CHAPTER XXII

THEORETICAL CALCULATIONS OF CYLINDRICAL COLLECTOR

Cuma ÇETİNER[1]

[1](Assoc. Prof.), Harran University Faculty of Engineering Mechanical Department Osmanbey Campus, Sanlıurfa
E-mail: ccetiner@gmail.com
ORCID: 0000-0002-5105-5699

1. Introduction

This section will theoretically calculate the thermal losses, thermal efficiency, thermal power and absorber dimensions of cylindrical and flat absorber collectors. The calculations are made assuming that the absorber is a flat collector. The cylindrical surface reflector mirror is a parabolic surface because it has a paraxial property [Gencelli,1995].

1.2 Thermal Analysis of Cylindrical Collector

Some of the radiation reaching the absorber from the collector is thrown into the low-temperature environment due to losses. At the same time, the fluid absorbs the other part and becomes useful energy. Although the absorber is insulated to prevent energy loss, it is not possible to prevent it completely. Losses in the absorber occur through conduction, convection and radiation from the glass cover and insulation to the environment. In steam-generating collectors, absorbers are usually placed in a glass envelope to prevent losses from the external environment. The absorber cover material used in this experimental set is a flat absorber, and flat glass and 50 cm thick heat insulation are used to protect the upper part of the absorber from the external environment. When the absorber shape is turned upside down, it resembles a flat collector. Therefore, the calculation method used for flat collectors can also be used here. Useful energy gained in flat collectors Hottel-Whillier-Bliss equation, which is most commonly used to find the useful energy gained in flat collectors, is also used in concentrated collectors [Duffie,1991];

$$Q_u = A_a F_R \left[I_Y - \frac{A_r}{A_a} U_T (T_g - T_{çev}) \right] \quad (1)$$

Useful energy here (Q_u), Aperture field (A_a), absorber area (A_y), heat gain factor (F_R), radiation intensity per unit absorber area (I_Y), Total heat transfer coefficient (U_T) from the absorber to ambient, fluid inlet temperature (T_g), shows the ambient temperature (T_{Cev}). The radiation intensity falling on the absorber area is

$$I_Y = \tau\rho\gamma\alpha I_d \quad (2)$$

Here, I_d amount of direct radiation per unit aperture area, γ intercept factor, τ transmittance coefficient of transparent cover, α Absorption rate of the absorber, ρ It shows the reflectivity rate of the reflective material. The following relation can be used for the thermal energy passing into the water in the absorber.

When the equation 2 is placed in the equation given above,

$$Q_u = F_R A_a I_d (\tau\alpha)\gamma\rho - F_R U_T A_y (T_g - T_{çev}) \quad (3)$$

it is found. The collector heat gain factor (F_R) in the equation represents the heat transfer capacity of the collector. When the useful thermal energy of the collector is divided by the direct radiation of the aperture area, the general efficiency expression used in collector collectors is found.

$$\eta_0 = \frac{Q_u}{A_a I_d} = F_R (\tau\alpha)\gamma\rho - F_R U_T \frac{A_y}{A_a} \frac{(T_g - T_{çev})}{I_d} \quad (4)$$

Ayrıca aşağıda verilen eşitlikten ısıl güç bulunabilir.

$$Q_u = mc_p (T_ç - T_g) \quad (5)$$

Efficiency by direct radiation is defined as the ratio of the amount of heat transferred to water to the direct solar energy reaching the reflective surface of the water.,

$$\eta = \frac{mc_p(T_ç - T_g)}{I_d A_a} \quad (6)$$

It is found. Here is the total area of the mirrors (A_a), our temperature ($T_ç$) and I_d It shows the direct radiation on the reflective mirrors. The enthalpies of the inflow of water and the outflow of steam in studies with steam are shown in the tables [Çengel,2012]. It can be found from the following equation by taking it.

$$\eta = \frac{m(h_ç - h_g)}{I_d A_a} \quad \textbf{(7)}$$

1. 3 Heat Transfer Coefficient of the Absorber

In the design of solar collectors, the total heat transfer coefficient (U_T) is found by calculating the heat losses through conduction, convection and radiation depending on the shape of the collector. When the absorber is considered to be a flat collector, some assumptions must be made for the analysis of energy losses according to the electrical resistance simulation rules. These assumptions are listed below;

- The solar radiation coming onto the collector is uniform and constant. The collector's performance is in steady-state conditions.
- The temperature of the absorber's top plate and glass cover is uniform.
- Energy losses are one-dimensional towards the bottom, top and side surfaces.
- The ambient temperature around the collector is assumed to be the same.
- Since the absorber in Figure 1 resembles a flat collector, the method applied in flat collectors is also valid in this absorber. Flat glass is used in this absorber as in other collectors to reduce the heat transfer losses of the absorber pipes.

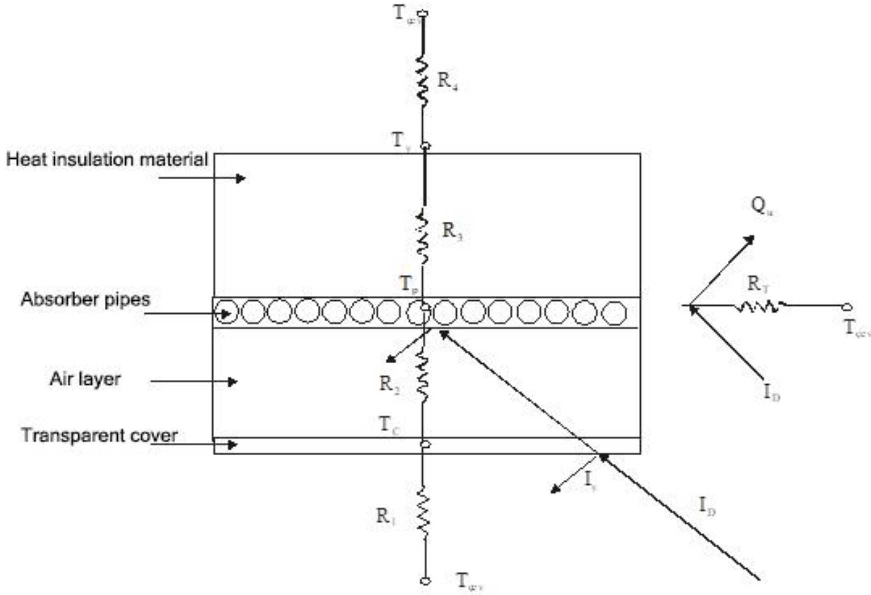

Figure 1. Thermal resistances in the absorber according to the electrical simulation method

If the properties of equations are similar to each other and the boundary conditions are similar to each other, the solutions of such equations can also be solved by analogy. Since the electric current occurs due to the volt difference and the heat transfer occurs due to the temperature difference, thermal resistors can also be used in heat transfer problems, similar to the resistors in series and parallel connection in electric current. The total thermal resistance RT in the absorber, whose thermal resistance is shown in Figure 1, the thermal resistance between the external environment and the glass cover R_1 at the bottom, the thermal resistance R_2 in the air gap between the glass cover and the absorber, the thermal resistance R_3 in the insulation layer, and the thermal resistance R_4 between the upper part of the absorber and the external environment dirençlerin toplamına eşittir. Yutucuda ısıl dirençlere göre enerji denklemi yazılırsa

$$Q_u = \frac{(T_P - T_{çev})}{R_T} = U_T A_y (T_p - T_{çev}) \tag{7}$$

Burada yutucunun yüzey alanı A_y, yutucu plaka sıcaklığı T_p, ortam sıcaklığı $_{Tçev}$ ile göstermektedir. Seri bağlı bir devredeki toplam direnç RT bütün dirençlerin toplamına eşit olacağından $R_T=R_1+R_2+R_3+R_4$ olur. Yutucudaki ısı geçiş katsayısı için yutucunun alt, üst ve yanlarından olan ısıl kayıplarını bulmak gerekir. Yutucuda yansıtıcı aynadan gelen ışınları yutan alt kısım için ısı geçiş katsayısı yazılırsa,

$$U_{alt} = \frac{1}{R_1 + R_2} \tag{8}$$

Here the following relation can be written for the resistance R_1:

$$R_1 = \frac{1}{h_{t-d} + h_{1-d}} \tag{9}$$

where ht-d is the heat transport coefficient due to outside air, which is given below by equation 10 [Duffie,1991,Kılıç1983]. V (m/s) in the equation indicates the wind speed. The unit of heat transport coefficient is W/m^2.

$$h_{t-d} = 5.7 + 3.8\ V \tag{10}$$

H1-D is the equivalent heat convection coefficient [Incoperia,1991], which includes the heat loss by radiation between the transparent cover below and the environment.

$$h_{1-d} = \varepsilon_c \sigma (T_c + T_{çev})(T_c^2 + T_{çev}^2) \tag{11}$$

It is found with. Here, the glass temperature is represented by Tc, the Stefean-Boltzman constant, σ and the radiant emission coefficient of the surface by ε_c. The thermal resistance between the transparent cover and the absorber is R2,

$$R_2 = \frac{1}{h_{1-p} + h_{k,p-c}} \tag{12}$$

It is calculated by equality. Here, since the heat convection coefficient between the transparent cover and the absorber plate (pipes) is considered as two parallel surfaces, the heat convection coefficient equivalent to radiation is [Halıcı,2011]

$$h_{1,p} = \frac{\sigma(T_P - T_c)(T_p^2 - T_c^2)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_c} - 1} \tag{13}$$

It is found by equality. The transparent cover with the absorbing surface is an oblique surface. The convection coefficient between the absorber surface and the glass cover must be known. For the proposed convection coefficient on an inclined surface, the following relation is found [Heseih 1986, Duffie 1991].

$$h_{k,p-c} = \left\{1 + 1{,}44\left[1 - \frac{1708}{Ra\cos s}\right]\left[1 - \frac{(\sin 1{,}8s)^{1{,}6}(1708)}{Ra\cos s}\right] + \left[\left(\frac{Ra\cos s}{5830}\right)^{1{,}3} - 1\right]\right\} \tag{14}$$

Where s is the slope of the absorber, and Ra is the Rayleigh number, the equality of which is given below.

$$Ra = \frac{2gL^3\,\mathrm{Pr}(T_p - T_c)}{\nu^2(T_p + T_c)} \tag{15}$$

Insulation materials of different thicknesses were used to reduce the heat transfer of the external environment to the upper part of the absorber by convection. Heat transfer coefficient for the upper insulation plate

$$U_{üst} = \frac{1}{R_4 + R_3} \tag{16}$$

Writable. As can be seen in Figure 1, synthetic insulation, amount layer, aluminium plate, and glass wool were used in various thicknesses and covered with stainless steel to protect it from the external environment. R3 resistance for insulation materials

$$R_3 = \frac{l_1}{k_1} + \frac{l_2}{k_2} + \frac{l_3}{k_3} + \frac{l_4}{k_4} + \frac{l_5}{k_5} \tag{17}$$

It is found with. Here it shows the thicknesses of l1 synthetic insulation, l2 amyant layer, l3 aluminum plate, l4 glass wool and l5 stainless steel. For R4 resistor

$$R_4 = \frac{1}{h_{d-d}} \tag{18}$$

Equality can be used. Here there is some radiation loss between the upper part of the absorber and the periphery. However, since the temperature difference between the upper part of the swallow and the environment is very small, it is negligible.

Coefficient of heat loss from the sides of the absorber [Heseih,1995, Duffie,1991]

$$U_k = 0,6\left[\frac{A_{y-çevresi}}{A_y}\right] \tag{19}$$

It is found by equality. When the equations 8, 16, and 19 for the absorber above are summed, the following relation is found for the heat transfer coefficient.

$$U_T = \left[U_{üst} + U_{alt} + U_k\right] \tag{20}$$

Heat gain factor for a collector in Equation 5 above

$$F_R = \frac{mc_p}{A_y U_T}\left[1 - \exp\left(-\frac{A_y U_T F}{mc_p}\right)\right] \tag{21}$$

It is found by correlation. The efficiency factor F here is a factor that is independent of the working conditions and depends on the construction of the collector. This value is found in the following equation [Heseih, 1986, Duffie,1991 Halıcı].

$$F = \frac{1}{1 + \dfrac{U_T}{h_1 + \dfrac{1}{\dfrac{1}{h_2} + \dfrac{1}{hr}}}} \tag{22}$$

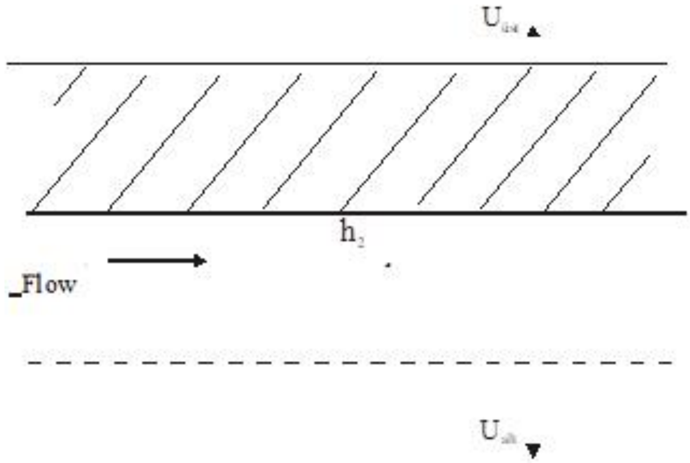

Figure 2. Streaming within a channel

From Figure 2, h1 = h2 was accepted. The following equation can be used for convection due to radiation in the glass cover [Incoperia,2001]. where Tort is the average temperature of the plate with the glass.

$$h_r = \frac{4\sigma T_{ort.}^3}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_c} - 1} \tag{23}$$

According to Reynold and the average temperature of the water, Pr and other physical values are taken from the table. The following equation was used for the Nu number in liquids and gases [Halıcı,2011].

$$N_u = 3{,}65 + \frac{0{,}19(\mathrm{Re}\,\mathrm{Pr}\,D_h / L)^{0{,}8}}{1 + 0{,}117(\mathrm{Re}\,\mathrm{Pr}\,D_h / L)^{0{,}467}} \tag{24}$$

Here, Dh is the hydraulic diameter of the absorber, and L is its length. The convection coefficient was determined according to the number of Nu. The values found are put in 22 and F is found. It is substituted in equation 20 used for F_R from F. After calculating the total heat transfer coefficient with equation 19, the thermal power can be calculated in equation 1 (Q_u) when the inlet and outlet temperatures of the absorber and the absorber and reflective areas are known.

1.4. Determination of Absorber Size and Focal Length

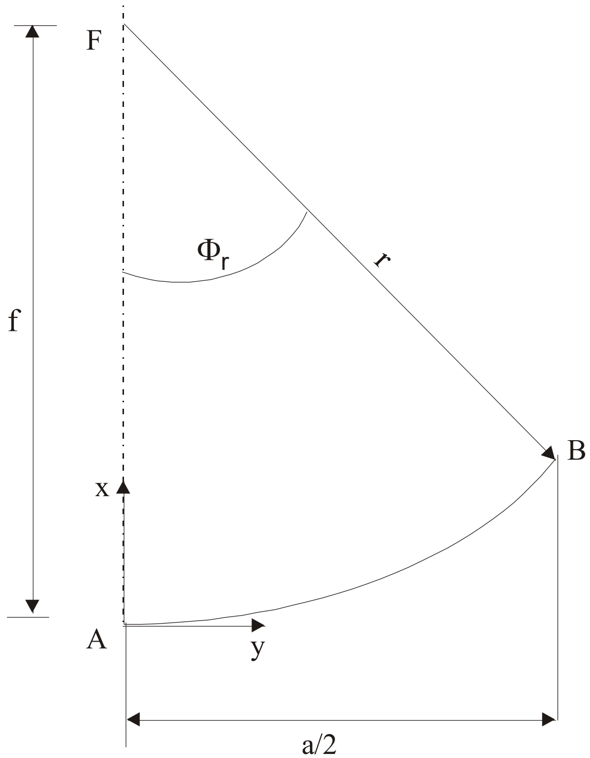

Figure 3. Focal length and edge angle in a parabolic concentrator

As can be seen in Figure 3, in a parabolic concentrator, the focal length f shows the aperture of the parabolic reflector a, the radius r, and the edge angle r between AFB ϕ. Parabola equation

$$y^2 = 4fx \tag{25}$$

and edge angle (ϕ_r),

$$\phi_r = \arctan g\left[\frac{8(f/a)}{16(f/a)^2 - 1}\right] = \arcsin\left(\frac{a}{2.r}\right) \tag{26}$$

$$y = f - \frac{x^2}{4f} \tag{27}$$

$$r = \frac{2f}{\sin^2 \phi_r}(1-\cos\phi_r) = \frac{2f}{1+\cos\phi_r} = \frac{f}{\cos^2(\phi_r/2)} \tag{28}$$

As can be seen in Figure 4, the sun appears from the Earth at an angle of 32. W The following relations are used for the theoretical full focus of the image to show the image of the sun on the concentrator.

$$w = 2r\tan 16` = \frac{4f}{1+\cos\phi_r}\tan 16` \tag{29}$$

for projection on the plane perpendicular to the optical axis in the figure

$$w` = \frac{w}{\cos\phi_r} = \frac{4f}{\cos(1+\cos\phi_r)}\tan 16` = \frac{2r^2}{2f-r}\tan 16` \tag{30}$$

r=R alınırsa

$$W` = \frac{4f}{\cos(1+\cos\phi_r)}\tan 16` = \frac{2R^2}{2f-R}\tan 16`$$

It is equality. Where W' denotes the width of the absorber.[Kılıç 1983]

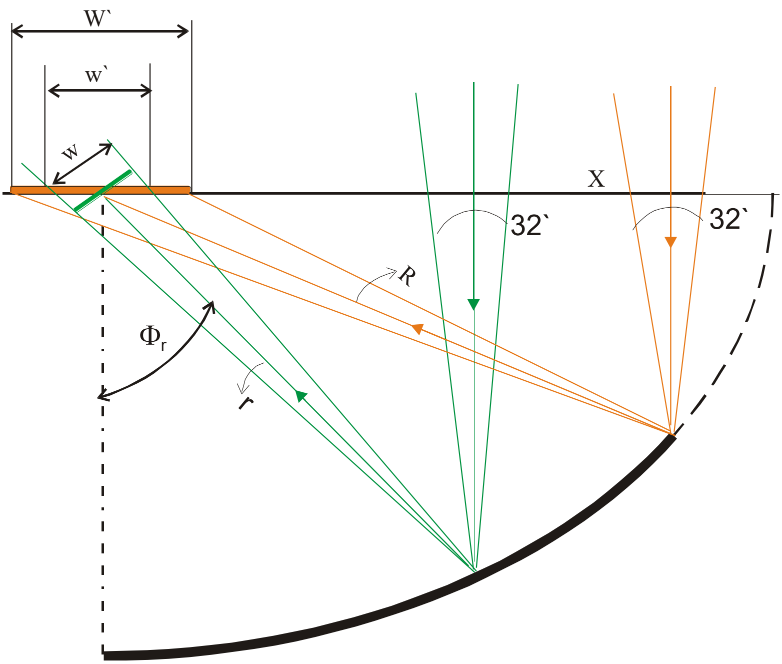

Figure 4. Illustration of absorber dimensions for a parabolic reflector

2. Conclusion

In this study, theoretical calculations were made on the thermal efficiency and thermal power of a cylindrical collector with a fixed absorber. The absorber used is flat and the fluid passes through thin pipes. In theoretical calculation, it is shown how thermal efficiency, thermal power and heat transfer coefficient are calculated with the help of equations used in thermodynamics and heat transfer.

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