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Performance Analysis Methodology for Parabolic Dish Solar Concentrators for Process Heating Using Thermic Fluid

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Abstract: *Solar Energy Technology has an important role to play in the present Energy and Environment crises. Solar Concentrator Technology has good potential for various applications, including power generation and process heating applications. Scope of Research and Development is ample in this area, as very little work has been done on it. Most of it has been done in the US and Europe, in solar thermal power systems, such as solar concentrators coupled to a Stirling engine or to a single or double circuit Rakine Cycle system. Parabolic Solar Dish Concentrators (PDSCs) can be very useful in Industrial Process Heat applications, which use about 20% of total oil consumption in India. If only 25 to 30% of this can be saved by putting up Solar Concentrators, it will save import of 4.5 MT oil per year, which is about 6% of our oil imports. So, solar concentrator for process heat requirements of community, industrial and commercial establishments is an emerging and exciting opportunity in India, which is gaining attention from scientists, engineers and developers. The present paper presents a Performance Analysis Methodology developed for a PDSC system used for heating thermic fluid for process heating application.*

Keywords: *Solar Concentrators, Performance Analysis, Solar Process Heating Application, Parabolic Dish Solar Concentrators.*

I. Introduction

Energy is necessary for Life and Sun is the main source of Energy for Earth. Solar Energy converted and inter-converted to and from various forms, supports Life on Earth. The development of human ability to harness and utilize energy from various sources has had a major influence on evolution of Human Life and Civilization. Industrial Revolution resulted into great changes in energy conversion and utilization patterns, which brought about great changes in human life style, socio-economic and political conditions and environment, as well. Energy Crisis and Global warming are glaring effects of the outcomes of those changes which commend urgent concern for developing and implementing cleaner and greener methods of sustaining Life. Solar Energy Technology has a major role to play in this context [1]. Sunlight is free, clean, renewable and technically exploitable in most part of the inhabited earth [2]. Solar Energy can be utilized through Solar Photovoltaic and Solar Thermal Systems. Solar Photovoltaic Systems generate electricity through photovoltaic (PV) cells. This technology directly converts sunlight into electricity. Some of the main drawbacks for the solar technology are due to the high investment cost and long payback period [2, 3, 4]. Solar Thermal Systems can be Flat Plate type (for low temperature ranges) or Concentrator type (for medium and high temperature ranges), which show considerable potential as best options to overcome the crisis / problems.

Most of the work done so far in the field of Solar Concentrator Technology is for generation of electricity. Solar thermal power systems utilize the heat generated by concentrating and absorbing incident solar energy to drive a heat engine / generator and produce electric power. Three generic solar thermal power systems, trough, power tower, and dish/engine systems, are being employed for power production [6, 7]. Trough systems use linear parabolic concentrators to focus sunlight along the focal lines of the collectors. In a power tower system, a field of two-axis tracking mirrors, called heliostats, reflects the solar energy onto a receiver that is mounted on top of a centrally-located tower. Dish/engine systems, the third type of solar thermal power system, comprise a parabolic dish concentrator, a thermal receiver, and a heat engine/generator located at the focus of the dish to generate power. Of the three solar thermal power technologies, trough electric systems are the most mature. They have a sun concentration ratio of about 75 and operate at temperatures of around 400^{o}C at an annual efficiency of about 10% [8]. Power towers operate at a concentration ratio of around 800 and temperatures of about 560^{o}C and have annual efficiencies of about 15%. Dish/engine systems are characterized by high efficiency, modularity, autonomous operation, and an inherent hybrid capability (the ability to operate on either solar energy or a fossil fuel, or both). A number of thermodynamic cycles and working fluids have been considered for dish/engine systems. These include Rankine cycles, using water or an organic working fluid; Brayton cycles, both open and closed cycles; and Stirling cycles. Among these, Stirling engines have been mainly used and developed in the US and Europe [9, 10]. The dish/Stirling systems seem potential to become one of the least expensive sources of renewable energy due to their high solar-to-electric conversion efficiency (29.4% reported in 1984 [11], and a new record of 31.25% in 2007 [12]). However, dish/Stirling systems are quite heavy with very high initial costs. Also, because of the high pressure (>20 MPa) and high temperature (>700^{o}C), the engine are expensive to make [13]. In addition, the engine in a dish/engine system converts heat to mechanical power in a manner similar to conventional engines; compressing a working fluid when it is cold, heating the compressed working fluid, and then expanding it through a turbine or with a piston to produce work. Therefore, the motion of machinery parts is inevitable during the operation of the dish/engine system. Moving parts are also invariably associated with problems of wear and tear due to the friction. Although these can be reduced with proper lubrication, there is virtually no way they can be eliminated. Moreover, lubrication comes with its attendant problems like oil leakages, sealant problems, and possibly undesirable chemical interaction with other components. In addition, moving parts and lubricant movement (if liquid) will give rise to problems related to the dynamic stability of the structure. Noise and vibration may also be a problem [5]. A way to alleviate the problems discussed above is using a new power conversion unit (PCU)—alkali metal thermal to electric converter (AMTEC) with dishes. The AMTEC is a thermally regenerative electrochemical device for the direct conversion of heat to electrical energy [14]. It has many inherent advantages over conventional forms of electric generation, such as high efficiency (20–40%) [15], high density (up to 0.5 kW/kg) [16], absence of moving parts, good reliability, maintenance-free, modular design, competitive manufacturing costs, and compatible with many heat and fuel sources, including radioisotope, concentrated solar, external combustion, and nuclear reactor etc. The use of Solar Concentrator Technology in other industries, including the Process Industry, which holds considerable potential in medium temperature applications, is in the process of gaining due attention from scientists, engineers and developers [1].

In India, Dish Technology to harness solar energy for generating electricity is yet to get established. However, pioneering work on use of Scheffler Dish for SOLAR COOKING was done by Shree Deepak Gadhia of Gadhia Solar, Valsad, Gujarat to overcome the shortcomings of box type solar cooker (could only boil and roast the food) and mini concentrating dish type solar cookers (cumbersome outdoor usage). It was introduced for community cooking about two decades back and was successfully used for other applications like desalination, food processing, etc. [1]. Gadhia Solar supplied and installed several Solar steam cooking systems of different sizes ranging to cook from 500 to 40,000 people and for different user groups starting with temples and ashrams and Army, Hospitals, Industrial Canteens, Hostels etc. [17] Parabolic collectors using thermic fluid as working medium were developed and are successfully working for cooking at Muni Seva Ashram, Goraj, Gujarat to overcome the limitations of steam as working medium – better thermal storage, transfer and control [18]. Some other efforts initiated for use of Solar Concentrators in IPH in India are as follows :

- Prototype design of solar parabolic dish collector with truncated cone shaped helical coiled receiver made up of copper, coated with nickel chrome at focal point. Instantaneous efficiency of 63.9% has been achieved in this system, which can be used for heating boiler feed water, laundry applications and other steam generation applications [19].
- Development of ARUN brand of SCS for pasteurization of milk at Mahananda Dairy in Latur, Maharashtra [20].
- A parabolic dish collector is developed from low cost technology and tested outdoors. The absorber, made of aluminium alloys and coated with black paint, is placed on the focal receiver. The calculated overall heat transfer co-efficient varies from 130 to 180 W/m^2K for actual climate conditions at Tiruchirappalli, India. The thermal efficiency of collector is found to be 60% and the cost is minimized to half the cost of a collector that is available in the market with same specifications [21].
- The Megawatt System (MWS Solar Dish Concentrator) - A High Efficiency 2-Axis Tracking Concentrator: Megawatt is a Co. in Delhi, which has developed indigenous parabolic dish type collector of $90m^2$ area. Prototype has been built and is being tested at Solar Energy Center of Ministry of New and Renewable Energy (MNRE). Now, this technology is undergoing field testing under R & D support of MNRE for industrial heating applications at M/s. Universal Medicap Limited, Vadodara, Gujarat (India) [1].

To study and analyze the performance of the Parabolic Dish Solar Concentrator (PDSC) system, the geometry of the SPDC and its influence / relation with the optical behavior of the system, which in turn affects its thermal performance, needs to be studied. The parameters which affect the performance of PDSC can be classified into three categories : design specifications, measured (operating) parameters and calculated / derived parameters. Also, solar radiation geometry, which depends upon the geographic and climatic conditions of the location where PDSC system in installed, and parameters related to it affect the amount of incident solar radiation and hence, the performance of the system [22]. The paper discusses the role of each of these types of parameters to arrive at a basic methodology to analyze the performance of a Solar Parabolic Dish Concentrator.

II. Design Specifications Of Parabolic Dish Solar Concentrator

Concentration of the sun rays in parabolic dish type solar concentrator, is done, as the name suggests, according to the geometry of the concentrator i.e. parabola. A parabola is the locus of a point that moves so that its distances from a fixed line and a fixed point are equal. Fig. 1 shows a parabola, where the fixed line is called the directrix and the fixed point F, the focus. The length FR equals the length RD. The line perpendicular to the directrix and passing through the focus F is called the axis of the parabola. The parabola intersects its axis at a point V called the vertex, which is exactly midway between the focus and the directrix [23].

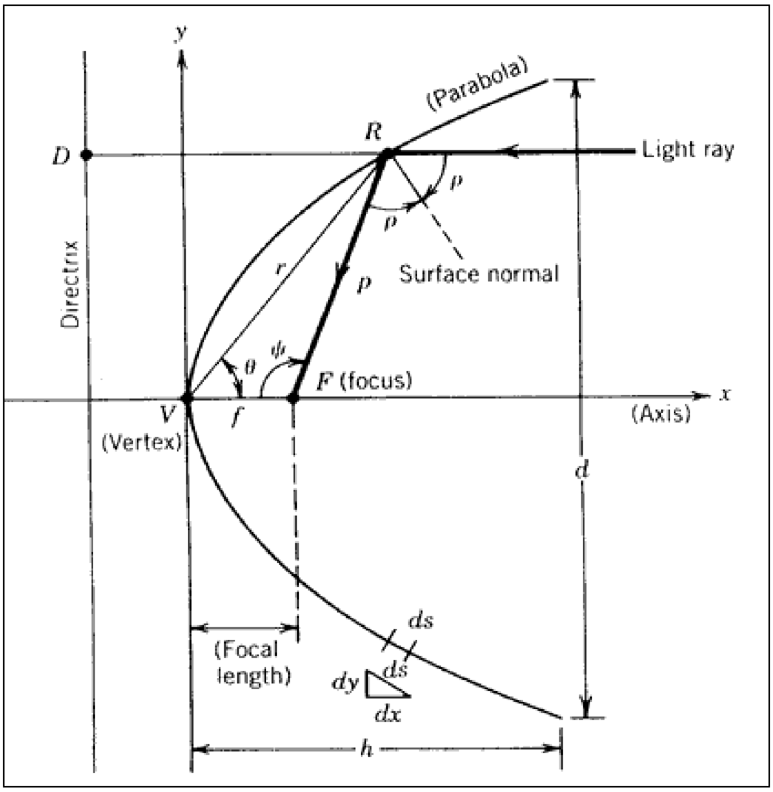


Fig. 1: The Parabola [23].

Important and useful definitions / terminology related to Parabolic Dish Solar Concentrator are as follows [24] :

- Aperture area of dish concentrator (m^2) : The total surface area of concentrator upon which solar energy is incident.
- Diameter of dish aperture (d) : The diameter of dish (denoted by d in the Fig. 1).
- Focal length (f) : The focal length is the distance VF from the vertex to the focus.
- Dish rim angle (ψ) : Solar concentrators use a truncated portion of the basic parabola curve. The extent of this truncation is usually defined in terms of the rim angle or the ratio of the focal length to aperture diameter f/d.
- Height of the curve (h) : The maximum distance from the vertex to a line drawn across the aperture of the parabola.
- Arc length (s) : Found by integrating a differential segment of the basic parabola curve and applying the limits x = h and y = d/2.

The method of deriving these parameters is as follows [22, 23, 24] :

If the origin is taken at the vertex V and the x-axis along the axis of the parabola, the equation of the parabola is :

$$y^2 = 4fx \tag{1}$$

where f, the focal length, is the distance VF from the vertex to the focus.

When the origin is shifted to the focus F as is often done in optical studies, with the vertex to the left of the origin, the equation of a parabola becomes :

$$y^2 = 4f(x + f) \tag{2}$$

In polar coordinates, using the usual definition of r as the distance from the origin and θ, the angle from the x-axis to r, we have for a parabola with its vertex at the origin and symmetrical about the x-axis :

$$\frac{sin^2\theta}{\cos\theta} = \frac{4f}{r} \tag{3}$$

Often in solar studies, it is more useful to define the parabolic curve with the origin at F and in terms of the angle ψ in polar coordinates with the origin at F. The angle ψ is measured from the line VF and the parabolic radius p, is the distance from the focus F to the curve. Shifting the origin to the focus F, we have :

$$p = \frac{2f}{1+\cos\psi} \tag{4}$$

The parabolic shape is widely used as the reflecting surface for concentrating solar collectors as it has the property that, for any line parallel to the axis of the parabola, the angle ρ between it and the surface normal is equal to the angle between the normal and a line to the focal point. Since solar radiation arrives at the earth in essentially parallel rays and by Snell's law the angle of reflection equals the angle of incidence, all radiation parallel to the axis of the parabola will be reflected to a single point F, which is the focus. So, from Fig. 1, it is clear that :

$$\psi = 2\rho \tag{5}$$

The general expressions given so far for the parabola define a curve infinite in extent. Solar concentrators use a truncated portion of this curve. The extent of this truncation is usually defined in terms of the rim angle ψ_{rim} or the ratio of the focal length to aperture diameter f/d. The scale (size) of the curve is then specified in terms of a linear dimension such as the aperture diameter d or the focal length f. This is readily apparent in Figure 2, which shows various finite parabola curves having a common focus and the same aperture diameter.

A parabola with a small rim angle is relatively flat and the focal length is long compared to its aperture diameter. Once a specific portion of the parabolic curve has been selected, the height of the curve, h may be defined as the maximum distance from the vertex to a line drawn across the aperture of the parabola. In terms of focal length and aperture diameter, the height of the parabola is :

$$h = \frac{d^2}{16f} \tag{6}$$

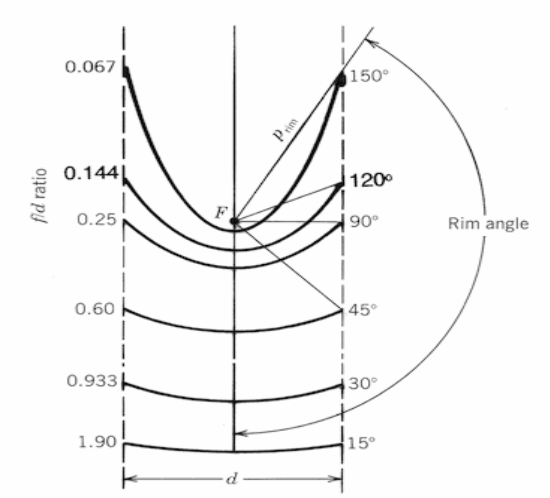


Fig. 2: Segments of a parabola having a common focus F and the same aperture diameter.

Thus, the rim angle ψ_{rim} in terms of the parabola parameters is :

$$tan\psi_{rim} = 1/(d/8h) - (2h/d) \tag{7}$$

Another property of the parabola that may be of use in understanding solar concentrator design is the arc length s. This may be found for a particular parabola by integrating a differential segment of this curve and applying the limits x = h and y = d/2. The result is :

$$s = \left[\frac{d}{2}\sqrt{\left(\frac{4h}{d}\right)^2 + 1}\right] + 2f\ 1n\left[\frac{4h}{d}\sqrt{\left(\frac{4h}{d}\right)^2 + 1}\right] \tag{8}$$

Where d is the distance across the aperture (or opening) of the parabola as shown in fig. 2 and h is the distance from the vertex to the aperture. The cross sectional area of the space enclosed between a parabola and a line across its aperture and normal to the axis is given by :

$$A_x = \frac{2}{3} d.h \tag{9}$$

However, this is not the reflecting surface area of a parabolic trough or dish or their aperture areas. It is given by :

$$A_a = \frac{\pi(2\rho sin\psi_{\text{rim}})^2}{4} = 4\pi f^2 \frac{sin^2\psi_{rim}}{(1+cos\psi_{rim})^2} \tag{10}$$

The important parameters used while evaluating parabolic geometry and related optical derivations are as follows :

$$tan\Psi_{rim} = (f/d)/2(f/d)^2 - 1/8 \tag{10}$$

$$\tan\left(\frac{\Psi_{rim}}{2}\right) = 1/4\left(\frac{f}{d}\right) \tag{11}$$

$$\frac{f}{d} = \frac{1+cos\Psi_{rim}}{4sin\Psi_{rim}} \tag{12}$$

$$\frac{f}{d} = 1/\{4\tan\left(\frac{\Psi_{rim}}{2}\right)\} \tag{13}$$

Thus, design parameter like rim angle ψ_{rim} and arc length s are derived from the basic equations of the geometry of the parabola which are further used in finding out the derived parameters for evaluating the performance analysis of the parabolic dish solar concentrators.

Apart from these, other system parameters like the thermo-physical properties of the thermic fluid (density, specific heat, viscosity, etc.), the type and specifications of the receiver, etc. are also needed.

III. Operating (Measured) Parameters

Important measured parameters needed to be known to assess the performance of PDSC include solar insolation (DNI) incident on the collector, wind velocity, ambient temperature, mass flow rate of thermic fluid, inlet and outlet temperature of the thermic fluid into and from the receiver respectively, temperature inside the receiver cavity, angle of inclination of the receiver from the horizontal, etc. Definitions of some of the operating parameters with respect to the present study are :

Stagnation temperature: Solar Thermal Collector performance testing utilizes the term stagnation temperature to indicate the maximum achievable collector temperature with a stagnant fluid (no motion).

Ambient temperature: The temperature of the surroundings (same as room temperature indoors).

Effective absorbance reflectance product: Solar reflectance measures the amount of solar energy that is reflected back into the atmosphere after coming into contact with a given material. Solar absorption refers to the amount of solar energy that is transmitted into the material itself and is used as a measure to determine the amount of heat and cool energy that is transmitted through the material.

Atmospheric transparency: Atmospheric transparency is how far one can see through the atmosphere. The transparency of the atmosphere depends on the atmospheric absorption rate. The absorption rate is the rate at which X-rays, ultraviolet, and infrared radiation emitted by the sun are absorbed into the atmosphere.

IV. Derived / Calculated Parameters

Derived or calculated parameters include Concentration Ratio, Optical Efficiency and Thermal Efficiency, explained as follows:

Concentration Ratio : The term "concentration ratio" is used to describe the amount of light energy concentration achieved by a given collector. Two different definitions of concentration ratio are in general use, briefly defined as follows:

- Optical Concentration Ratio (CR_o) : The averaged irradiance (radiant flux) (I_r) integrated over the receiver area (A_r), divided by the insolation incident on the collector aperture [25].

$$\text{CR}_\text{o} = \frac{1/A_r \int I_r dA_r}{I_o} \tag{14}$$

- Geometric Concentration Ratio (CR_g) : The area of the collector aperture A_a divided by the surface area of the receiver A_r [25].

$$CR_g = A_a/A_r \tag{15}$$

Optical concentration ratio relates directly to lens or reflector quality; however, in many collectors the surface area of the receiver is larger than the concentrated solar image.

Thermal Efficiency: The collector thermal efficiency is defined as the ratio of the useful energy delivered to the energy incident on the concentrator aperture [26], i.e.

$$\eta_c = Q_u / Q_s \tag{16}$$

Considering that the dish concentrator has an aperture area A_a and that it receives solar radiation at the rate Q_s from the sun, as shown in fig. 3, the net solar heat transferred Q_s is proportional to A_a, and the direct normal insolation (DNI) per unit of collector area I_s, which varies with geographical position on the earth, the orientation of the dish concentrator, meteorological conditions and the time of day. Considering the system is in steady state, we have,

$$Qs = I_s A_a \tag{17}$$

Under steady state conditions, the useful heat delivered by a solar collector system is equal to the energy absorbed by the heat transfer fluid, which is determined by the radiant solar energy falling on the receiver minus the direct or indirect heat losses from the receiver to the surroundings, i.e.

$$Q_u = Q_r - Q_l \tag{18}$$

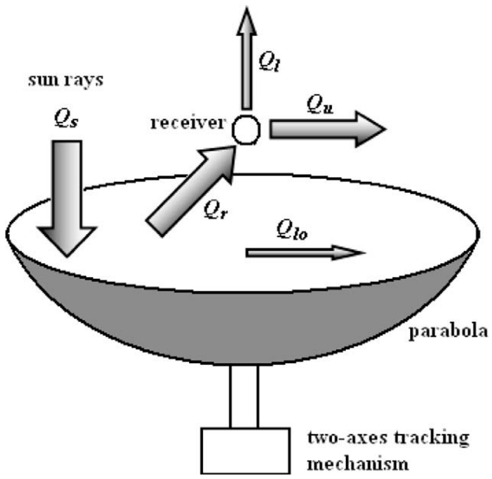


Fig. 3 : The Dish Concentrator Thermal Efficiency.

The radiation falling on the receiver Q_r is a function of the optical efficiency η_o, which is defined as the ratio of the energy falling on the receiver to the energy incident on the concentrator's aperture. And the receiver efficiency η_r is defined as the ratio of the useful energy delivered to the energy falling on the receiver [26], i.e.

$$\eta_o = Q_r / Q_s \tag{19}$$

$$\eta_r = Q_u / Q_r \tag{20}$$

Combining equations (16) to (20), the collector thermal efficiency can be written as,

$$\eta_c = \frac{Q_u}{Q_s} = \frac{Q_r}{Q_s} \cdot \frac{Q_u}{Q_r} = \eta_o \cdot \eta_r = \eta_o \cdot \frac{Q_r - Q_l}{Q_r} = \eta_o \left(1 - \frac{Q_l}{Q_r}\right)$$
$$= \eta_o \left(1 - \frac{Q_l}{\eta_o Q_s}\right) = \eta_o - \frac{Q_l}{Q_s} \tag{21}$$

Here, Q_l is the total heat loss rate of the receiver. So, to determine the thermal efficiency of the collector, the optical efficiency η_o and the total heat loss rate of the receiver Q_l has to be obtained.

Optical Efficiency: The optical efficiency η_o depends on the optical properties of the materials involved (e.g. the reflectance of dish and the optical properties of the receiver glazing etc.), the geometry of the collector, and the various imperfections arising from the construction of the collector. It can be analyzed by identification of the different losses mechanisms [27, 28].

- Cosine loss — It is the quotient of total reflective area and its projected area, as seen from the sun.
- Shading loss — It is the part of the reflective area of the dish shadowed by the receiver, which is located at the focal point of the dish. But this shading can be kept a minimum if the dish aperture size is much larger than the receiver, generally, giving less than 1% shading [27].

- Reflectivity loss — It is the quotient of reflected energy and energy impinging on the reflective dish surface. A clean mirror made of low-iron glass with a silver back-coat should provide a reflectivity of 90–94% [27].
- Transmission–absorption loss — It is the energy lost through transmission of the reflected sunlight in the air from dish to the receiver, causing about 2–4% loss [27]; on the other hand, fraction of reflected radiation hitting the receiver cavity surface is absorbed. An ideally insulated, isothermal receiver, without a transparent window at the receiver entrance, can be considered as a blackbody with the effective absorptance of one [29, 30]. In this study, a transparent window is placed at the receiver entrance to block off dust and wind, hence the effective absorptance of the receiver is assumed to have about 5–8% loss. Hence, these would cause a total of 7– 12% loss.
- Spillage loss — It is the fraction of radiation arriving outside the entrance aperture of receiver and may cause about 1–3% additional loss [9].

Therefore, the following equation form can be used to perform an approximated optical efficiency analysis,

$$\eta_o = \Lambda\rho\tau\alpha\gamma\cos\theta \quad (22)$$

As the solar parabolic dish concentrator maintains its optical axis always pointing directly towards the sun to reflect the beam, which means the incidence angle of solar beam into the dish is zero degree, and the cosine loss equals zero [31]. Thus, equation (22) can be written as

$$\eta o = \Lambda\rho\tau\alpha\gamma \quad (23)$$

Based on the analysis above, the optical efficiency used in the current study, including all of these losses can be assumed to be a value between 0.83-0.85 [5, 27, 28].

The Total Heat Loss Rate of the Receiver Q_l : The total heat loss rate of the receiver Q_l includes three types of losses, as shown in fig. 4 :

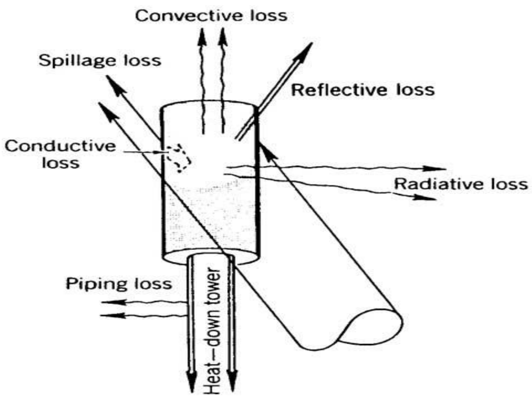


Fig. 5 : Various losses in the receiver.

- Convective heat loss through the receiver aperture, Q_{lc} : Convective heat loss of a solar collector receiver is proportional to the surface area of the absorber or receiver, and the difference in temperature between the absorber surface and the surrounding air.

There are two types of convection loss inside the receiver :

1. Natural convection loss : Natural convection loss inside the receiver cavity occurs due to the density difference of hot air inside the receiver cavity and ambient air as shown in fig. 5.

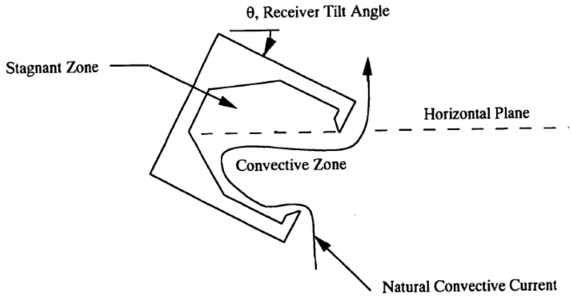


Fig. 5 : Natural convection loss inside receiver.

Calculations involved are [32]:

$Q_{lc,nat} = h_{int,nat}\, A_{int,cav}\, (T_{cav} - T_{amb})$ (24)

Where , $h_{int,nat} = Nu_L k_{air} / L$ (25)

Now, $Nu_L = 0.088 Gr_L^{1/3} (T_{cav} / T_{air})^{0.18} \cos(\Theta)^{2.47} (d_{ap}/d_{cav})^m$ (26)

Where $m = 1.12 - 0.982(d_{ap}/d_{cav})$

$GrL = g\, \beta\, (T_{cav} - T_{amb}) L^3 / v^2$ (27)

All properties are taken at film temperature, i.e., the average of receiver's cavity temperature and ambient temperature.

2. Forced convection loss : Forced convection losses occur due to wind flowing through the receiver. Now the direction of wind flowing through the receiver is possible in two ways as shown in fig. 5 [32].

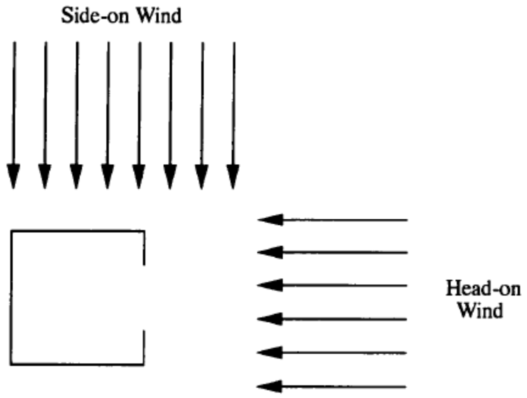


Fig. 5: Effect of forced convection loss on receiver.

Head on wind flowing into the receiver cavity will be responsible for forced convection loss inside the cavity. Side on wind on the receiver wall will carry away the convection loss which is transferred to the cavity wall due to conduction of heat from cavity to the outer surface of the receiver wall. This will be the forced convection loss outside the receiver.

So forced convection heat loss inside the receiver cavity is given as [32]:

$Q_{lc,for} = h_{int,for}\, A_{int,cav}\, (T_{cav} - T_{amb})$ (28)

$h_{int,for} = f(\Theta) V^{1.401}$ (29)

Here, $f(\Theta) = 0.163 + 0.749\sin(\Theta) - 0.502\sin(2\Theta) + 0.327 \times \sin(3\Theta)$ and V is the wind velocity (m/s)

So total convection loss inside the receiver cavity is :

$Q_{conv,total} = Q_{lc,nat} + Q_{lc,for}$ (30)

Major factors affecting convection loss:

1. Velocity of flowing air (m/s) : The forced convection losses in the receiver will greatly depend upon the limit of wind speed flowing at the place. It is seen that losses upto the wind speed of 18 mph is almost negligible. Forced convection losses come into effect after this considerable wind speed and can further be involved in the calculations.
2. Angle of inclination of receiver from horizontal (θ) : Fig. 6 shows the angle of inclination of receiver from the horizontal. It can be seen that as the tilt angle approaches 0 degree i.e. the receiver is aligned parallel to the horizontal, there will be maximum natural convective loss. And when the receiver is at right angles to the surface no convective loss will be there. At 0° tilt angle (receiver facing wind), natural convective currents occupy most of the receiver, with air temperature being lowest at the bottom and hotter as it rises and picks up heat. The hottest air is in the stagnant zone above the top of the receiver aperture since the air there has nowhere to escape. As the receiver is tilted downward, the stagnant zone becomes larger, which results in an overall increase in receiver average air temperature. With the aperture facing straight down, the entire cavity is a stagnant zone, resulting in the highest receiver average air temperature. It is the increase in the size of the stagnant zone and the resultant increase in average air temperature that causes natural convective heat loss to decrease as the receiver is tilted downward. Fig. 7 sows the effect of angle of inclination of receiver on natural convection loss.

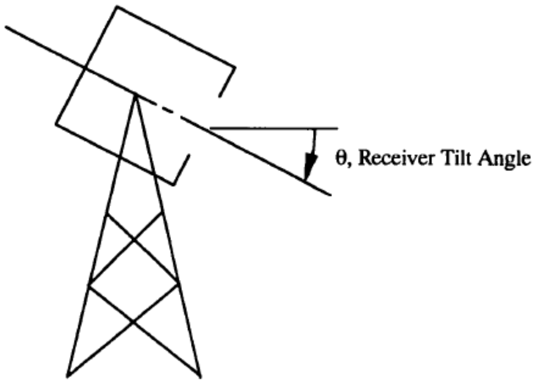


Fig. 6: Angle of inclination of receiver

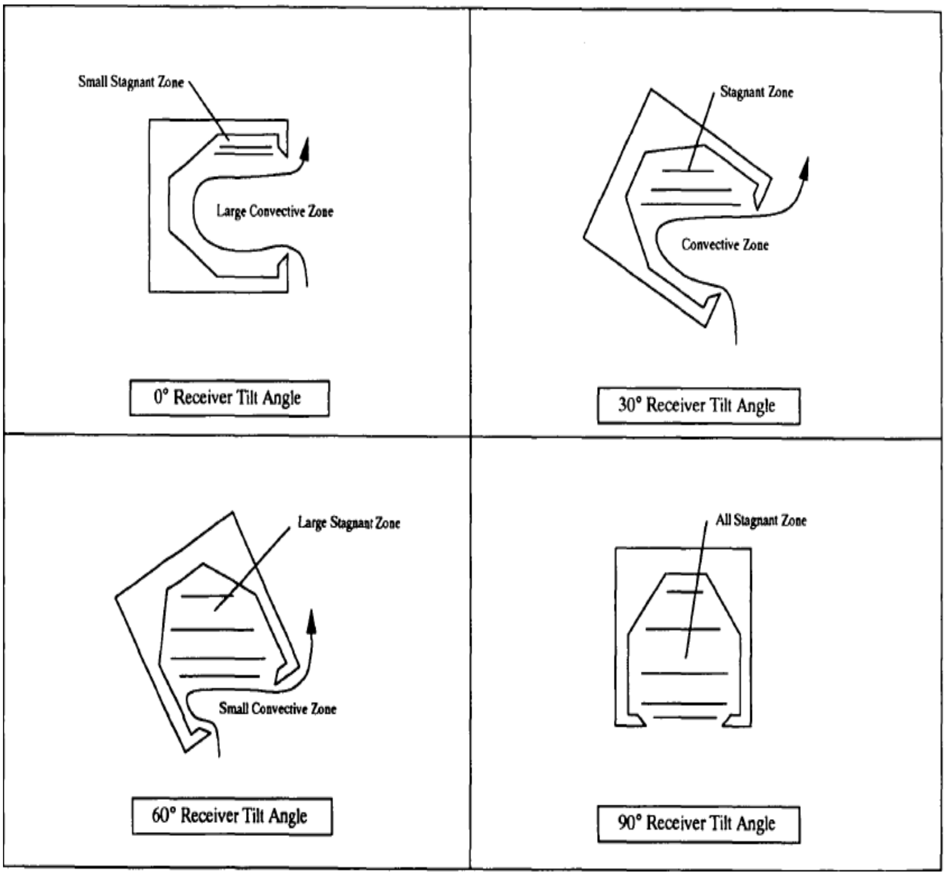


Fig. 7 : Effect of angle of inclination of receiver on natural convection loss.

3. Effect of head on wind on forced convection loss inside the receiver : At a receiver tilt angle of 0°, a head-on wind does not alter internal air temperatures very much from that resulting from natural convection. It seems that wind blowing directly into the aperture does not induce significant air currents inside the receiver, For a receiver tilted partially downward, but not straight down, the effects of head on wind appear to be dependent upon wind speed. Low-speed head-on wind appears to result in air circulation mainly in the lower portion of the receiver, while higher-speed wind results in air circulation throughout a larger portion of the receiver. At a receiver tilt angle of 90°, head-on and side-on winds are essentially the same. The wind velocity is parallel to the receiver aperture. At low wind speeds, the effects of wind are only felt in the lower portion of the receiver. However, at high wind speeds, wind effects are felt everywhere in the receiver. Fig. 8 shows effect of head on wind on forced convection loss inside the receiver.

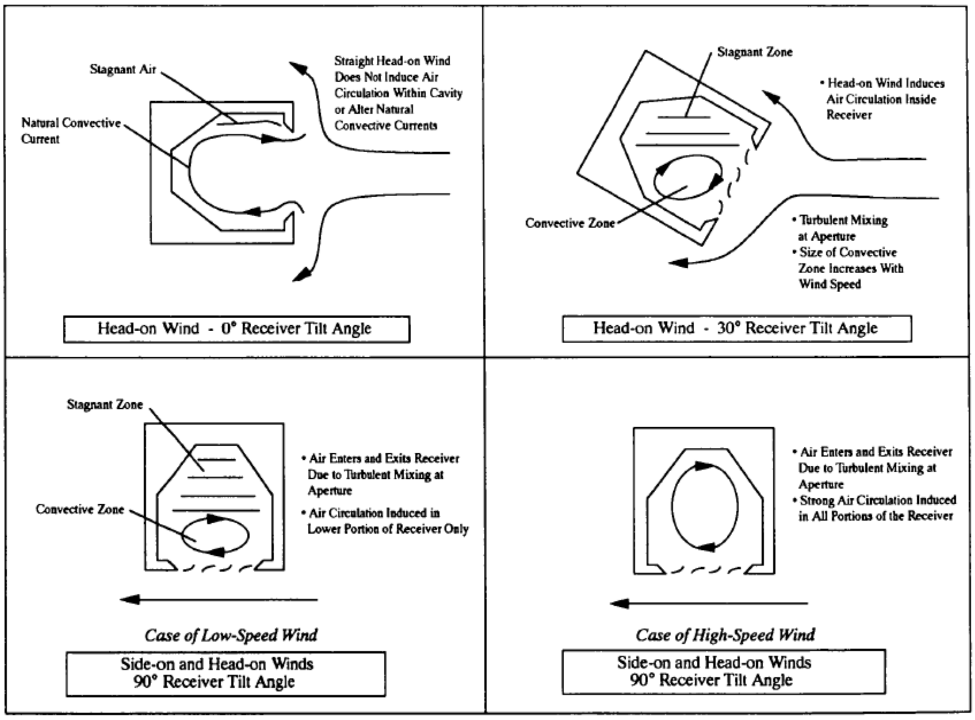


Fig. 8 : Effect of head on wind on forced convection loss inside the receiver.

- Radiative heat loss through the receiver aperture, Q_{lr} : Radiation heat loss is important for collectors operating at temperatures only slightly above ambient, and becomes dominant for collectors operating at higher temperatures. The rate of radiation heat loss is proportional to the emittance of the surface and the difference in temperature to the fourth power. The radiation heat loss is given by [33] :

$$Q_{lr} = A_{int,cavity}\, \mathcal{E}_{eff}\, \sigma\, (T_{cav}^4 - T_{amb}^4) \tag{31}$$

$$\text{Here, } \mathcal{E}_{eff} = 1 / [1 + (\frac{1}{\mathcal{E}c} - 1)\frac{Arec,aper}{Acav}] \tag{32}$$

- Conductive heat loss from receiver, Q_{lk} : The final mode of heat loss to consider in collector design is heat conduction. This is generally described in terms of a material constant, the thickness of the material and its cross-section area.

  The conduction heat loss in the cavity is transferred through the walls of the cavity and is then transferred to the environment by convection and radiation.

$$Q_{cond} = \frac{(Tcav - Tamb)}{L/Krec\ Aint,cav + \frac{1}{(hext,cav\ Aext,cav)}} + Aext,cav\ \mathcal{E}\ \sigma\ (T_{rec}^4 - T_{amb}^4) \tag{33}$$

The total convection heat transfer coefficient involves the natural and forced convection, as shown in fig. 9.

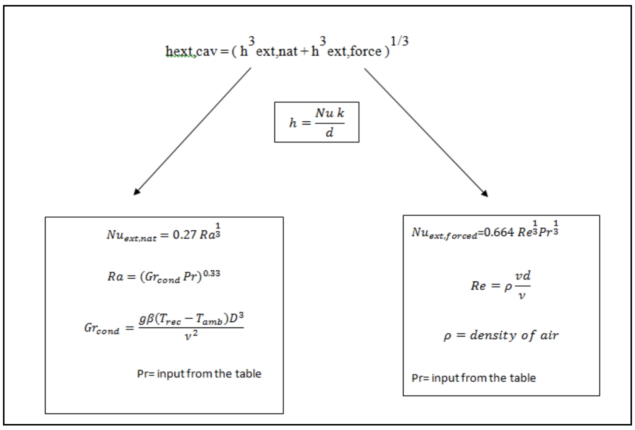


Fig. 9 : Convection heat transfer coefficients for conducted heat dissipated from receiver.

The total heat loss rate Q_l can be expressed as:

$Q_l = Q_{lr} + Q_{lc} + Q_{lk}$ (34)

V. Experimental Errors

It is found that independent errors in the conformity of the mirror to a true parabolic shape occur in a PDSC. Typically, these errors are assumed to be random and are reported in standard deviation units. Their combined effect is statistically determined. Although the sun's intensity is not normally distributed across its disc, it may be approximated as a standard distribution so that it can be treated similarly to the concentrator errors. The errors can be [24] :

- One dimensional errors : One-dimensional errors are those errors that contribute to beam spreading in the plane of curvature. They include –

Slope error : Angular change due to segments of mirror.

Tracking error : Error between movement of mirror surface and receiver.

Receiver alignment error : Error arising if receiver is not aligned properly.

- Two-dimensional error : It occurs in parabolic troughs, when an incident ray does not lie within the plane of curvature. It is not applicable for PDSCs.
- Error for deviation of sun's rays angle that get capture in concentrator : This analysis is based on the fact that the sun's rays are not truly parallel. Due to the finite angular size of the sun's disc (about 33 minutes of arc or 9.6 mrad), the sun's rays reaching the concentrator are not parallel [24]. This forms an image of the sun rather than a point leading to spreading of beam on receiver.

Total error will be given by [24] :

$$\sigma_{tot} = \sqrt{\left(\sigma_{1D}^2 + \sigma_{2D}^2\right)} \qquad (mrad) \tag{35}$$

VI. Performance Assessment Model For PDSC

Using the above methodology, a simple excel program was prepared for computing the predicted and actual efficiency of the parabolic dish solar collectors. Fig. 10 shows a snapshot of that program. Here, the predicted efficiency (Ideal Efficiency) is worked out on the basis of the above model, using the following equation :

$\eta_{ideal} = Q_r - (Q_{cond} + Q_{conv} + Q_{rad}) / Q_s$ (36)

The actual efficiency of the concentrator system is the ratio of heat gained by the fluid to the total energy available from the sun.

Now the heat gained by the fluid will be :

$Q_{useful} = m\ Cp\ (T_{out} - T_{in})$ (37)

So,

$\eta_{actual} = Q_{useful} / Q_s$ (38)

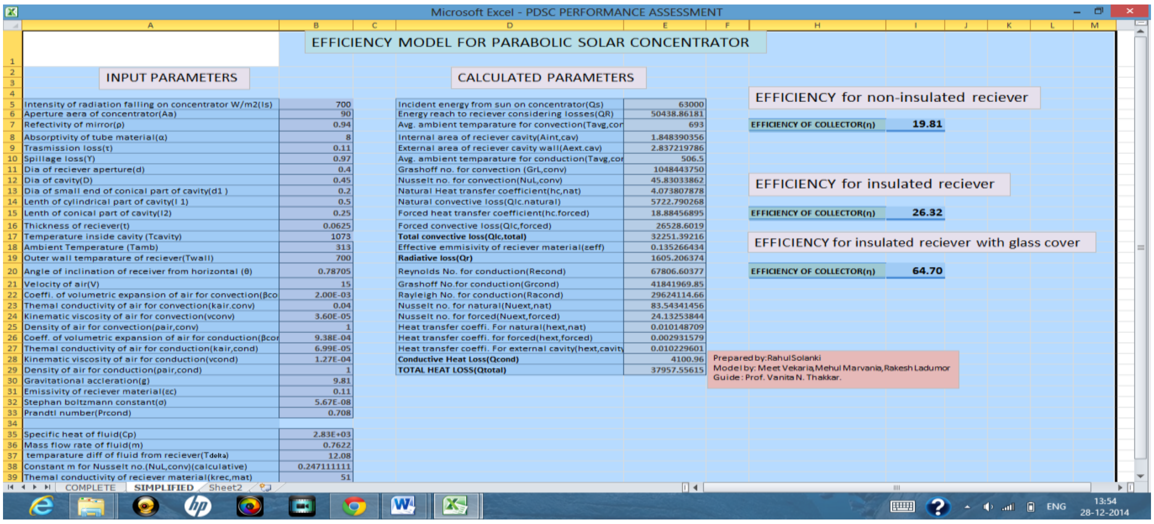


Fig. 10 : Screen shot of the Performance Assessment Model in MS Excel

Also, if we place a glass cover below the receiver cavity then no wind will be able to circulate inside the cavity so total convection loss inside the cavity can be neglected, but due to glass cover, the concentrated radiation need to pass through that glass cover and so its intensity will be reduced as glass won't allow full radiation to pass through it.

According to the suitability, a suitable glass cover having appropriate, optimal thickness can be used in order to ensure increase in efficiency.

If more thick glass will be used then its transmittance will be low and efficiency will decrease while if thin glass is used then there is a possibility of it to break. So, suitable optimized thickness glass is necessary.

As it is observed that convection loss will only be more at high wind velocities, so, we may even plan of an automated mechanism which places a glass cover automatically if wind velocity exceeds a set limit.

This aspect has also been considered and programmed, as can be seen in fig. 10.

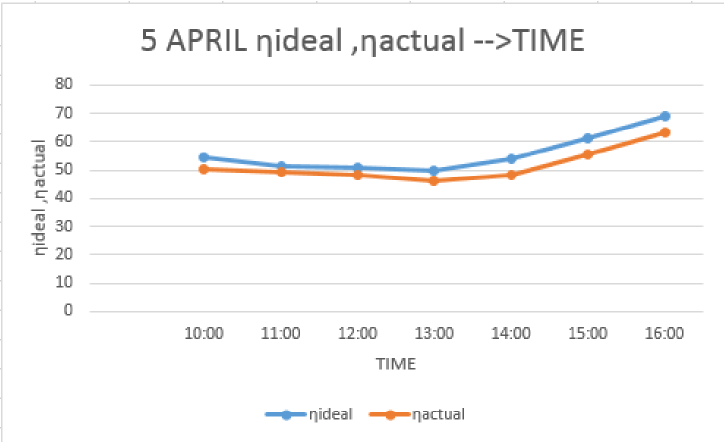
Using the above model, some graphs were plotted for the inputs obtained for 5th and 6th April, 2014. The sample tables and graphs are shown in fig. 11 and fig. 12.

EFFICIENCY TABLE

05-Apr-14	TIME	ηideal	ηactual	ηdiff
	10:00	54.68	50.08	4.6
	11:00	51.2	49.3	1.9
	12:00	50.62	48.05	2.57
	13:00	49.56	46.04	3.52
	14:00	53.82	48.02	5.8
	15:00	61.52	55.4	6.12
	16:00	69.26	63.32	5.94

06-Apr-14	TIME	ηideal	ηactual	ηdiff
	10:00	53.93	48.9	5.03
	11:00	49.2	46.88	2.32
	12:00	49.92	47.2	2.72
	13:00	49.24	47.4	1.84
	14:00	51.61	49.6	2.01
	15:00	61.8	58.99	2.81
	16:00	65.72	64.01	1.71

Fig. 11: The result table for Efficiency Vs. Time for 5th and 6th April, 2014.



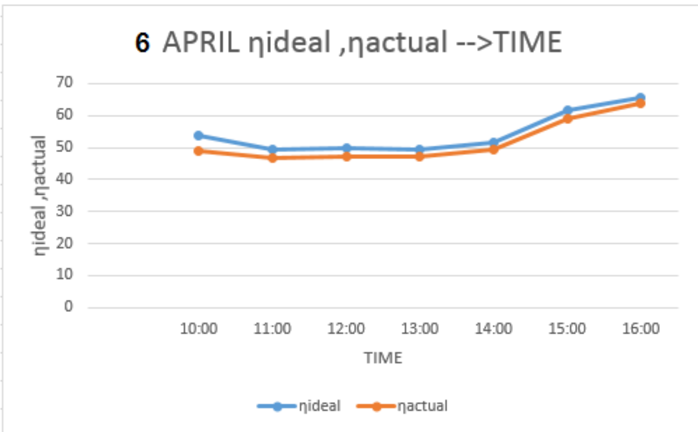


Fig. 12: The graph for Efficiency Vs. Time for 5th and 6th April, 2014.

VII. Conclusion

There is a good potential of use of Parabolic Dish Solar Collectors in process industries. Most of the work done on PDSCs is for its use in power generation. The present work presents a performance assessment model for PDSC for process heating application, using thermic fluid. Considering the effect of various system parameters, the expected and actual efficiencies are worked out. This can help in assessing the performance as well as detecting possible faults. It also gives an idea about losses or errors which have remained unaccounted, also providing a scope of their detection and tracing remedies for their removal / prevention. The suggestion and calculation of providing a glass cover on the receiver to minimize convective heat losses, which contribute maximum to the tally of heat losses from the receiver, can be implemented and experimented on, especially for systems working in high wind velocity terrains.

Further updates are possible in the model according to the requirements, specifications or constraints applicable to a specific system.

Also, only preliminary testing of the analysis model has been done. It can be useful and be improved by subjecting it to detailed experiments on systems working on a regular basis. The methodology developed is to be applied to actual working of the PDSC systems installed at UML, Vadodara.

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