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Mathematical Modeling and Performance Analysis of V-Grooved single pass, V-Grooved Series and Parallel Flow Double Pass Solar Air Collectors

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Abstract: ***Solar collector is a device of mechanism to collect solar radiation energy and convert it into heat energy which in turn used as source of energy for different purposes. Solar air collectors are desired to increase the temperature of air that admit from the atmosphere passing over the solar collector absorber plate. Now day different mechanisms of configurations solar collectors are developed which can be appropriate for different applications as its energy and amount of working fluid requirement. In this project modeling and performance analysis of v-grooved single pass, parallel and series flow double pass solar air collector was conducted for the area around Wolaita sodo. In this comparison the useful heat energy gain of v-corrugated single pass is 788.01W, v-corrugated parallel flow double pass is 788.01W, and series flow double pass is 1575.43W assuming 40 ^oC average temperature difference.***

The output temperature to the mass flow rate of working fluid, output temperature to flow contact area, and overall efficiency of the collector to mass flow rate of the working fluid modeling result simulation was conducted by using math lab software for these v-grooved single pass, series and parallel flow double pass to their similarity and difference.

Key words: - *Output Temperature, Performance, V-Grooved Solar Air Collector, Working Fluid*

1. Introduction and Background

Since the 1970’s, residential solar technology has emerged as a result of the increasing cost of energy sources. Many attempts have been made thereafter to save domestic heat energy consumptions costs and today from simple to more sophisticated and modernized models of solar energy utilization mechanisms are developed. Basically solar energy conversion system is by using solar photovoltaic system and solar thermal energy conversion usually by using solar collector commonly used now a day.

Solar collector is a device of mechanism to collect solar radiation energy and convert it into heat energy which in turn used as sources of energy for different purposes. Solar radiation is energy in the form of electromagnetic radiation from the infrared (long) to the ultraviolet (short) wavelengths. The quantity of solar energy striking the Earth's surface (solar constant) averages about 1,000 watts per square meter under clear skies, depending upon weather conditions, location and orientation. Stationary category of solar collector classified in to two; concentric and non-concentric solar collectors. Concentric type solar collectors are; parabolic trough solar collector, parabolic dish type solar collector and central receiver with heliostats. Non-concentric type solar collectors are; different type and configured flat-plate solar collector, v-grooved, porous material integrated solar collectors, and evacuated tube solar collector. For non-concentric type solar collectors primary working fluid is air.

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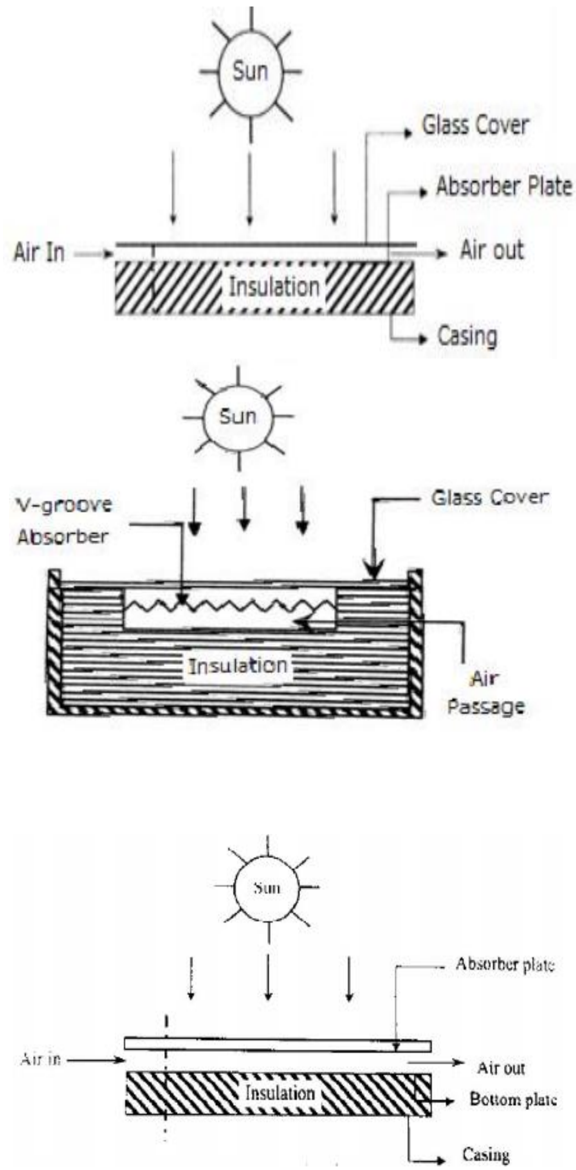
Using solar air collectors is to increase the temperature of air that admit from the atmosphere to the solar collector absorber plate which may exist if different structure and mechanism configurations which can be suited for different applications as its energy and working fluid requirement. As the air passes through over the absorber plate of the solar collector, the heat that is gained by the sun will be transferred to the air thereby raising its temperature. Thus, the heat energy contained in air have been used for several applications which include space heating, food drying, swimming pool heating, thermal industrial process, and solar drying of crop, vegetable, and fruits.

There are several designs configurations of solar air collectors which are currently used in different applications. The designs are based on the required amount of heat energy, and to increase the efficiency of the collector which can be achieved by increasing the heat energy gain by decreasing its losses through the bodies of the collector, increasing the air contact area, and also increasing the air flow length of the collector. Based on these concepts v-grooved and sometimes finned solar collector absorber are introduce and also making double pass of air is possible in extending the flow length of the collector. There are series and parallel flow double pass solar air collectors. In series flow double pass solar collectors same fluid passes twice below and above of collector absorber while different fluid pass below and above of collector absorber parallel way and directly admit to the dry chamber in case of parallel flow double pass air solar collectors.

However, efficiency of collectors in air heating is low due to low convective heat transfer coefficients between absorber and heat losses to the surrounding through the top cover. In order to improve the performance of the collector, flow arrangement could be changed to reduce the heat loss and the absorber shape could be changed to a v-groove and sometimes finned plate used in order to increase the absorbent area. Some designs of the air solar collectors are shown in the figure-1 below.



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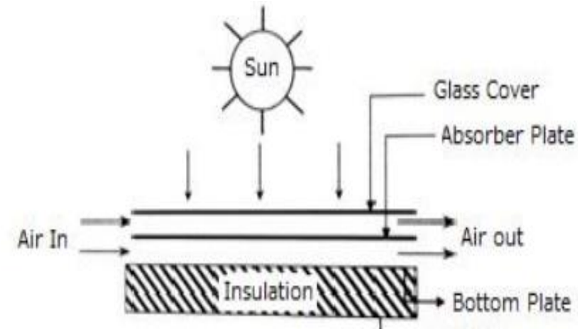

Figure-1: *Types of Solar Air Collectors*

The flat plate single pass flow air collector has efficiency of 41%, flat plate double pass flow air collector has efficiency of 55%, v-groove single pass flow collector has efficiency of 54% and v-groove double pass flow collector has efficiency of 56% [7]. This investigation indicates that that v-groove double pass flow solar air collector has the highest efficiency value due to the larger absorber air contacting area created by the v-grooves and air contact with the absorber from both sides of the grooved collector absorber.

V-Groove Collector

Since the heat transfer between the absorber plate and the flowing air is low on the flat plate collector, this configuration results in less air being heated. In order to improve the performance of the system, the v-groove plate was used as the absorber. In this type of collector, the solar radiation passes through the glass to the absorber, which in this case is a v-groove as shown in Figure-1 above. This type of collector can operate in single or double pass mass flow. According to the past research, a v-grooved absorber with double flow has 4-5% additional efficiency compared to that of the single mode.

In this project attempt was made to conduct mathematical modeling, and performance analysis v-grooved single pass, series and parallel flow double pass solar collectors. The main goal is to show the relation and difference of performance, output temperature, and working fluid mass flow rate. The main rationale of this research project is to show the performance relation of these v-grooved types solar collectors that is how identify collectors for the activities may require high output temperature, or high air flow rate.

2. Materials and Methods

A. Types and Configuration of solar collectors

The atmospheric air enters from the bottom of solar air collector and it moves up by density difference due to gaining heat energy from the collector absorber plate. Absorber plate painted high absorption material black paint. Solar air collector insulated from both sides and bottom to minimize heat energy loss and single glass cover considered to reduce back reflection of solar radiation from grooved absorber plate and to direct air flow. In all cases corrugated absorber palate v-angle of 60^0. In a single pass case, air flows below the corrugate plate once through the flow direction. In series flow double pass, air come to the collector absorber plate above the v-grooved absorber plate same fluid returns come to out from below of the absorber plate. Parallel flow double pass solar collector one has different fluid streams passes parallel above and below absorber plate same time as indicated the configuration differences indicated figures below.

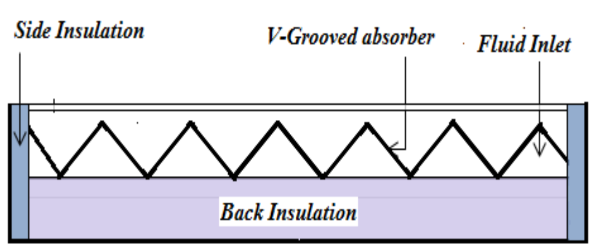

Figure-2: *V-grooved single pass*

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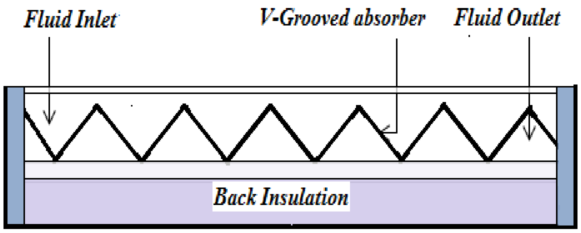

Figure-3: *V-grooved series flow double pass*

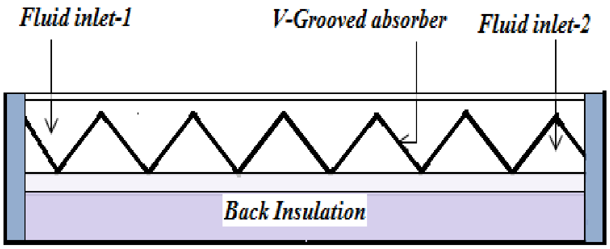

Figure-4: *V-grooved parallel flow double pass*

The amount of air flow rate depends on the natural convection flow speed, and flow cross sectional area normal to the flow streams of air. The output air temperature has also direct relation the length at which the fluid streams in contact with the absorber plate.

TABLE-1:

SPECIFICATIONS OF SOLAR COLLECTORS

Parts	***Specifications***
Total dimensions and type of collector	2.25m*1.155*(0.1m)2, v-grooved
Absorber material	Grade 1100 aluminum stainless steel
Absorber paint	Matt black, black chrome selective
Plate type	v-grooved (60^0) (reflection of solar insolation can be bounced on the other spikes) height of 50 mm with 20 grooves and *same gap* below absorber plate for double pass collectors parallel air passes from both side to the dry chamber
Back insulation	Fiberglass wool
Number of glazing and type	One , normal window glass (thickness 5mm)
Side insulating	Polystyrene, wood and silicon rubber
Sealing	Silicon rubber
Collector frame material	Cast iron (thickness 20 mm)

B. Assumptions made for modeling the solar dryer system

In order to simplify the collector modeling, the following assumptions were made.

- The collectors are at steady state thermal performance.
- The temperature drop through the glass cover, absorbing plate, and bottom plate are negligible.
- The heat flow through the back insulation is one-dimensional, that is the direction perpendicular to the air flow.
- The sky is assumed to be a blackbody for long-wavelength radiation at an equivalent sky temperature.
- The front and back surfaces losses are the same as the ambient temperature.
- Dust and dirt on the collectors are considered negligible in its effect of shading of the collector absorbing plates.
- The thermal inertia of collector components is neglected.

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- The operating temperatures of collector components and mean air temperatures in air channels are assumed to be uniform.
- Temperature of the air varies only in the flow direction.
- The channels are free of air leakage.
- The thermal losses through the collector backs are neglected

3. Modeling And Performance Analysis

A. Mathematical Modeling of the System Based on Energy Balances

Energy balance through the single pass collector

The thermal network for single-pass solar air collector is illustrated in Figure-5 below. The equations indicated step by step of energy balance will be used to calculate the thermal performance using the matrix method for its mathematical model.

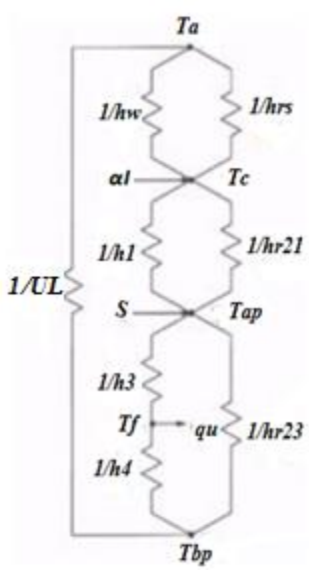

Figure-5: *Thermal Network of Single-Pass Solar Air Collector*

- ✓ Energy balance equation for the top cover:

$$\mathrm{S1} + (\mathrm{h1} + \mathrm{hr21})(\mathrm{Tap} - \mathrm{Tc}) = (\mathrm{hw} + \mathrm{hrs})(\mathrm{Tc} - \mathrm{Ta})$$

- ✓ Energy balance equation for the absorber plate; the amount of solar energy absorbed by the plate of solar collector:

$$\mathrm{S2} = (\mathrm{h1} + \mathrm{hr21})(\mathrm{Tap} - \mathrm{Tc}) + \mathrm{hr23}(\mathrm{Tap} - \mathrm{Tbp}) + \mathrm{h3}(\mathrm{Tap} - \mathrm{Tf})$$

- ✓ Energy balance equation for the fluid's second pass:

$$\mathrm{h3}(\mathrm{Tap} - \mathrm{Tf}) = \mathrm{qu} + \mathrm{h4}(\mathrm{Tf} - \mathrm{Tbp})$$

-where qu is useful heat (The amount of heat transferred working fluid; qu=Cp(T_{f0}-T_{fi}))

- ✓ Energy balance equation for the bottom plate:

$$\mathrm{hr23}(\mathrm{Tap} - \mathrm{Tbp}) + \mathrm{h4}(\mathrm{Tf} - \mathrm{Tbp}) = \mathrm{Ub}(\mathrm{Tbp} - \mathrm{Ta})$$

The solar radiation absorbed by the absorbing plate per unit area which is equal to the difference between the incident solar radiation and the optical loss, is calculated by:

S=αI

The convection heat transfer coefficient between the glass cover and the absorbing plate is:

h_1=Nu $_{\text{ap-c}}$ $\frac{K}{H'}$

H' −is the mean gap thickness between the cover and the absorbing H′ (m) is calculated by

H′ =Hc + 0.5Hg where Hg is the high of the v-groove absorber.

For flat plate and v-groove collector, Nusselt number for natural convection can be estimated by the following correlation:

$$\mathrm{Nu}_{ap-c} = 1 + 1.44\left[1 - \frac{1708(sin1.8\theta)^{1.6}}{Ra\cos\theta}\right]\left[1 - \frac{1708}{Ra\cos\theta}\right]^{+} + \left[\left(\frac{Ra\cos\theta}{5830}\right)^{173} - 1\right]^{+}$$

The equation above is valid for 0° ≤ θ ≤ 75°

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Ra is the Rayleigh number, which is defined as:

$$Ra = \frac{\rho^2 C_p g \beta (T_{ap} - T_c) H_c^3}{k\mu}$$

Radiation heat transfer coefficients between the glass cover and the absorbing plate and between the absorbing plate is represented as:

$$hr21 = \frac{\sigma(Tap^2 + Tc^2)(Tap + Tc)}{\frac{1}{\varepsilon ap} + \frac{1}{\varepsilon c} - 1}$$

The overall top loss coefficient is: *Ut=hw+hrs*, whereas the convection heat transfer coefficient from the glass cover due to wind is hw = 5.7 + 3.8Vw and the radiation heat transfer coefficient from the glass cover to sky referred to the ambient air temperature Ta can be obtained as follows:

$$hrs = (Tc + Ts)(TC^2 + Ts^2)\frac{(Tc - Ts)}{(Tc - Ta)}$$

The sky temperature Ts is estimated by;

Ts= $0.0552Ta^{1.5}$

Energy balance through the double pass collectors

The thermal network for double-pass solar air collector is illustrated in Figure-6 below. The equations indicated step by step of energy balance will be used to calculate the thermal performance using the matrix method for mathematical modeling.

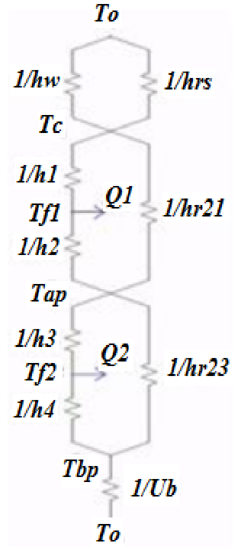

Figure-6: *Thermal Network of Double-Pass Solar Air Collector*

- ✓ Energy balance equation for the top cover:

$$S1 + hr21(Tap - Tc) + h1(Tf1 - Tc) = Ut(Tc - Ta)$$

- ✓ Energy balance equation for the fluid's first pass:
  $h2(Tap - Tf1) = Q1 + h1(Tf1 - Tc)$
  -where Q1 is The amount of heat transferred in the first pass fluid; Q1=mCp(T_{f0}-T_{fi})
- ✓ Energy balance equation for the absorber plate; the amount of solar energy absorbed by the plate of solar collector:

$$S2 = h2(Tap - Tf1) + hr21(Tap - Tc) + h3(Tap - Tf2) + hr23(Tap - Tbp)$$

- ✓ Energy balance equation for the fluid's second pass:
  $h3(Tap - Tf2) = h4(Tf2 - Tbp) + Q2;$
  -where Q_2 is The amount of heat transferred in the second pass fluid; Q1=mCp(T_{f0}-T_{fi})
  Energy balance equation for the bottom plate:

$$hr23(Tap - Tbp) + h4(Tf2 - Tbp) = Ub(Tbp - Ta)$$

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The solar radiation absorbed by the absorbing plate per unit area which is equal to the difference between the incident solar radiation and the optical loss, is calculated by:

$S_1 = \alpha I$

The convection heat transfer coefficient between the glass cover and the absorbing plate is:

$h_1 = Nu_{ap\text{-}c} \frac{K}{H'}$

H' —is the mean gap thickness between the cover and the absorbing H′ (m) is calculated by

H′ =Hc + 0.5Hg where Hg is the high of the v-groove absorber.

For flat plate and v-groove collector, Nusselt number for natural convection can be estimated by the following correlation:

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The equation above is valid for $0° \leq \theta \leq 75°$.

Ra is the Rayleigh number, which is defined as:

$$Ra = \frac{\rho^2 C_p g \beta (T_{ap} - T_c) H_c^3}{k\mu}$$

Radiation heat transfer coefficients between the glass cover and the absorbing plate and
between the absorbing plate is represented as:

$$hr21 = \frac{\sigma(Tap^2 + Tc^2)(Tap + Tc)}{\frac{1}{\varepsilon ap} + \frac{1}{\varepsilon c} - 1}$$

The overall top loss coefficient is: *Ut=hw+hrs,* whereas the convection heat transfer coefficient from the glass cover due to wind is hw = 5.7 + 3.8Vw and the radiation heat transfer coefficient from the glass cover to sky referred to the ambient air temperature Ta can be obtained as follows:

$$hrs = (Tc + Ts)(TC^2 + Ts^2)\frac{(Tc - Ts)}{(Tc - Ta)}$$

The sky temperature Ts is estimated by Ts = $0.0552 Ta^{1.5}$

The incident solar radiation absorbed by the absorbing plate is

$S2 \approx 0.97 \tau_C \alpha_{ap} I$

The Nusselt number for V-groove collector is estimated by the following correlation

When Re< 2800

$Nu_{ap\text{-}f} = 2.821 + 0.126Re \frac{Hg}{L}$

When $2800 \leq Re \leq 10^4$

$Nu_{ap\text{-}f} = 1.9 \times 10^{-6} Re^{1.79} + 225 \frac{Hg}{L}$

When $10^4 \leq Re \leq 10^5$

$Nu_{ap\text{-}f} = 0.0302 Re^{0.74} + 0.242 Re^{0.74} \frac{Hg}{L}$

Where Reynolds number for a v-groove plate absorber is: $Re = \frac{Dh\rho Uf}{\mu}$

Hydraulic diameter of the airflow channel formed by the absorbing plate and the bottom
plate is Dh (m). And the hydraulic diameter for a v-groove absorber is given as:

Dh=(2/3)*Hg

The radiation heat transfer coefficients between the glass cover and the bottom plate are predicted by:

$$hr23 = \frac{\sigma(Tap^2 + Tbp^2)(Tap + Tbp)}{\frac{1}{\varepsilon ap} + \frac{1}{\varepsilon bp} - 1}$$

The convection heat transfer coefficients for the fluid moving on the absorbing plate and on the bottom plate are calculated by:

$$h3 = Nu\ ap - f\ \frac{K}{Dh}$$

The conductive heat transfer coefficient between the second pass fluid and the bottom plate is assumed to be equal, therefore:

$h_3 = h_4$

The heat gain equation is given as: $Qu = \gamma(T_{fo} - T_{fi})$,

where, $\gamma = Cp\dot{m}a$

The conduction heat transfer coefficient across the insulation from bottom and sides of the collector is estimated by:

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$$Ub = \frac{Ki}{\Delta t}$$

Finally, the collector efficiency can be determined by:

$$\eta = \frac{mCp(T\dot{o} - Tf\iota)}{I}$$

The following empirical correlations can be used to estimate the specific property of air for T from 280 K to 470 K:

$\rho = 3.9147 - 0.016082T + 2.9013 \times 10^{-5}T^2 - 1.9407 \times 10^{-8}T^3$

Thermal conductivity: $k = (0.0015215 + 0.097459T - 3.3322 \times 10^{-5}T^2) \times 10^{-3}$

$Cp \cong 1.005$ kJ/kg K can be assumed.

TABLE-2:

VALUES OF THE CONSTANTS

Parameters	*Value*	*Unit*
I	600	W/m^2
Θ	30	degrees
$Ta=Tfi$	300	K
Ki	0.025	W/mK
ϵap	0.94	---
ϵbp	0.92	---
ϵc	0.94	---
αap	0.06	---
αc	0.95	---
τc	0.90	---
Vw	4.32	m/s

B. Performance Analysis of the Collectors

Thermal analysis

The thermal performance of a solar collector depends on many parameters such as:

- ✓ Ambient conditions (Ambient temperature, wind speed, solar radiation)
- ✓ Geometry of collector
- ✓ Characteristics of working fluid (Cp, K, ρ)
- ✓ Inlet temperature of fluid (T_{in})
- ✓ Flow rate ($\dot{m}$)
- ✓ Choice of the absorber material
- ✓ Location of the construction: (Inclination angle, direction)

The energy balance on the absorber of solar collector is obtained by equating the total heat gained from sun to the total heat lost by the heat absorber of the solar collector. Therefore:

$$IA_c = Q_U + Q_{cond} + Q_{conv} + Q_R + Q_P$$

The three heat loss terms, Qcond, Qconv and Qradia are usually combined into one-term (Q_L) that is:

$$Q_L = Q_{cond} + Q_{conv} + Q_R + Q_P$$

Then, the useful energy will be available on the absorber of the solar collector is:

$$Qu = IAc - Q_L$$

Energy Balance Equation

In steady state, the performance of a solar collector can be described by the useful gain energy from the collector, *Qu*, which is defined as the difference between the absorbed solar radiation and the thermal loss or the useful energy output of a collector:

$$Q_U = A_P S - Ac\ UL\ (Ta_p - T_a)$$

Where *Ac* and *AP* are the gross and aperture area of the collector, respectively

The first term is the absorbed solar energy and the second term represents the heat loss from the collector. The solar radiation absorbed by a collector per unit area of absorber *S* can be calculated using the optical properties of covers and a plate. The thermal energy loss from the collector to the surroundings can be represented as the product of a heat loss coefficient U_L times the difference between the mean absorber plate temperature *Tap* and the ambient temperature *Ta*.

Gross collector area *Ac* is defined as the total area occupied by a collector and the aperture collector area *Ap* is the transparent frontal area.

ASHRAE Standard employs the gross area as a reference collector area in the definition of thermal efficiency of the collector. The useful gain from the collector based on the gross collector area becomes:

Qu = A_C FR(Sc -U_L (Tfi - Ta))

Where *Sc* is the absorbed solar radiation per unit area based on the gross collector area, defined as:

Sc = S (Ap/Ac)

Since the radiation absorption and heat loss at the absorber plate is considered based on the aperture area in this study, it is convenient to make the aperture collector area the reference collector area of the useful gain.

Collector Heat Removal Factor:

The collector heat removal factor, F_R, is the ratio of the actual useful energy gain of a collector to the maximum possible useful gain if the whole collector surface were at the fluid inlet temperature. It is defined as:

$$FR = \frac{\mathrm{mCp\ (Tfo\ - Tfi\ )}}{\mathrm{Ap\ (\ S - UL'(Tfi\ - \ Ta)\ )}}$$

Where the aperture area *Ap* is used as a reference area for the useful gain from the collector.

$$FR = \frac{\mathrm{mCp}}{\mathrm{AcUL}}\left(1 - \frac{\mathrm{S/UL(Tfo\ - \ Ta)}}{\mathrm{S/UL(Tfi\ - \ Ta)}}\right)$$

Physically the collector heat removal factor is equivalent to the effectiveness of a conventional heat exchanger. By introducing the collector heat removal factor and the modified overall heat transfer coefficient, the actual useful energy gain *Qu* can be represented as:

Qu = ApF_R [S – UL(Ti − Ta)]

Using Equations, the useful energy gain can be calculated as a function of the inlet fluid temperature not the mean plate temperature.

Collector Thermal Efficiency

The efficiency of solar collector is defined as the ratio of the useful energy gain to the incident solar energy. Efficiency of the solar collector is given by

$$\eta = \frac{\mathrm{Qu}}{\mathrm{IAc}} = \frac{\mathrm{mCp(Tfo - Tfi)}}{\mathrm{IAc}}$$

The efficiency of a solar collector is defined as the ratio of the amount of useful heat collected to the total amount of solar radiation striking the collector surface during any period of time. Useful heat collected for an air-type solar collector can be expressed as:

Qu = ṁaCp(Tfo - Tfi)

The calculation of the overall loss coefficient (U_L) is based on simulation convection and re radiation losses from the absorber plate to the atmosphere.

The radiation absorbed flux by unit area of the absorber plate (S) is defined as:
S = (τcαc)I = 0.90*0.95*600=513 W/m^2

Collector overall loss coefficient

The solar thermal efficiency depends essentially on thermal losses from outer surfaces of the collector. To attain higher efficiency of the solar collector, the losses of it must be minimized for the practically application.

The top heat loss coefficient, Ut:

The top heat loss coefficient Ut determined by using the empherical formula indicated below which is in joint term of radiation heat loss, wind loss and convection losses:

$$\mathrm{Ut}=\left[\frac{N}{\frac{C}{Tp}+(\frac{Tp-Ta}{N+F})e}\frac{1}{hw}\right]^{-1} + \frac{\sigma(Tp^2-Ta^2)(Tp-Ta)}{\frac{1}{d}+\frac{2N+f-N}{\epsilon c}-N}$$

Where:

C = $\frac{204.429(\cos\theta)^{0.252}}{L^{0.24}}$

d = ϵc+0.0425N (1-ϵc)

f = $\left(\frac{9}{hw} - \frac{30}{hw^2}\right)\left(\frac{Ta}{316.9}\right)(1 + 0.091N)$

e = 0.252

hw =5.7+3.8*Vw, where Vw-wind speed (in Wolaita sodo town V=4.32 m/se)

hw =21.66

Substituting all the values Ut=2.725 W/m^2

The bottom heat loss coefficient, Ub:

The bottom heat loss coefficient Ub is mainly the conduction heat loss through the thickness of the bottom insulation neglecting the radiation loss.

Ub = $\frac{Ki}{\Delta t}$

Where ki conduction coefficient, Δt insulation thickness from bottom of the collector

Ub = $\frac{0.0252}{0.050} = 0.504$W/$m^2$

Heat loss coefficient from sides of the collector, Ue:

The side or collector edge heat loss coefficient Ue is similar to that of the bottom heat loss coefficient is mainly the conduction heat loss through the thickness of the edges of the collector's insulation neglecting the radiation loss.

Ue = $\frac{Ke}{\Delta t} = \frac{0.0252}{0.0250} = 1.008$ (loss coefficient for both sides of the collector 2.016 W/m^2)

Therefore, the overall heat loss coefficient U_L which is the sum of all loss coefficients from the collector:

U_L = Ut + Ub + Ue

= 2.725 W/m^2+ 0.504W/m^2 + 2.016 W/m^2

=5.245 W/m^2

The useful heat gained by the collector

Qu=A_P (S - U_L(Tp - Ta))

Tp-the mean absorber plate temperature K

Ta -ambient temperature K

AP-absorber plate area m^2

The overall thermal efficiency of the collector which is the ratio of useful heat energy gained by the collector absorber to the total solar radiation come to the collector absorber thus which is estimated as:

η = Qu/I*Ap

The energy gain by the working fluid of the system which is in turn equal to the energy gain to the collector absorber, thus useful energy is estimated as:

Qu = $\dot{m}a$ Cp (Tfo - Tfi)

Where:

- Tfo-fluid output temperature in K from collector absorber to dry chamber
- Tfi-fluid inlet temperature in K to the collector absorber
- Cp-specific heat capacity
- $\dot{m}a$-mass flow rate of air through collector absorber

4. Results and Discussions

In this research project the type of solar collector which modeling and performance analysis have been conducted in v-corrugated single pass, v-corrugated parallel flow double pass, and series flow double pass solar air collectors. In this comparison the useful heat energy gain of v-corrugated single pass is 788.01 W, v-corrugated parallel flow double pass is 788.01 W, and series flow double pass is 1575.43 W assuming

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40 ^{o}C average temperature difference. This shows that the useful heat gain of series flow double pass is twice of that the two others due to high areal contact of the working fluid with the absorber plate of the solar collector. But, in the case of parallel flow double pass collector, the mass flow rate of fluid is twice of that of single pass and series flow double pass solar collectors.

The output temperature to the air mass flow rate graph simulation was made taking the inlet air temperature 300K and the graph shows that the output temperatures of the series flow double pass greater than single pass and which is in turn greater than that of parallel flow double pass collector in same air mass flow rate.

In other hand the overall efficiency of the solar air collectors to mass flow rate graph relation was plotted emphasizes the linear relation of collectors efficiency and mass flow rate of the system. As the air mass flow rate increases the efficiency of the solar collectors also increase. Similarly based on the useful solar heat energy gain of the collector, the efficiency of the collector, the output working fluid temperature rather than mean absorber temperature has direct relation with the aperture air contact area of the collector.

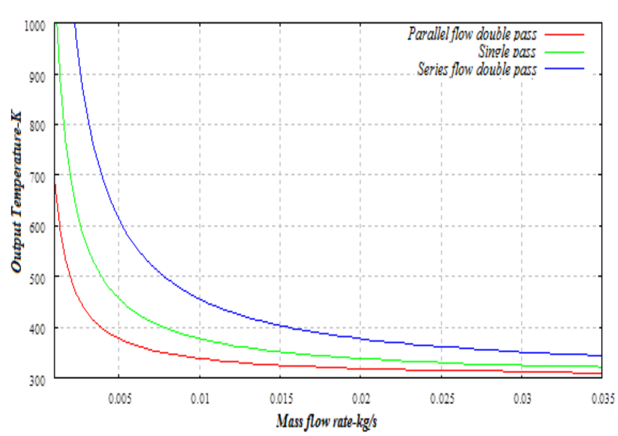

Figure-7: *Output temperature to air mass flow rate*

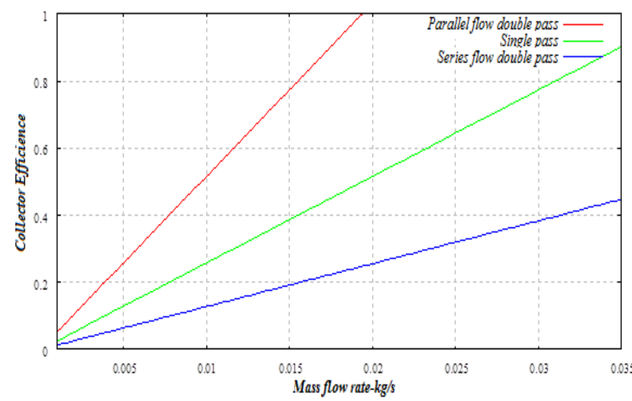

Figure-8: *Collector overall efficiency to air mass flow rate*

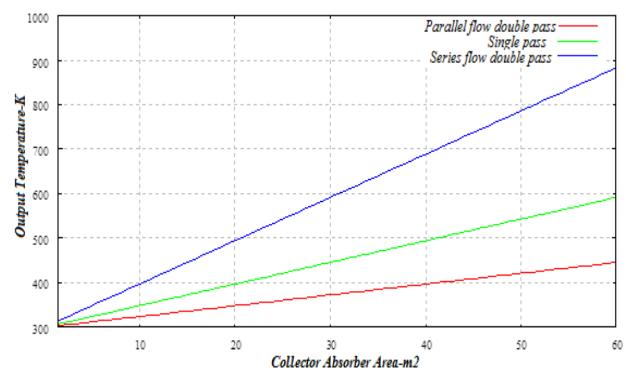

Figure-9: *Output temperature to collector absorber area*

5. Conclusion and Recommendation

There are several designs configurations of solar air collectors which are currently used in different applications. The designs are based on the required amount of heat energy, and to increase the efficiency of the collector which can be achieved by increasing the heat energy gain by decreasing its losses through the bodies of the collector, increasing the air contact area, and also increasing the air flow length of the collector.

In this project, modeling and performance analysis of v-corrugated single pass, parallel and series flow double pass solar air collector was conducted to show their efficiency relationship.

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However, modeling and performance analysis carried out analytically and it should be further witnessed practically developing parts prototype and conducting experimental investigation.

Nomenclature

I - Solar insolation rate incident on the glass cover (W/m^2)

Tc - Transmissivity of solar radiation of the glass cover

αap - Absorptivity of solar radiation of the absorbing plate

S - Solar radiation absorbed by glass cover and absorber plate (W/m^2)

Qu - Useful energy gain (W)

Ts - Sky temperature (k)

Tc - Mean temperatures on the glass cover (K)

Tap - Mean temperature on the absorbing plate (K)

Tf - Mean air temperature (K)

Tbp - Mean temperature on the bottom plate (K)

Ta - Ambient air temperature (K)

Tfi - Inlet temperature of collector (k)

Tfo - Outlet temperature of collector (k)

Cp - Specific heat of air (J/kgK)

h_1 - Thermal losses to the glass cover by natural convection

***h*r21** - Thermal losses to the glass cover by thermal radiation (W/m^2K)

hr23 - Thermal losses to the glass cover by the bottom plate by thermal radiation (W/m^2K)

h4 - Convection heat transfer coefficient of fluid on the bottom plate (W/m^2K)

hw - Convection heat transfer coefficient from the glass cover due to wind (W/m^2K)

hrs - Radiation heat transfer coefficient from the glass cover to sky (W/m^2K)

h_3 - Convection heat transfer coefficients for the fluid (W/m^2K)

σ - Stefan-Boltzmann constant (5.67×10^{-8} $W/m^2 K^4$)

εc - Emissivity of thermal radiation of the glass cover

εap - Emissivity of thermal radiation of the absorbing plate

εbp - Emissivity of thermal radiation of the bottom plate

References

[1]. Sarsavadia PN, Sawhney RL, Pangavhane DR, Singh SP, "*Drying behavior of brined onion slices*". J Food Eng. 1999;40: 219–226. doi: 10.1016/S0260-8774(99)00058-8

[2]. McLean KA., "*Drying and storage of combinable crops*". Suffolk Farm Press, 1980.

[3]. Ekechukwu OV., "*Experimental studies of integral-type natural-circulation solar-energy tropical crop dryers*". Kingdom, 1987.

[4]. Arinze EA, Schoenau G, Bigsby FW., "*Solar-energy absorption properties of some agricultural products*". ASAE paper, no. 79-3071, 1979.

[5]. Brooker DB, Bakker-Arkema FW, Hall CW., "*Drying cereal grains*". Westport AVI, 1974.

[6]. O.V. Ekechukwu, B. Norton, "*Review of solar-energy drying systems II: an overview of solar drying technology*". Energy Research Centre, University of Nigeria, Nsukka, Nigeria 1997

[7]. M.A. Karim and Z.M. Amin, "*Mathematical Modeling and Performance Analysis of Different Solar Air Collectors*". *IIUM Engineering Journal, Vol. 16, No. 2, 2015*

[8]. Ong KS. (1995), "*Thermal performance of solar air heaters: Mathematical model and solution procedure*". Solar Energy, 55:93-109.

[9]. Mudafer kareem Abdullah and Ahmed Abed Gatea, "*Performance evaluation of a v-groove solar air collector for drying maize (Zea mays) in lraq*". January, 2011

[10]. Hall CW., "*Drying and storage of agricultural crops*". Westport AVI, 1980.

[11]. Bashria A et al., *Analysis of Single and Double Passes V-Grooves Solar Collector With and Without Porous Media*, Volume 1, 2007

[12]. Mulugeta Tadesse, "*Design, Modeling, and Performance Analysis of Parallel Flow Double Pass V Groove Solar Air Collector for Preservation of Onion in University Students' Cafeteria: Wolaita Sodo University", found online at http://www.ijrdet.com/Volume6Issue6.php*