.73	31.00	32.30	34.40	41.56	39.36	31.30	27.60	16.72	13.15

6. Results & Discussion:

The analysis for this research shows the possibility to predict the temperature variation on the plain plate absorber, the temperature rise between the inlet and outlet air flow and showing the parameters that affect the temperature rise and variation of air passing the collector.

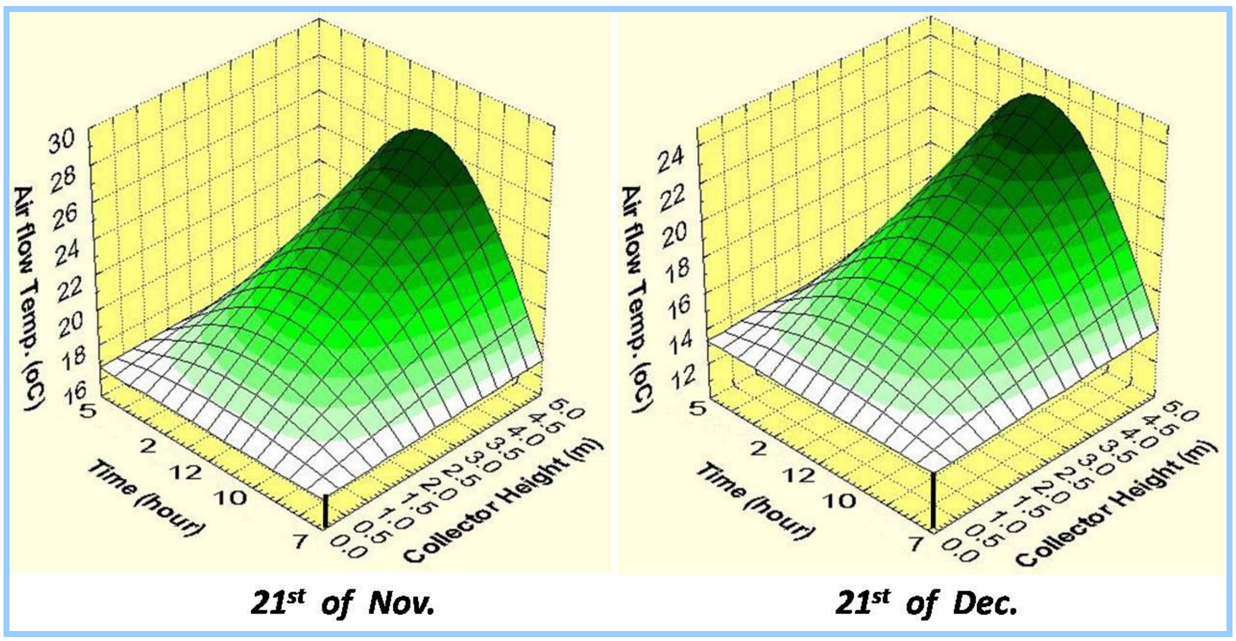

Figure (7): The temp. rise over the initial inlet temp. for 21st of Nov. & Dec.

In figure (7) above, the months of Nov. & Dec., were taken into consideration because these two months have approximately equal solar flux radiation incident on the wall which is clear in table (1) above showing that (I_{noon}) for both months are equal to (**762 W/m^2, 766 W/m^2**) respectively. It appears from figure (7), that the temperature rise over the initial inlet temperature [although it is different (**16.72 ^{o}C**) for **Nov.** and (**13.15 ^{o}C**) for **Dec**.)] is approximately equal to (**10 ^{o}C**) for both **21st** of **Nov.** & **21st** of **Dec**., we can conclude that the main factor affecting the temperature rise is the solar flux radiation incident on the wall, and the temperature rise aren't affected by the variant initial inlet temperature (T_{fi}) when all the other parameters kept the same.

The other parameter has a significant impact on the temperature rise is the Air flow rate or the Air velocity inside the air cavity along the depth (L). Figure (8) below shows that when the air velocity inside the air cavity is increased by reducing the air cavity (the depth L) from (10 cm) to (5 cm), then the temperature rise is increased because the heat transfer coefficient for both collector to air flow and wall to air flow are increased and vise versa.

Increasing the depth reduces the amount of air coming in contact with the back side of the collector and further reduces the heat transfer rate which lowers the solar collector efficiency.

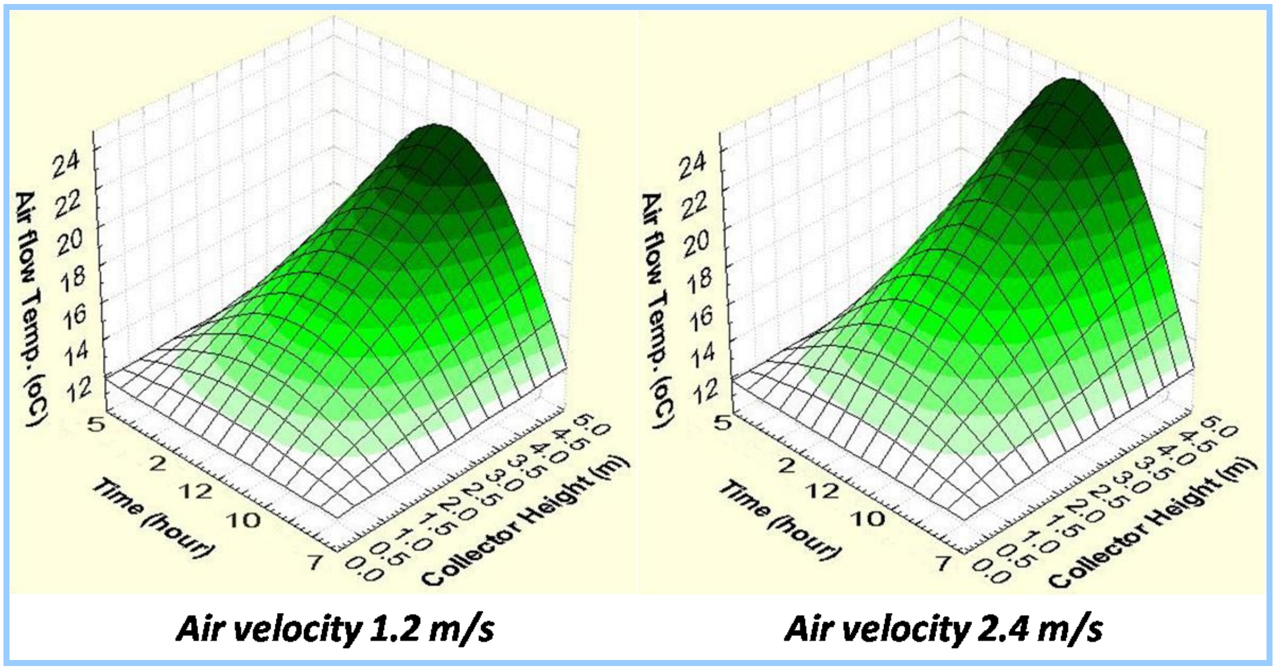

Fig. (8): The temp. rise over the initial inlet temp. for 21st of Jan. for different air flow velocities.

Figure (8), shows that the increment in the temperature rise when the air velocity is increased to (2.4 m/s) rather than (1.2 m/s) is about (2.3 ^{o}C), the increasing in air velocity will increase the temperature rise above initial inlet temperature but this increment should be balanced with the pressure drop across the collector which means a higher fan or blower should be used.

As well, the parameter (x) which is the height of the collector affects the temperature rise over the initial inlet temperature, as the height (x) is increased that means the contact time of air flow with absorber (collector plate) is increased and rising the air flow temperature.

Figure (9) above shows that when the height is reduced from (5 m) to (3 m), the temperature rise is reduced by about (3.5 ^oC). So increasing the collector height will increase the temperature rise as shown in fig. (10), but this fact will conflict by decreasing the collector efficiency because of increasing the heat loss to the ambient due to the high difference between the collector temperature and the ambient temperature.

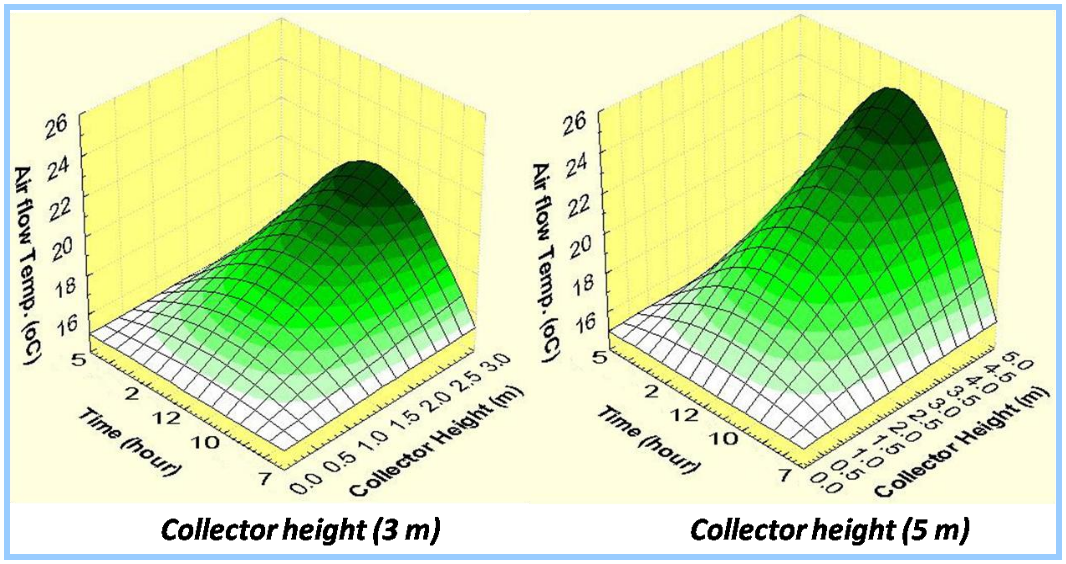

Fig. (9): The temp. rise over the initial inlet temp. for 21st of Feb. for different collector height.

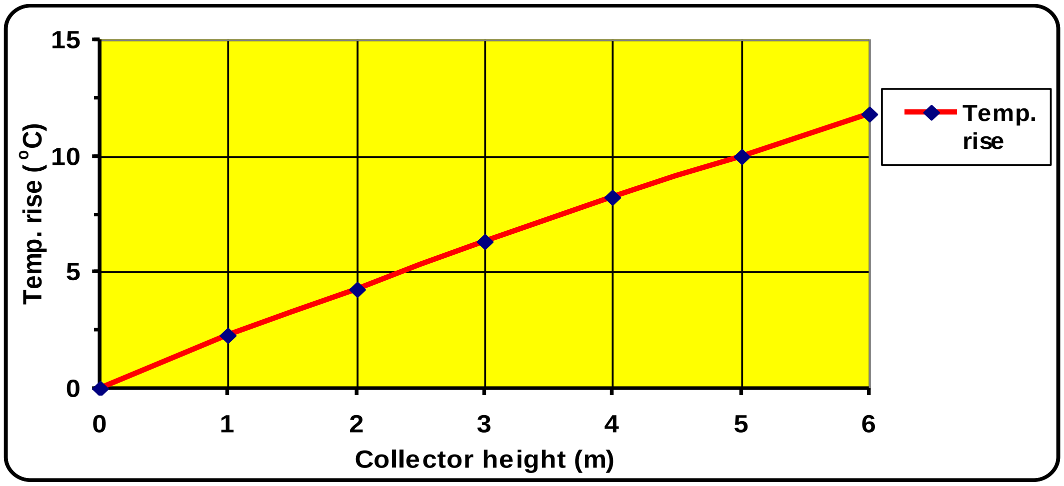

Fig. (10): Temp. rise over initial inlet temp. with collector's height for 21st of Nov.

The last affecting parameter is the wind velocity in which increasing the wind velocity will increase the heat transfer coefficient to ambient and then reducing the temperature rise and as well the collector's efficiency.

Figure (11) shows the temperature rise when the wind velocity changed from [1 m/sec to 2 m/sec], the decrease in temperature rise is about (2 ^oC)) which means that higher wind velocity will lower the temperature rise due to the increase of heat exchange between the collector and the ambient and as a result it will reduce the collector efficiency of the system.

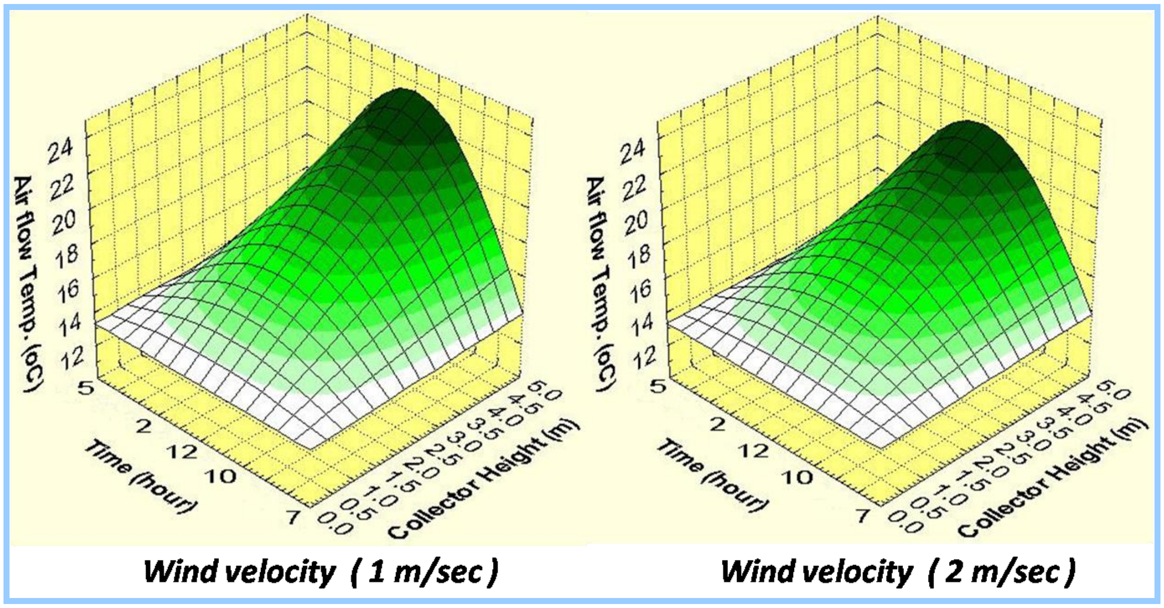

Fig. (11): The temp. rise over the initial inlet temp. for 21st of Dec. for different wind velocities.

As a conclusion, this type of collectors are suitable for cladding of exterior walls (which face the south) and due to required a large surface walls area it is used for larger building such as industrial, commercial buildings, maintaining garages.

It has also been used for drying agricultural crops as the required temperature rise must be kept relatively low to prevent damaging the crops.

The system will work best on sunny, still days, and to serve as a preheated to high temperature for industrial drying systems.

Unglazed plain plate collectors are a low cost, low performance system suited to buildings that otherwise may not have any form of heating such as warehouses, factories, gymnasia etc.

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