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Simulation Of Annular Gap Effect On Performance Of Solar Parabolic-Trough Collector Model TE38 In Bauchi

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ABSTRACT: *Annular gaps of 0.010m, 0.015m, 0.020m, 0.025m, 0.030m, 0.035m, 0.040m, 0.045m, 0.050m and 0.055m are used to simulate the performance of the solar parabolic-trough collector, while maintaining the same geometric dimension of the reflector, the same absorber-tube design features, as well as the same meteorological and radiation data, employing developed thermal energy equations. The higher the annular gap between the absorber-tube and the enveloping glass-cover, with corresponding increase in the diameter of the enveloping glass-cover, the more is the total heat loss of the system and therefore, the lower is the thermal efficiency of the system.*

KEYWORDS: *Absorber-tube, Enveloping-glass-cover, Annular-gap, Transmissivity, Reflectivity, Absorptivity and Emissivity.*

I. INTRODUCTION

The sun is an enormous source of heat and the furnace of all solar systems. The sun radiates, through continuous process of thermonuclear fusion, approximately 83.3 million billion-kilowatt hours (8.33×10^{25} kWh) of energy into space every day [1]. While the earth's daily receipt of a minute fraction of this energy depends on its distance from the sun, as well as sunspot activity on the solar surface, it always amounts to very close to 4.14 million billion-kilowatt hours (4.14×10^5 kWh) each day [2]. Based on recent measurements in space, the currently accepted value of the solar constant is 1377 W/m^2 [3]. The extraterrestrial radiation striking the earth varies throughout the year primarily because of the change in the sun-earth distance, due to sunspots, flares and other random activity on the surface of the sun. The earth's outer atmosphere intercepts about one two-billionth of the energy generated by the sun, or about 1500 quadrillion (1.5 x 10 E18) kWh/year. The sun generates an enormous amount of energy - approximately 1.1 x 10 E20 kilowatt-hours every second. A kilowatt-hour is the amount of energy needed to power a 100 Watt light bulb for ten hours. The earth's outer atmosphere intercepts about one two-billionth of the energy generated by the sun, or about 1500 quadrillion (1.5 x 10 E18) kilowatt-hours per year [4]. Because of reflection, scattering, and absorption by gases and aerosols in the atmosphere, as schematically represented in Fig. 1, however, only 47% of this, or approximately 700 quadrillion (7 x 10 E17) kilowatt-hours, reaches the surface of the earth.

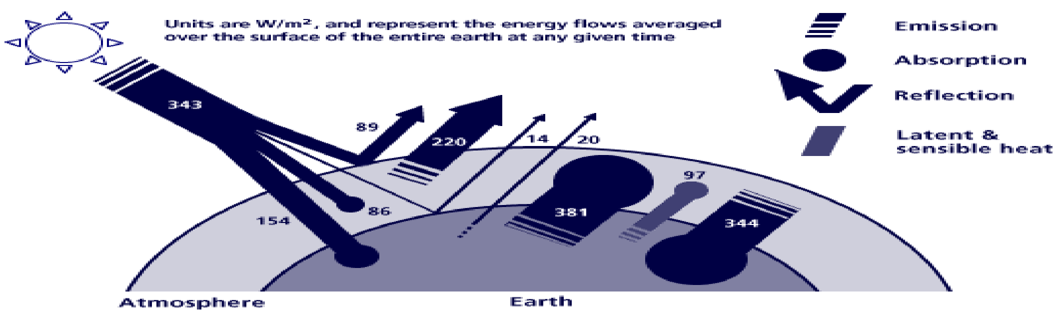

Fig. 1: The Sun radiation reaching the surface of the earth

At the earth's surface, the sum of the incident solar radiation from all directions is called the global insolation. The portion that comes directly from the sun without a change in direction (not scattered) is called the beam or direct insolation. Its value ranges from about 90% of the global insolation on extremely clear day to practically zero on an overcast day. The diffuse or non-direct insolation from all directions except directly from the sun makes up the remainder of the global insolation [1]. Therefore, there are two different components of solar radiation that strike any solar collector surface; these are direct and diffuse (deflected by atmospheric scattering effects and reflected rebound from surrounding terrain) as shown in Fig 2.Harnessing the sun's light and heat is a clean, simple, and natural way to provide all forms of energy we need. The sun's heat can be collected in a variety of different ways. It can be concentrated by parabolic mirrors to provide heat at up to several thousands degrees Celsius. This heat can be used either for heating purposes or to generate electricity. Solar thermal electric power plants generate heat by using lenses and reflectors to concentrate the sun's energy. Because the heat can be stored, these plants can generate power when it is needed, day or night, rain or shine.

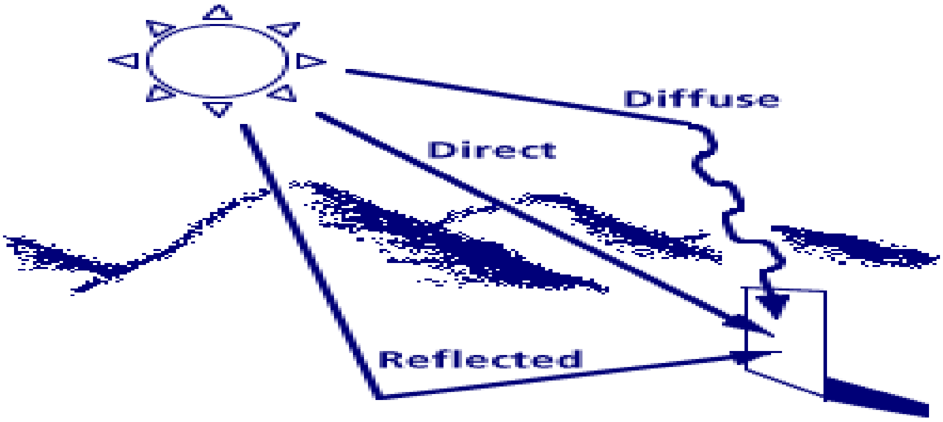

Fig. 2: Solar global radiation

Solar parabolic trough collector (SPTC) is a reflecting surface made of highly polished stainless steel reflector, fixed on a parabolic contour, supported by steel framework and mounted on reflector's support structure as shown in Fig. 3. The reflector is used to concentrate sunlight onto a receiver tube that is positioned along the focal line of the parabolic-trough. A transparent glass-tube envelops the receiver tube to reduce heat loss. Solar parabolic-troughs often use single-axis or dual-axis tracking [5]. In rare instances, they may be stationary. Temperatures at the receiver can reach 400 °C and produce steam for generating electricity [6].

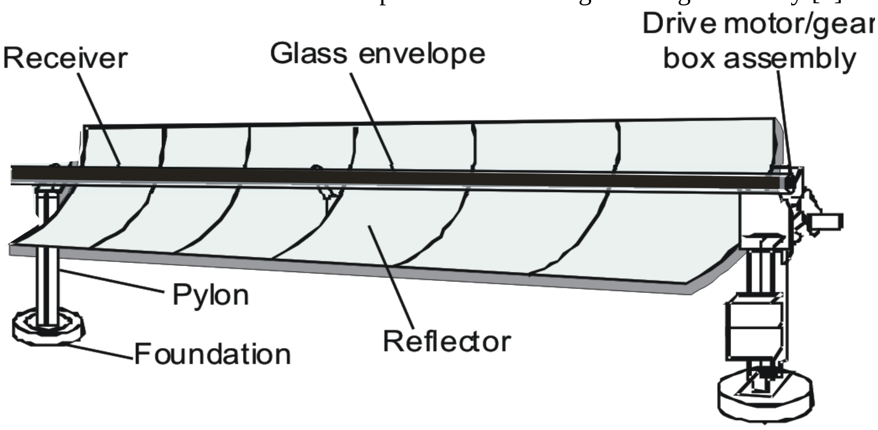

Fig. 3: Solar Parabolic-trough Collector

The mathematical property of a parabola makes it advantageous for steam production using the parabolic trough concentrator [7]. Rays of light that reaches the reflecting surface are reflected onto the focal line of the trough where the receiver assembly which consists of a transparent enveloping glass-tube and an absorber-tube is placed. Working fluid flow inside the absorber-tube and carries away the heat either trapped to the load directly as in active systems or initially stored as in passive systems.

II. METHODS

Energy balance equations for SPTC have been developed [8]. The equations considered the heat-energy-gain, the heat-energy-loss and the heat-energy-transfer between the components, i.e. the reflecting surface, the glass-cover and the absorber-tube, the thermal properties of the materials of the components and geometric dimensions of the SPTC. The equation for the enveloping glass-cover heat energy transfer is given as follows;

$$\alpha_g 2RL\left((I_{beam}R_b)+I_{diff}\right) + \rho_c\alpha_g\left(\frac{(W-D)L}{\pi}\right)(I_{beam}R_b) + \frac{A_t\sigma(T_t^4-T_g^4)}{\frac{1}{\varepsilon_t}+\frac{A_t}{A_g}\left(\frac{1}{\varepsilon_g}-1\right)}$$

$$-\sigma\varepsilon_g A_g(T_g^4-T_{sky}^4) - A_g h(T_g-T_{surr}) = m_g Cp_g\left(\frac{dT_g}{dt}\right) \quad \text{... 1}$$

The equation for the absorber-tube heat energy transfer is given as follows;

$$\alpha_t\tau_g\left(\frac{2RL}{\pi}\right)\left[(I_{beam}R_b)+I_{diff}\right] + \tau_g\rho_c\alpha_t\left[\frac{(W-D)\times L}{\pi^2}\right]\left[I_{beam}R_b\right]$$

$$-\frac{A_t\sigma(T_t^4-T_g^4)}{\frac{1}{\varepsilon_t}+\frac{A_t}{A_{g,}}\left(\frac{1}{\varepsilon_g}-1\right)} - \frac{A_{tin}(T_t-T_f)}{\left[\frac{1}{h_f}+\frac{\ln\left(\frac{r}{r_1}\right)A_{tin}}{2\pi KL}\right]} = m_t Cp_t\left(\frac{dT_t}{dt}\right) \quad \text{... 2}$$

The equation for the fluid heat transfer is given as follows;

$$\frac{A_{tin}(T_t-T_f)}{\left[\frac{1}{h_f}+\frac{\ln\left(\frac{r}{r_1}\right)A_{tin}}{2\pi KL}\right]} - \frac{2A_c(T_f-T_{surr})}{\frac{1}{h_{surr}}+\frac{1}{h_f}} = m_f Cp_f\frac{dT_f}{dt} \quad \text{... 3}$$

Employing these equations, a Mat Lab program was used to simulate the effect of annular gap which is the evacuated portion shown in Fig. 4.

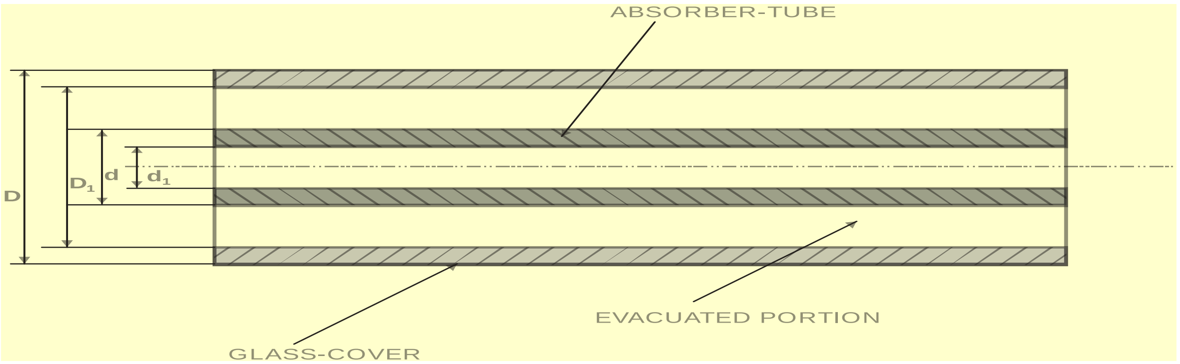

Fig. 4: Receiver assembly dimensions.

The absorber-tube is made of copper tube, coated with black paint. The outer diameter of the absorber tube, d, is given as 0.013 m. With a thickness of the absorber-tube, $x_t = 0.0015$ the inner diameter of the absorber-tube, d_1, can be computed as follows:

$$d_1 = d - 2.x_t \quad \text{... 4}$$

With varying annular gap, (a), the inner diameter of the concentric transparent glass-cover, D_1, can be computed as follows:

$$D_1 = d + 2a \qquad \text{... 5}$$

It therefore follows that the outer diameter of the enveloping glass-cover, (D), with a thickness, (x_g), of 0.002 m , can be computed as follows:

$$D = D_1 + 2x_g \qquad \text{... 6}$$

Soda lime's transmittance, (τ_g) is 0.83 [9]. The absorptance (α_g) is 0.14. Emittance value of the glass-cover (ε_g) is 0.84 [8]. The Absorptivity of the copper is 0.70, and that of black paint is between 0.90 – 0.97 [3]. The absorptance (α_t) adopted is derived as follows:

$$\alpha_t = \frac{\alpha_c + \alpha_{bp}}{2} = \frac{0.70 + 0.90}{2} = 0.8$$

where α_c = Absorptivity of copper, α_{bp} = Absorptivity of black paint coating material. For the purpose of this work, the emittance of the absorber-tube material; $\varepsilon_t = 0.02$

The reflector is made of stainless steel of length (*L*) = 300mm, aperture width (*W*) = 800mm. Emissivity, $\varepsilon = 0.08$ [10]. Therefore, the reflectivity (ρ) can be computed using equation 7 [11].

$$\rho_c = 1 - \varepsilon = 1 - 0.08 = 0.92 \qquad \text{... 7}$$

The fluid is Air and assuming a mean temperature of the fluid to be $77\,^{o}\mathrm{C} = 350\ \mathrm{K}$, the following properties are selected from [12]:

Density (ρ_f) $= 1.009\ kg/m^3$, Thermal conductivity (K_f) $= 0.03003\ W/m.K$, Prandtl number ($\Pr$) = 0.697 , Specific heat capacity (Cp_f) = $1.0090\ J/kg.K$ and Viscosity (μ_f) = $2.075 \times 10^{-5}\ kg/m.s$,

Grashof number (Gr_d) can be computed using equation 5 [13].

$$Gr_d = \frac{\rho_f^2 g \beta \Delta T d_1^3}{\mu^2}, \qquad \text{... 8}$$

where density of the fluid (ρ_f) $= 1.009\ kg/m^3$, gravitational constant (g) $= 9.81\ \mathrm{m/s}$, inner diameter of the absorber-tube (d_1) $= 0.0100$ m and coefficient of expansion of the fluid (β) equals the reciprocal of T_f .

Therefore, $\beta = \dfrac{1}{350}$

Assuming a mean temperature of the absorber-tube (T_t) to be $130\,^{0}\mathrm{C}$ and the mean temperature of the fluid (T_f) to be $80\,^{0}\mathrm{C}$. Therefore, the mean temperature difference is computed as follows:

$$\Delta T = T_t - T_f = 130 - 80 = 50\,^{0}C .$$

Grashof number, $$Gr_d = \frac{(1.009)^2 \times 9.81 \times \frac{1}{350} \times 50 \times (0.0100)^3}{(2.075 \times 10^{-5})^2} = 3313.7$$

The expression used for the calculation of Nusselt number is given by [12]. It is written in equation 9 as follows:

$$Nu^{1/2} = 0.60 + 0.387 \left\{ \frac{Gr_d \Pr}{\left[1 + (0.559 / \Pr)^{9/16}\right]^{16/9}} \right\}^{1/6} \quad for \; 10^{-5} < Gr.\Pr < 10^{12} \qquad ...9$$

Therefore ,

$$Nu^{1/2} = 0.60 + 0.387 \left\{ \frac{(3313.7 \times 0.697)}{\left[1 + (0.559 / 0.697)^{9/16}\right]^{16/9}} \right\}^{1/6} = 3.5374$$

Correspondingly;

$$h_f = \frac{Nu \times K_f}{d_1} = \frac{12.5132 \times 0.03003}{0.0100} = 38.3383 \; W / m^2.K$$

where h_f = Convective heat-transfer coefficient for the inside of the absorber-tube, d_1 = inner diameter of the absorber-tube and K_f = thermal conductivity of the fluid.

The hourly thermal efficiency (□)of the collector is given by:

$$\eta = 1 - \frac{Q_{losses}}{Q_{input}} \qquad ...10$$

The hourly heat-energy supplied to the receiver (Q_{input}) can be computed as follows:

$$Q_{input} = I_g \times A_a = [(I_{beam}.R_b) + I_{diff}][WL] \qquad ...11$$

where $I_{beam} R_b$ = beam radiation on a tilt surface, I_{diff} = diffuse radiation, W = aperture diameter and L = length of the concentrator. The hourly tilt factor ($R_{b_{hr}}$) is the ratio of the beam radiation on a tilted surface to that on a horizontal surface and can be computed using Equation 12 [14].

$$R_b = \frac{\cos(\phi - s)\cos\delta\cos\omega + \sin(\phi - s)\sin\delta}{\cos\phi\cos\delta\cos\omega + \sin\phi\sin\delta} \qquad …12$$

The latitude of Bauchi is $\phi = 10.33^0$, A slope angle of 2^0 is adopted. Sukhatme [14] has given the following simple relation for calculating the declination:

$$\delta = 23.45 \sin[\frac{360}{365}(284 + n)] \qquad …13$$

Equation 13 is dependent on n , which is the day of the year.

The hour angle (ω) is the angular measure of time and is equivalent to 15^0 per hour. It is measured from noon based on local apparent time (LAT) being positive in the morning and negative in the afternoon.

The total heat-energy loss in the system (Q_{losses}) is considered as the sum of the radiative heat-loss from the surface of the enveloping glass-cover to the surroundings (q_{r_2}), the convective heat-loss from the surface of the enveloping glass-cover to the surroundings (q_c), the radiative heat-loss from the surface of the absorber-tube (q_{r_1}) and the conductive/convective heat-loss inside the absorber-tube (q_1). This can be expressed as in Equation 14.

$$Q_{losses} = q_{r_2} + q_c + q_{r_1} + q_1 \qquad ...14$$

The hourly radiative heat-loss from the enveloping glass-cover by radiation to the surroundings is given by:

$$q_{r_2} = \sigma\varepsilon_g A_g (T_g^4 - T_{sky}^4) \qquad ...15$$

where σ = Stefan-Boltzman’s constant, ε_g = emittance of the enveloping glass-cover material, A_g = surface area of the enveloping glass-cover, T_g = temperature of the enveloping glass-cover, and T_{sky} = sky temperature. The sky temperature can be expressed as

$$T_{sky} \approx T_{surr} - 6 \qquad \text{...16}$$

where T_{surr} = ambient temperature. The hourly convective heat-loss from the enveloping glass-cover to the surroundings is computed using Equation 17 as follows:

$$q_c = A_g . h(T_g - T_{sky}) \qquad \text{...17}$$

where h = the convective heat transfer coefficient given as:

$$h = 3.8 V_{wind} + 5.7 \text{ [14]} \qquad \text{...18}$$

where V_{wind} = wind velocity.

The hourly radiative heat-loss from the absorber tube to the enveloping glass-cover q_{r_1} can be computed using Equation 19.

$$q_{r_1} = \frac{A_t \sigma \left(T_t^4 - T_g^4\right)}{\dfrac{1}{\varepsilon_t} + \dfrac{A_t}{A_g}\left(\dfrac{1}{\varepsilon_g} - 1\right)}, \qquad \text{... 19}$$

where A_t = the surface area of the absorber-tube, T_t = temperature of the absorber-tube and ε_t = *emittance* of the absorber-tube material. The hourly conductive/ convective heat-losses, by the working fluid inside the absorber-tube (q_1) can be computed using Equation 20, as

$$q_1 = \frac{2 A_c \left(T_f - T_{sky}\right)}{\dfrac{1}{h_{surr}} + \dfrac{1}{h_f}} \qquad \text{...20}$$

This work employed the developed energy equations and geometric dimensions of a solar parabolic collector model TE. 38. The meteorological and radiation data were obtained from the work of [15]. The simulation maintained the same thermal properties of the collector materials and Mat lab program was used to simulate hourly thermal efficiency (η) of the system, varying the annular gap between the absorber-tube and the enveloping glass-cover.

III. RESULTS AND DISCUSSION

Table 1: Simulated values of hourly thermal efficiencies of the system for various annular gaps

TIME	SIMULATED THERMAL EFFICIENCIES OF THE SYSTEM FOR VARIOUS ANNULAR GAPS									
	$\eta_{0.010}$	$\eta_{0.015}$	$\eta_{0.020}$	$\eta_{0.025}$	$\eta_{0.030}$	$\eta_{0.035}$	$\eta_{0.040}$	$\eta_{0.045}$	$\eta_{0.050}$	$\eta_{0.055}$
08:00hr	85.648	81.785	77.923	74.060	70.197	66.335	62.472	58.610	54.747	50.884
09:00hr	91.428	89.118	86.810	84.496	82.186	79.876	77.565	75.255	72.944	70.634
10:00hr	94.227	92.669	91.115	89.553	87.996	86.438	84.880	83.323	81.765	80.207
11:00hr	96.110	95.060	94.015	92.962	91.913	90.864	89.815	88.766	87.716	86.667
12:00hr	96.662	95.762	94.866	93.964	93.064	92.165	91.265	90.366	89.467	88.567
13:00hr	97.716	97.103	96.492	95.877	95.264	94.651	94.038	93.425	92.812	92.199
14:00hr	97.431	96.743	96.057	95.368	94.680	93.992	93.304	92.617	91.929	91.241
15:00hr	96.824	95.975	95.125	94.276	93.426	92.577	91.728	90.878	90.029	89.180
16:00hr	94.307	92.785	91.262	89.741	88.219	86.697	85.175	83.653	82.131	80.609
17:00hr	87.837	84.579	81.321	78.063	74.805	71.547	68.288	63.030	61.773	58.514

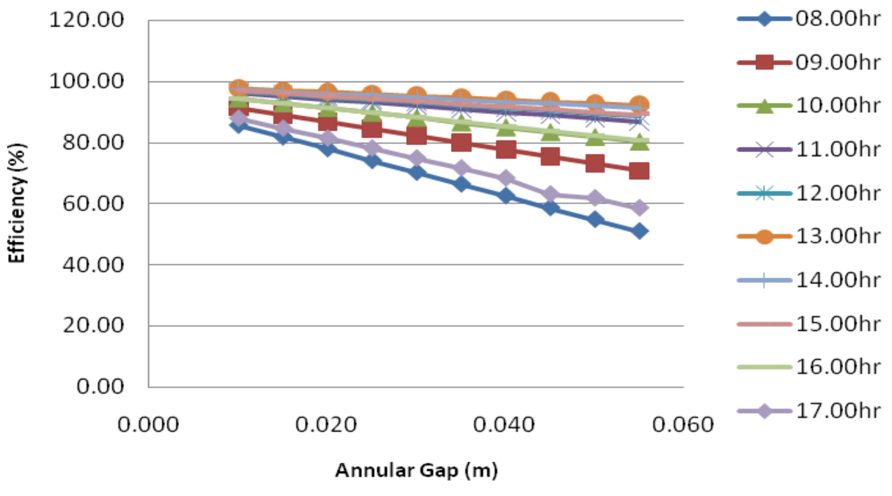

Fig. 5: Simulated Thermal Efficiency for Various Annular Gap

The simulation results in Table 1 and Figure 5 show that increase of annular gap reduces the thermal efficiency of the system. The higher the annular gap between the absorber-tube and the enveloping glass-cover, without corresponding increase in the diameter of the absorber-tube, the more is the total heat loss of the system and therefore, the lower is the thermal efficiency of the system.

IV. CONCLUSION

This paper investigated the annular gap effect on the performance of a solar parabolic-trough collector. The simulation results show that as the annular gap increases the thermal efficiency of the system reduces within the investigation range. The higher the annular gap between the absorber-tube and the enveloping glass-cover, the more is the total heat loss of the system and therefore, the lower is the thermal efficiency of the system.

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