THERMAL PERFORMANCE OF PARABOLIC CONCENTRATING COLLECTORS

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Abstract For many applications it is desirable to deliver energy at temperature higher than those possible with flat-plate collectors. Decreasing the area from which heat losses occur can increase energy delivery temperatures. This is done by interposing an optical device between of source of radiation and the energy-absorbing surface. He small absorber will have smaller heat losses compared to a flat-plate collector at the same absorber temperature. This paper present a method to evaluate the thermal performance of parabolic concentrating collectors.

Keywords Parabolic collectors, collector efficiency, thermal performance

1. CONCENTRATING COLLECTORS

Concentrations ratios (the ratios of collector aperture area to absorber area, which are approximately the factors by which radiation flux on the energy-absorbing surface is increased) can very over several orders of magnitude. Increasing ratios mean increasing temperatures at which energy can be delivered and increasing requirements for precision in optical quality and positioning of the optical system. Thus the cost of delivered energy from a concentrating collector is a function of the temperature at which it is available. From an engineering point of view, concentrating collectors present problems in addition to those of flat-plate collectors. They must (except at very low end of the concentration ratio scale) be oriented to "track" the sun so that the beam radiation will be directed onto the absorbing surface. There are also new requirements for maintenance, particularly to retain the quality of optical systems for long periods of time in the presence of dirt, weather, and oxidizing or other corrosive atmospheric components.

2. CONCENTRATION RATIO

The most common definition of concentration ratio is an area concentration ratio, the ratio of the area of aperture to the area of the receiver. The aria concentration ratio is:

$$C = \frac{A_a}{A_r} \tag{1}$$

Consider the circular concentrator with aperture area A_a and receiver area A_r viewing the sun of radius r at distance R, as shown in Figure 1. The half-angle subtended by the sun is Θ_S . (The receiver is shown beyond the aperture for clarity, the argument is the same if it is on the same side of the aperture as the sun.

If the concentrator is perfect, the radiation from the sun on the aperture (and thus also on the receiver) is the fraction of the radiation emitted by the sun, which is intercepted by the aperture.

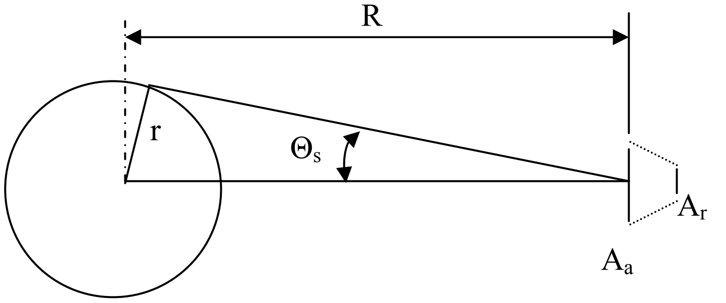

Fig. 1 - Schematic of sun at T_s at distance R from a concentrator with aperture area A_a and receiver area A_r

Although the sun is not a blackbody, for purposes of an approximate analysis it can be assumed to be a blackbody at T_s:

$$Q_{s \to r} = A_a \frac{r^2}{R^2} \sigma T_s^4 \tag{2}$$

A perfect receiver radiates energy equal to $A_r T_r^4$, and a fraction of this, E_{r-s}, reaches the sun:

$$Q_{r \to s} = A_r \sigma T_r^4 E_{r-s} \tag{3}$$

When T_r and T_s are the same, the second law of thermodynamics requires the $Q_{s \to r}$ be equal to $Q_{r \to s}$. So from equations 2 and 3

$$\frac{A_a}{A_r} = \frac{R^2}{r^2} E_{r-s} \tag{4}$$

and since the maximum value of E_{r-s} is unity, the maximum concentration ratio for circular concentrators is

$$\left(\frac{A_a}{A_r}\right)_{circular,\max} = \frac{R^2}{r^2} = \frac{1}{\sin^2 \Theta_s} \tag{5}$$

A similar development for linear concentrators leads to

$$\left(\frac{A_a}{A_r}\right)_{circular,\max} = \frac{1}{\sin \Theta_s} \tag{6}$$

Thus with Θ_s = 0.27º, the maximum possible concentration ratio for circular concentrators is 45.000, and for linear concentrators the maximum is 212.

3. THERMAL PERFORMANCE OF CONCENTRATING COLLECTORS

The absorbed radiation per unit area of aperture S must be estimated from the radiation and the optical characteristics of the concentrator and receiver. Thermal losses from the receiver must be estimated, usually in terms of a loss coefficient U_L which is based on the area of the receiver. In principle, temperature gradients on the receiver can be accounted for by a flow factor F_R to allow the use of inlet fluid temperatures in energy balance calculations. With F_R and U_L known, can be calculated the collector useful gain. For the calculation of thermal loss coefficient U_L , consider an uncovered cylindrical absorbing tube that might be used as a receiver with a linear concentrator. Assume that there are no temperature gradients around the receiver tube. The loss coefficient considering convection and radiation from the surface and conduction through the support structure is:

$$U_L = h_w + h_r + U_{cond} \tag{7}$$

The linearized radiation coefficient can be calculated from:

$$h_r = 4\sigma\varepsilon\overline{T}^3 \tag{8}$$

where $\overline{T}$ is the mean temperature for radiation and ε is the emittance of the absorbing surface. If a single value of h_r is not acceptable due to large temperature gradients in the flow direction, the collector can be considered as divided into two or more segments each with constant h_r. Estimation of conductive losses must be based on knowledge of the details of construction or on measurements on a particular collector.

If the annulus is evacuated so that the convection coefficient $h_{r,c-a}$ is negligible U_L based on the absorber area A_r is:

$$U_L = \left[\frac{A_r}{\left(h_w + h_{r,c-a}\right)A_c} + \frac{1}{h_{r,r-c}}\right]^{-1} \tag{9}$$

The procedure is to estimate T_c (which will under these conditions be much closer to T_a than T_r), calculate U_L and check T_c by an energy balance on the cover. If solar radiation absorbed by the cover is negligible, this balance is:

$$A_c\left(h_{r,c-a} + h_w\right)\left(T_c - T_a\right) = A_r h_{r,r-c}\left(T_r - T_c\right) \tag{10}$$

Solving for the cover temperature:

$$T_c = \frac{A_r h_{r,r-c} T_r + A_c\left(h_{r,c-a} + h_w\right)T_a}{A_r h_{r,r-c} + A_c\left(h_{r,c-a} + h_w\right)} \tag{11}$$

The overall heat transfer coefficient from the surroundings of the fluid is:

$$U_o = \left[\frac{1}{U_L} + \frac{D_o}{h_{fi} D_i} + \frac{D_o \ln\left(D_o / D_i\right)}{2k}\right]^{-1} \tag{12}$$

where D_i and D_o are the inside and outside tube diameters, h_{fi} is the heat transfer coefficient inside the tube, and k is the thermal conductivity of the tube.

The useful energy gain per unit of collector length $q_u^{'}$, expressed in terms of the local receiver temperature T_r and the absorbed solar radiation per unit of aperture area S, is:

$$q_u^{'} = \frac{A_a S}{L} - \frac{A_r U_L}{L}\left(T_r - T_a\right) \tag{13}$$

where A_a is the unshaded area of the concentrator aperture and A_r is the area of the receiver ($\pi D_o L$ for the cylindrical absorber). In terms of the energy transfer to the fluid at local fluid temperature T_f

$$q_u^{'} = \frac{\left(A_r / L\right)\left(T_r - T_f\right)}{\frac{D_o}{h_{fi} D_i} + \left(\frac{D_o}{2k}\ln\frac{D_o}{D_i}\right)} \tag{14}$$

If T_r is eliminated from equations 13 and 14, we have:

$$q_u^{'} = F^{'}\frac{A_a}{L}\left[S - \frac{A_r}{A_a}U_L\left(T_f - T_a\right)\right] \tag{15}$$

where the collector efficiency factor $F^{'}$ is:

$$F' = \frac{1/U_L}{\frac{1}{U_L} + \frac{D_o}{h_{fi} D_i} + \left(\frac{D_o}{2k} \ln \frac{D_o}{D_i} \right)} \tag{16}$$

or

$$F' = U_o / U_L \tag{17}$$

$$Q_u = F_R A_a \left[S - \frac{A_r}{A_a} U_L (T_i - T_a) \right] \tag{18}$$

The collector flow factor F'' is:

$$F'' = \frac{F_R}{F''} = \frac{\dot{m} C_p}{A_r U_L F'} \left[1 - \exp\left(- \frac{A_r U_L F'}{\dot{m} C_p} \right) \right] \tag{19}$$

In the next paragraph is presented an example.
A cylindrical parabolic concentrator with width 2.5 m and length 10 m has an absorbed radiation per unit area of aperture of 430 W/m^2. The receiver is a cylinder painted flat black and surrounded by an evacuated glass cylindrical envelope. The absorber has a diameter of 60 mm, and the transparent envelope has a diameter of 90 mm. The collector is designed to heat a fluid entering the absorber at 200 C at a flow rate of 0.0537 kg/s. The fluid has C_p=3.26 kJ/kgC. The heat transfer coefficient inside the tube is 300 W/m^{2}C and the overall loss coefficient is 10.6 W/m^{2}C. The tube is made of stainless steel (k=16 W/mC) with a wall thickness of 5 mm. If the ambient temperature is 10 C, calculate the useful gain and exit fluid temperature.
The solution is based on equation 18. The area of the receiver is:

$$A_r = \pi DL = \pi \cdot 0.06 \cdot 10 = 1.88 m^2 \tag{20}$$

taking into account shading of the central part of the collector by the receiver

$$A_a = (2.5 - 0.09) 10 = 24.1 m^2 \tag{21}$$

To calculate F_R, we first calculate F' for this situation from equation 16:

$$F' = \frac{1/10.6}{\frac{1}{10.6} + \frac{0.06}{300 \cdot 0.05} + \frac{0.06 \ln(0.06/0.05)}{2 \cdot 16}} = 0.96 \tag{22}$$

Then F_R from equation 19 is calculated:

$$\frac{\dot{m} C_p}{A_r U_L F'} = \frac{0.0537 \cdot 3260}{1.88 \cdot 10.6 \cdot 0.96} = 9.15 \tag{23}$$

$$F'' = 9.15 \left(1 - e^{-1/9.15} \right) = 0.95 \tag{24}$$

$$F_R = F'' \times F' = 0.95 \cdot 0.96 = 0.91 \tag{25}$$

the useful gain is:

$$Q_u = 24.1 \cdot 0.91 \left[430 - \frac{1.88 \cdot 10.6}{24.1} (200 - 10) \right] = 5980 W \tag{26}$$

The exit fluid temperature is:

$$T_o = T_i + \frac{Q_u}{\dot{m} C_p} = 200 + \frac{5980}{0.0537 \cdot 3260} = 234\ C \tag{27}$$

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