Heat Loss from an Open Cavity

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ABSTRACT

This report presents the results of an investigation into the heat-loss characteristics of a cavity-type receiver for a parabolic dish concentrating solar collector. The receiver is similar to the type used in the Solar Total Energy Project in Shenandoah, Georgia. This investigation examines the effects of aperture size, orientation, and operating temperature on the heat loss of the receiver. The total receiver heat loss is quantitatively separated into its three modes: radiative, conduction, and convection. The testing was performed in a controlled environment, thereby eliminating any potential wind contribution. It was executed off flux, i.e., with no incident insulation. The receiver was operated in reverse of its typical operating configuration, whereby the heat-transfer fluid was heated externally. Previous heat-loss models or correlations with similar cavity receivers are compared with the experimental results from this study. A convective heat-loss correlation is presented from these experimental results. A theoretical model for the radiative heat loss is developed and compared with two methods used to quantitatively determine the radiative component of total heat loss.
ACKNOWLEDGMENTS

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<td>ADC</td>
<td>analog to digital converter</td>
</tr>
<tr>
<td>AC</td>
<td>alternating current</td>
</tr>
<tr>
<td>HTF</td>
<td>heat transfer fluid</td>
</tr>
<tr>
<td>N</td>
<td>number of sectioned elements of the receiver cavity internal surface area</td>
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<td>STEP</td>
<td>Solar Total Energy Project</td>
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<tr>
<td>TC</td>
<td>thermocouple</td>
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I. INTRODUCTION

Cavity type receivers are used extensively in concentrating solar thermal energy collecting systems. The Solar Total Energy Project (STEP) in Shenandoah, Georgia is a large scale field test for the collection of solar thermal energy. The STEP experiment consists of a large field array of solar collectors used to supplement the process steam, cooling and other electrical power requirements of an adjacent knitwear manufacturing facility.

The main components of each collector are the concentrator, the tracking mechanism, and the receiver (Fig. 1). The concentrator is a 7 meter diameter parabolic dish with a highly reflective coating on the inside surface. Each collector has two axes of rotation for tracking the sun all throughout the days of the year. The collectors are subjected to continuous changes in ambient conditions such as wind, solar insulation, and ambient temperature.

These environmental variations, as well as changes in receiver tilt, affect the overall receiver performance. The receiver used in this study is a parabolic collector.
A thorough understanding of receiver heat loss characteristics is essential for future development of solar receivers, and optimization of the STEP system presently in use. The system boundary of the receiver is defined as the outer skin of the receiver and the aperture opening (Fig. 2). The portion of the boundary formed by solid walls are only subject to conductive heat transfer. The aperture, however, is subject to convective, conductive and radiative heat transfer. Mass transfer occurs across the aperture and through the heat transfer fluid lines crossing the system boundary.

![Diagram of heat transfer system boundary](image)

**Figure 2. Heat transfer system boundary**

Analytical methods for predicting the conductive and radiative heat losses from a cavity receiver are fairly straightforward. This, however, is not the case for convective heat loss analysis. The complex geometry of the cavity makes it difficult to use existing analytical models for predicting convective heat loss. Few convective heat loss correlations, for cavity receivers, exist due to the lack of significant empirical data. Correlations for receivers with simple geometry are not considered valid for this receiver.
An extensive search of current literature produced only a few studies on heat loss from cavities. A study was performed by LeQuere, Penot, and Mirenayat in which heat loss characteristics of two different sized cubical cavities were examined. They considered variations in receiver operating temperature and angle, in their study. A study performed by Koenig and Marvin, presented by Harris and Lenz, gave an empirically derived correlation for convective heat loss from cylindrical cavity type receivers, including the effects of variation in operating temperature and angle. An analytical model for convective heat loss for an open cubical cavity receiver was presented by A. M. Clausing. The Clausing model was developed for a central receiver operating at much higher temperatures than the receiver studied here. Siebers and Kraabel presented a model for the convective heat loss from a central cylindrical cavity receiver.

There is some experimental data available for this type of receiver from previous tests on off-flux field measurements conducted with limited instrumentation at STEP. Field measurement experiments, such as the one conducted at STEP, provide no control over environmental conditions such as wind, and ambient temperature. In order to control the environmental conditions, receiver testing for this study, took place indoors.

The purpose of the tests, conducted for this study, was to isolate and quantify the radiative, conductive, and convective components of total heat loss, and to determine the effects of operating temperature, receiver angle, and aperture size on cavity heat loss. An analytical model for radiative heat loss was developed and compared with two other methods used to determine radiative heat loss. A proposed convective heat loss correlation, including effects of aperture size, receiver operating temperature, and receiver angle is presented. The resulting data is a source to evaluate the STEP measurements.
II. APPARATUS

A. Receiver

A drawing of the receiver studied in this work is shown in cross section in Figure 3. The cavity of the receiver is composed of a single tube wound in a conical frustum-cylinder shape with the aperture at the cylindrical end of the tube bundle. The tube bundle is wrapped in a thick blanket of Kaowool® insulation. The outer skin of the receiver is formed by a single cylindrical wrap of sheet metal. The outer skin extends beyond the aperture face to serve as a wind break. The flow lines to and from the receiver are heavily insulated. The inlet and outlet lines for these tests, as shown in Figure 3, are the reverse of those for an on-flux receiver in field operation.

![Cavity receiver cross section](image)

The receiver is cradled in a frame that allows it to rotate 180° (Fig. 4). The receiver can be fixed at 15° increments from -90°, where the aperture is upward, to +90°, where the aperture is downward (Fig. 5). The high pressure flexible lines on the sides of the receiver
test stand allow the receiver to rotate freely for each test position.

Thermocouples were used to measure the receiver inlet and outlet temperatures. Two in-house calibrated K-type thermocouples were immersed in each of the heat transfer fluid (HTF) inlet and outlet lines of the receiver. One of the thermocouples from the inlet and outlet lines measured absolute temperature. The remaining two thermocouples were connected in series to yield a direct measure of the temperature difference between the inlet and the outlet. The receiver was further instrumented with seventeen surface thermocouples and thirteen internal air thermocouples (Figure 6). The surface thermocouples were spot welded in place with the lead ends spaced approximately one-eighth inch apart.

Figure 4. Receiver test stand
**Thermocouple Identification**

- A: air TC
- T: tube surface TC
- S: outer skin surface TC
- C: back cover surface TC

**TC/Switch Correlation**

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**Figure 5. Receiver angles**

**Figure 6. Receiver thermocouples locations**
An insulating annulus with a corresponding center plug were fabricated for each of the aperture sizes tested (Fig. 7). The aperture sizes tested were 6 inch diameter, 12 inch diameter, 18 inch diameter, and 26 inch diameter. The 18 inch diameter aperture is the size presently being used for this model of receiver. The 26 inch diameter, the internal diameter of the cavity, aperture leaves no lip to the cavity. The aperture plug and annulus were fabricated of 1 inch thick solid insulation boards. The plug was tapered inward to fit snugly in the respective aperture annulus. The plugs were held in place by two wooden straps extending across the aperture end of the receiver.

![Annulus and plug](image)

**Figure 7. Annulus and plug**

**B. Flow Loop**

The heat transfer fluid was heated by a heating and pumping station adjacent to the receiver (Fig 8). The flow system consists of two parallel flow loops- the primary heating loop and the receiver feed loop. The primary heating loop has available two in-line 12 kW electric heaters. A minimum flow rate of 5 gpm was maintained in the heating loop to prevent excessive film temperatures of the HTF in the heaters. The receiver feed loop was throttled for a flow rate through the receiver of 1 gpm. The HTF flow rate through the receiver was measured using three turbine flow meters connected in a series on the inlet line to the receiver. The flow meters are located in a straight section of piping allowing ample upstream and downstream damping lengths. Three flow meters are used for measurement redundancy. One flow meter was factory calibrated. The remaining two flow meters were
calibrated against the factory calibrated flow meter.

Figure 8. Heat transfer fluid heating system
C. Radiometer Setup

One method used to determine the thermal radiative heat loss from the aperture was with a radiometer. The radiometer was mounted on a tripod and placed directly below and centered on the receiver. The radiometer was water cooled and the return water temperature was monitored. The radiometer signal output leads were connected to the computer data acquisition system. A Saran Wrap window assembly was positioned over the radiometer in place of the quartz window bezel. The radiometer window serves to isolate the temperature sensitive thermopile surface from localized convective currents. This is discussed further in radiometer calibration section.

D. Data Acquisition

A computer was used for data acquisition and display. This consisted of an Apple IIe computer with two 16-channel data acquisition cards each connected to a thermocouple cold junction terminal box. The following transducers were connected to the data acquisition system; receiver inlet and outlet thermocouples, two ambient thermocouples, three receiver flow meters; radiometer; radiometer water thermocouple; loop inlet and outlet thermocouples; and the main flow loop flow meter. The surface and air thermocouples on the receiver were connected to a digital display unit via a multi-channel thermocouple switch. The surface and air receiver temperature readings were recorded manually when steady state conditions were obtained for each test point.

III. TEST METHOD

The testing was conducted in two phases. Phase One of the testing examined the effects of receiver operating temperature and receiver angle on the receiver heat loss. Phase Two looked at the effects of receiver aperture size on the receiver heat loss. After Phase One, and before Phase Two, the receiver was overhauled. The overhaul of the receiver included
replacing the insulation in the walls, repainting the tube surfaces, and resealing the seams and ports through the outer skin of the receiver.

**A. Temperature and Receiver Angle Effects**

The model of the effects of temperature and receiver angle on heat loss developed in this study utilizes results from both Phase One and Phase Two tests. The overlap portion of the Phase One and Phase Two tests were compared for ease of repetition. The four nominal operating temperatures in Phase One testing were 300°F, 400°F, 500°F, and 600°F with the standard 18 inch aperture. The Phase Two study was conducted at operating temperatures of 400°F and 600°F and incorporated 6, 12, 18, and 26 inch apertures. The nominal operating temperature was based on the average of the inlet and outlet temperatures. For each operating temperature the receiver was tested at ten angles from -90° to +90° in Phase One. The receiver angles tested were from 0 to +90°, every 15° and from 0 to -90°, every 30°. Phase Two included receiver angles from 0° to +90°, every 15°. More angles were tested in the positive range, since most cavity receivers operate in this range. The negative angles were tested to provide a clearer understanding of heat loss characteristics, as a function of receiver angle. However, some concentrating solar collector designs operate with inverted cavity receivers.

At the beginning of each test, the receiver was placed in the +90° position with the aperture facing down. The flow system was started and the fluid was heated as close to the operating temperature as possible. Obtaining the nominal HTF temperature at the receiver inlet usually meant setting the heater temperature 25°F to 50°F higher, accounting for heat loss from the connecting line. The flow rate through the receiver was adjusted to approximately one gallon per minute. The one gallon per minute flow rate is typical for these type of receivers at STEP. Thermal stabilization is attained when there is less than 0.1 degree change in the inlet and outlet temperatures over the two minute data sampling.
interval. Reaching thermal stabilization often takes several hours depending upon the nominal operating temperature and the ambient laboratory conditions. When thermal stabilization was reached, temperature and flow measurements were recorded.

The convective heat loss from the cavity of the receiver is assumed to be negligible when the receiver is in the +90° position. This assumption is supported in two ways. First, smoke flow visualization techniques revealed negligible air flow across the aperture with the receiver in the +90° position. The second is the minimal variance between experimentally determined radiative heat loss at +90° and the calculated values and radiometer measurements.

Therefore, the total heat loss from the receiver in the +90° position is composed of radiative losses from the aperture and conductive losses through the side, back, and annulus walls of the receiver. The aperture of the receiver was then fitted with an insulated plug to eliminate the radiative component from the total heat loss. With the plug in place, the system was again allowed to reach steady state and again the temperature and flow measurements were recorded. In this manner, the radiative and conductive components of the total heat loss from the receiver for a particular operating temperature were isolated. The radiative and conductive heat losses are assumed constant with receiver angle.

The receiver was then positioned in the +75° attitude and the system allowed to stabilize. The temperature and flow measurements were again recorded. The total heat loss measurements were normalized linearly to the nominal test temperature for comparison purposes. This accounted for the variation in the operating temperature from one test setup to the next. The thermal stabilization procedure was repeated for each receiver angle tested. The entire procedure was repeated for each nominal operating temperature test point.
The total heat loss from the receiver can be expressed as a mathematical relation. The preceding appears as:

\[ q_{\text{total}} = \dot{m} \ C_p \ (T_{\text{in}} - T_{\text{out}}) \]  \hspace{1cm} (1)

where:

\[ T_{\text{in}} = \text{temperature of fluid at the inlet to the receiver} \ [\text{K}] \]
\[ T_{\text{out}} = \text{temperature of fluid at outlet to receiver} \ [\text{K}] \]
\[ \dot{m} = \text{mass flow rate of the fluid} \ [\text{g/m}] \]
\[ C_p = \text{specific heat of the fluid} \ [\text{K}] \]

The mass flow rate is given by:

\[ \dot{m} = \dot{v} \rho \]  \hspace{1cm} (2)

where:

\[ \dot{v} = \text{volumetric flow rate} \]
\[ \rho = \text{heat transfer fluid density at the inlet temperature} \]

The conductive heat loss is given by:

\[ q_{\text{conductive}} = q_{\text{plugged @+90°L}} \]  \hspace{1cm} (3)

The radiative heat loss is given by:

\[ q_{\text{radiative}} = q_{\text{unplugged @+90°L}} - q_{\text{plugged @+90°L}} \]  \hspace{1cm} (4)

The convective heat loss at any angle, \( \alpha \), is then given by:

\[ q_{\text{convection @} \alpha \text{L}} = q_{\text{total @} \alpha \text{L}} - q_{\text{conduction}} - q_{\text{radiative}} \]  \hspace{1cm} (5)

The HTF volumetric flow rate, inlet temperature, outlet temperature, and temperature difference were all measured with transducers. The density and heat capacity were both calculated as functions of temperature.

**B. Aperture Size Effects**

The second Phase of the testing examined the effects of receiver aperture size on the
receiver heat loss. This Phase of the testing was performed at two operating temperatures of 400°F and 600°F, and seven receiver angles from 0 to +90° at 15° increments. The four aperture sizes tested were 6 inches, 12 inches, 18 inches, and 26 inches. The 18 inch diameter aperture is the standard size for this receiver. The Phase Two testing followed the same procedure used in Phase One testing with the additional step of changing aperture size. Heat loss values from Phase One were compared with Phase Two.

V RESULTS
A. Temperature and Angle Effects on Total and Convective Heat Losses

The data summary of the receiver operating temperature and receiver angle effects from the Phase One tests are tabulated in Appendix 1. These results were first presented by Stine and McDonald(8). Total receiver heat loss varies, approximately, linearly with operating temperature (Fig. 9).

![Figure 9. Total receiver heat loss versus operating temperature](image)
The total receiver heat loss varies non-linearly with receiver angle (Fig. 10). The total heat loss is at a minimum when the receiver aperture orientation is downward. This supports the assumption of negligible convective heat loss with the receiver in this position. The maximum heat loss occurs when the receiver aperture is oriented at approximately +45° above horizontal. This particular receiver would not normally operate at angles above horizontal as these are negative angles.

![Graph showing total receiver heat loss versus receiver angle for 18 inch aperture](image)

Figure 10. Total heat loss versus receiver angle for 18 inch aperture

Results from duplicate test conditions from Phase Two are also presented in Figure 10. As discussed previously, the convective heat loss through the aperture is determined by the difference between the total heat loss at any angle and the total heat loss for the +90° angle (Fig. 11). This requires that the convective heat loss be zero when the receiver is positioned at +90°, with the aperture down. The maximum convective heat loss occurred with the receiver in the -45° position. The reduction in total heat loss from Phase One to Phase Two...
testing, for receiver angles less than +60°, was assumed to be primarily due to the improved insulating properties of the cavity walls. The larger difference in the total receiver heat loss from Phase One to Phase Two for receiver angles greater than +60° resulted from the combined effects of improved insulation in the cavity walls and better sealing of the outer receiver skin. Apparently, sealing of the skin had little effect on heat loss from the receiver for angles less than +60°.

Figure 11. Convective heat loss for 18 inch aperture versus receiver angle

B. Aperture Size Effects on Total and Convective Heat Losses

The data summary of the Phase Two aperture size testing are tabulated in Appendix 2. These results were first presented by Stine and McDonald (8). The effect of aperture size on the receiver total heat loss is shown (Fig. 12) for an operating temperature of 600°F. At a receiver angle of 0°, the total receiver heat loss increases by a factor of three as the aperture diameter increases from 6 inches to 26 inches. The total heat loss at the +75° and +90°
receiver angles are approximately equal. This agrees with a previous study (Stein and McDonald) on the effects of receiver angle\(^{(6)}\). At these positions the total receiver heat loss increases by a factor of two when the aperture diameter increases from 6 inches to 26 inches for a 600°F operating temperature. At +45° the total heat loss increases approximately linearly with increase in aperture size. The receiver angle has less effect on the total heat loss for small apertures. The maximum variation due to receiver angle in the total heat loss is 0.4 kW for the 6 inch diameter aperture as compared to 1.2 kW loss with no aperture (26 inch aperture).

![Graph showing total heat loss vs aperture size](image)

**Figure 12. Total Receiver Heat Loss versus Aperture Size for 600°F**

The effect of aperture size on the convective component of the total heat loss is shown in Figures 13 and 14. Aperture size has a much greater effect for low receiver angles than high receiver angles. The results also showed the convective loss increased dramatically when the aperture increased from 6 inches to 18 inches. There was little change in the convective loss as the aperture increased beyond 18 inches.
Figure 13. Convective Heat Loss versus Aperture Size for 400°F

Figure 14. Convective Heat Loss versus Aperture Size for 600°F
C. Radiative and Conductive Heat Losses

In these tests, the radiative heat loss was determined from the difference between the plugged and unplugged values of the total heat when the receiver is in the +90° position. The radiative heat loss was assumed constant for all receiver angles. The conductive heat loss, through the receiver walls, was given by the total receiver heat loss when the receiver was in the +90° position with the aperture plugged. The conductive heat loss was also assumed constant for all receiver angles. The radiative and conductive heat losses from Phase One and Two of the testing were compared for repeatability (Fig. 15). Some differences were expected as a result of overhauling the receiver after the Phase One tests.

![Radiative and Conductive Heat Losses versus Receiver Temperature](image)

Figure 15. Radiative and Conductive Heat Losses versus Receiver Temperature
An infrared radiometer was another method used to determine radiative heat loss. The difference between radiometer readings taken with the receiver plugged and unplugged measured the radiative heat loss through the aperture. The two methods for determining the radiative heat loss are compared for operating temperatures of 600°F and 400°F (Figs. 16 and 17). The radiative heat loss increased approximately with the square of the aperture area. For each of the two nominal operating temperatures, the analytical heat loss was slightly higher. The difference between the analytical and experimental data, radiative heat loss, increased positively with aperture size. The disparity was accounted for via the estimated parameters used in analysis. These parameters include surface emissivity, temperature distribution, refractory and non-refractory. For small aperture diameters to cavity volume ratios, the cavity will radiate through the aperture essentially as a black body regardless of the emittances of the internal surfaces. However, as the ratio increases the cavity becomes less of a black body emitter. For large aperture diameter to cavity volume ratios, the emittances of the internal surfaces become critical in determining the cavity emittance. Also, for larger apertures the temperature difference between the inlet and outlet becomes significant. The fixed difference used in the analysis for all aperture sizes could affect the results. Further investigation is required to provide more accurate temperature distributions.
Figure 16 Radiative Heat Loss versus Aperture Size at 400°F

Figure 17 Radiative Heat Loss versus Aperture Size at 600°F
The total heat loss make-up, in terms of the radiative, conductive, and convective components, changes with aperture size. With a 6 inch aperture and the receiver at 45° the conduction loss forms about 65% of the total heat loss. With the receiver at a typical operating angle of 45° and an operating temperature of 600°F, radiative and convective percentages of total heat loss are approximately equal to an aperture of 26 inches (Fig. 18).

![Graph showing heat loss modes versus aperture size](image)

Figure 18. Percent Heat Loss Modes versus Aperture Size for 600°F at 45° angle

VI. PREVIOUS CAVITY CONVECTION LOSS MODELS

A. LeQuere, Penot, and Mirenayat Model

LeQuere, Penot and Mirenayat presented an experimental correlation for Nusselt number as a function of Grashof number. Their study used a cubical cavity typical of those used in central receiver systems. The study investigated varying receiver temperatures and angles. The model was developed for a maximum temperature difference between cavity walls and
the ambient, 230°F. Variation in receiver angles used in the model were from 0°, where the aperture is upward, to 180°, where the aperture is downward.

The cavity used in their testing was modular in design so that each panel could be heated independently. The panels were electrically heated. The electrical power and temperature of each panel were measured. Modular design allowed for local as well as global heat loss analysis. The total heat loss of the cavity was determined from the total electrical power used by all the panels. They determined the radiative component for total heat loss by summing the radiative heat loss of each panel to the cavity aperture.

$$q_{\text{radiative}} = \sum_{i} N_m \varepsilon \sigma S (T_{\text{panel}} - T_{\text{ambient}}) F_{i-o}$$

(6)

where:

- $\sigma$ = Stefan-Boltzmann constant [W/m²-K⁴]
- $\varepsilon$ = emissivity of each panel
- $S$ = panel surface area [m²]
- $T_{\text{panel}}$ = individual panel temperature [K]
- $T_{\text{ambient}}$ = ambient temperature [K] opening
- $F_{i-o}$ = view factor for a panel to cavity opening

$N_m$ = total number of panels that compose the cavity

The conduction heat loss component is determined by the total heat loss of a plugged cavity. By plugging the aperture of the receiver, the radiative and convective components of the total heat loss are eliminated.

The convective heat loss is determined by subtracting the conductive and radiative
components from the total heat loss.

\[ q_{\text{convection}} = q_{\text{total}} - q_{\text{conductive}} - q_{\text{radiative}} \]  

(7)

Their Nusselt number is given by:

\[ \text{Nu} = \frac{q_{\text{convection}} L}{S k (T_{\text{panel}} - T_{\text{ambient}})} \]

(8)

where:

- \( L \) = dimension of cavity aperture [m]
- \( k \) = thermal conductivity of air [W/m*K]
- \( S \) = total interior cavity surface area [m^2]

The Grashof number is given by:

\[ \text{Gr} = \frac{g \beta (T_{\text{panel}} - T_{\text{ambient}}) L^3}{\nu^2} \]

(9)

where:

- \( g \) = local gravitational acceleration [m/s^2]
- \( \beta \) = thermal expansion coefficient of air [K^{-1}]
- \( \nu \) = kinematic viscosity of air [m^2/s]

All fluid properties were evaluated at the ambient temperature. The experimental correlation for Nusselt number as a function of the Grashof number is given by:

\[ \text{Nu} = a \text{Gr}^b \]

(10)

The coefficient 'a' and the exponent 'b' are empirically derived and are both a function of receiver angle. The values of 'a' and 'b' are presented in Table 1 for receiver angles of interest in this study. Equation 10 is valid for a Grashof number between \( 10^7 \) and \( 5 \times 10^9 \).
Table 1.
Empirical correlation coefficients and exponents

<table>
<thead>
<tr>
<th>Receiver Angle (θ)</th>
<th>Coefficient (a)</th>
<th>Exponent (b)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-90</td>
<td>0.0570</td>
<td>0.353</td>
</tr>
<tr>
<td>-75</td>
<td>0.0470</td>
<td>0.360</td>
</tr>
<tr>
<td>-60</td>
<td>0.0545</td>
<td>0.360</td>
</tr>
<tr>
<td>-45</td>
<td>0.0465</td>
<td>0.370</td>
</tr>
<tr>
<td>-30</td>
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</tr>
<tr>
<td>-15</td>
<td>0.0465</td>
<td>0.368</td>
</tr>
<tr>
<td>0</td>
<td>0.0925</td>
<td>0.330</td>
</tr>
<tr>
<td>15</td>
<td>0.0810</td>
<td>0.331</td>
</tr>
<tr>
<td>30</td>
<td>0.0640</td>
<td>0.332</td>
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<tr>
<td>60</td>
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<td>0.292</td>
</tr>
<tr>
<td>75</td>
<td>0.0330</td>
<td>0.302</td>
</tr>
<tr>
<td>90</td>
<td>NA</td>
<td>NA</td>
</tr>
</tbody>
</table>

Consideration must be given to the difference in the cavity geometry from receiver used by LeQuere, Penot and Mirenayat and that used in this study. The receiver used by LeQuere, Penot and Mirenayat was cubical whereas the receiver used in this test was cylindrical and conical. The LeQuere, Penot and Mirenayat receiver aperture is the same as the characteristic interior dimension. They did not study the effect of varying aperture sizes.

LeQuere, Penot and Mirenayat modeled convective heat loss through the aperture is given by:

\[
q_{\text{convective}} = hA(T_{\text{cav}} - T_{\text{ambient}})
\]  

(11)

where:

- \( h \) = convective heat transfer coefficient [w/m²K]
- \( A \) = total interior cavity surface area [m²]
- \( T_{\text{cav}} \) = area average cavity surface temperature [K]
- \( T_{\text{ambient}} \) = ambient air temperature [K]
The convective heat transfer coefficient is given by:

\[ h = \frac{k \text{Nu}}{L} \]  

where:

- \( L \) = dimension of cavity aperture [m]
- \( k \) = thermal conductivity of air [W/m-K]
- \( \text{Nu} \) = Nusselt number

A BASIC computer language program was written to solve for the convective heat loss from the receiver used in this study applying the LeQuere, Penot, and Mirenayat model. For a listing of the computer program see Appendix 3. The results are presented in Appendix 4. The variations in the convective heat loss with receiver angles for operating temperatures from 300°F to 600°F and aperture sizes from 6 inches to 26 inches are presented (Fig. 19).

![Figure 19. Convective heat loss for LeQuere, Penot, and Mirenayat model](image)
B. Koenig and Marvin Model

The Koenig and Marvin model for predicting convective heat loss from a cavity receiver is presented by Harris and Lenz.\(^{(10)}\) Their model is based on operating temperatures between 550°C(1022°F) and 900°C(1652°F) for an on-flux analysis. The operating temperature range used by Koenig and Marvin is considerably higher than any receiver temperature tested in this study. The Koenig and Marvin receiver was designed to operate at higher temperatures.

For the Koenig and Marvin model the convective heat loss through the cavity aperture is given by:

\[
q_{cav} = h A_T (T_{cav} - T_{amb})
\]

where:

- \(A_T\) = area of heat transfer tubing facing inside cavity [m\(^2\)]
- \(T_{cav}\) = inside cavity temperature or mean operating temperature [K]
- \(T_{amb}\) = ambient air temperature [K]
- \(h\) = cavity convective heat transfer coefficient [W/m\(^2\)K]

The heat transfer coefficient is given by:

\[
h = \frac{k \text{Nu}_{cav}}{L}
\]

where:

- \(k\) = thermal conductivity of air [W/m-K]
- \(\text{Nu}_{cav}\) = Nusselt number of the cavity
- \(L\) = characteristic length of the cavity [m]
The characteristic length of the cavity used by Koenig and Marvin is given by:

\[ L = \sqrt{2} \, R_{\text{cav},i} \]  \hspace{1cm} (15)

where:

\[ R_{\text{cav},i} = \text{mean inner cavity radius [m]} \]

The Nusselt number is given by:

\[ \text{Nu}_{\text{cav}} = 0.52 \, P(Q) \, \lambda_{c}^{1.75} (\text{Gr}_{L} \, \text{Pr})^{1/4} \]  \hspace{1cm} (16)

where:

\[ P(Q) = \text{is an expression that accounts for the effects of receiver angle} \]
\[ \lambda_{c} = \text{is an expression that corrects for aperture size} \]
\[ \text{Gr}_{L} = \text{Grashof number} \]
\[ \text{Pr} = \text{Prandtl number} \]

The receiver angle function is given by:

\[ P(\phi) = \cos^{3.2}\phi \hspace{1cm} \text{for} \hspace{0.5cm} 0^\circ \leq \phi \leq 45^\circ \]  \hspace{1cm} (17)

\[ P(\phi) = 0.707 \, \cos^{2.2}\phi \hspace{1cm} \text{for} \hspace{0.5cm} 45^\circ \leq \phi \leq 90^\circ \]  \hspace{1cm} (18)

where:

\[ \phi = \text{angle of cavity axis with the horizontal [degrees]} \]

The aperture size function is given by:

\[ \lambda_{c} = \frac{R_{\text{ap}}}{R_{\text{cav},i}} \]  \hspace{1cm} (19)

where:
\( R_{ap} = \text{cavity aperture radius [m]} \)

The Grashof number is given by:

\[
Gr_L = \frac{L^3 g \beta (T_{cav} - T_{amb})}{\nu^2}
\]  

(20)

where:

\( g = \text{gravitational acceleration [m/sec}^2] \)

\( \beta = \text{coefficient of volumetric expansion [K}^{-1} \)

\( \nu = \text{kinematic viscosity [m}^2\text{/sec]} \)

The volumetric expansion coefficient for air is calculated as:

\[
\beta = \frac{1}{T_{prop}}
\]  

(21)

where:

\( T_{prop} = \text{temperature at which the air properties are evaluated [K]} \)

The Koenig and Marvin air properties temperature is given by:

\[
T_{prop} = \frac{11}{16} T_{cav} + \frac{3}{16} T_{amb}
\]  

(22)

The thermal radiative losses through the cavity aperture from the hot interior surface is given by:

\[
\dot{q}_{\text{rad}} = \pi R_{ap}^2 e_\alpha \bar{\sigma} (T_{cav}^4 - T_{amb}^4)
\]  

(23)

where:
\[ \varepsilon_a = \text{emissivity of the aperture} \]
\[ \bar{\sigma} = \text{Stefan-Boltzmann constant [W/m}^2\text{-K}^4] \]
\[ \varepsilon_a = 0.9 \text{ for these studies} \]

The conduction heat loss through the walls of the cavity is given by:

\[ q_{\text{cond}} = \frac{k_i A_c (T_{\text{cav}} - T_{\text{amb}})}{t} \]  \hspace{1cm} (24)

where:

- \( A_c = \text{area for conduction through the cavity [m}^2\] \)
- \( k_i = \text{thermal conductivity of the cavity insulation [W/(m - K)]} \)
- \( t = \text{thickness of cavity insulation [m]} \)

The conduction area of the cavity for the receiver considered in this study is given by:

\[ A_c = \pi \left( \frac{R_{\text{cav},i} + R_{\text{cav},o}}{2} \right)^2 + 2 \pi \left( \frac{R_{\text{cav},i} + R_{\text{cav},o}}{2} \right) + \pi (R_{\text{cav},i}^2 - R_{\text{ap}}^2) \]  \hspace{1cm} (25)

where:

- \( R_{\text{cav},o} = \text{cavity outside radius [m]} \)

Application of the Koenig and Marvin model to the receiver under study here was accomplished using a BASIC computer language program. The program is listed in Appendix 5. The results of the Koenig and Marvin modeling are found in Appendix 6. Figure 20 summarizes the values in Appendix 6.
C. CLAUSING MODEL

The Clausing model of convective heat loss from cavities was developed for large central receivers as opposed to the small receiver, used in this study.\(^{11}\) The receivers utilized for the development of the Clausing model were simple in geometry with no curved surfaces. The Clausing model has been modified for application to the receiver provided in this study. The model was developed for on-flux mode of operation. For on-flux analysis the refractory surfaces are assumed to have a higher temperature than the active surfaces, whereas, for off-flux analysis the temperature conditions are reversed. Many of the temperature terms used in the Clausing model required modification to work for an off-flux situation.

Clausing's convective heat loss is based on an energy balance between the convective
energy loss within the cavity, $q_c$, and the energy transported through the aperture of the cavity, $q_a$ (i.e. $q_a = q_c$) (Fig 21).

![Convective heat loss balance](image1)

Figure 21. Convective heat loss balance

The cavity is divided into two zones: a convective zone and a stagnation zone (Fig. 22). The horizontal plane cutting through the upper lip of the cavity aperture divides the convective zone from the stagnation zone. The convective current in the cavity flow over the heated surfaces, the refractory surfaces, and the area of the horizontal plane dividing the stagnation zone from the convective zone. The heated and refractory walls in the stagnation zone do not participate in any convective heat transfer.

![Receiver internal cavity zones](image2)

Figure 22. Receiver internal cavity zones
1. Convective Energy Loss Through the Aperture

According to Clausing, the convective energy loss through the aperture is given by:

\[ q_a = c_p (\rho_\infty V_a A_a) (T_c - T_\infty) \]  \hspace{1cm} (26)

where:

- \( \rho_\infty \) = ambient air density [kg/m\(^3\)]
- \( V_a \) = average air flow velocity into the aperture [m/s]
- \( A_a \) = area of the aperture through which air flows into the aperture [m\(^2\)]
- \( c_p \) = specific heat of ambient air [J/kg-K]
- \( T_c \) = temperature of the exiting air [K]
- \( T_\infty \) = ambient air temperature [K]

The average exiting velocity is given by (11):

\[ V_a = \frac{1}{2} \sqrt{[(C_3 V_b)^2 + (C_4 V)^2]} \]  \hspace{1cm} (27)

where:

- \( C_3 = 1 \)
- \( C_4 = 1/2 \)
- \( V \) = wind speed [m/s]
- \( V_b \) = buoyancy induced velocity [m/s]

For the no-wind condition, the interest of study, the equation reduces to:

\[ V_a = \frac{1}{2} V_b \]  \hspace{1cm} (28)
The buoyancy induced velocity is given by:

\[ V_b = \sqrt{g\beta(T_c + T_\infty)L_a} \]  \hspace{1cm} (29)

where:

- \( g \) = local gravitational acceleration [m/s\(^2\)]
- \( \beta \) = coefficient of volume expansion [k\(^{-1}\)]
- \( L_a \) = projected vertical height of the aperture [m]

For air the temperature coefficient of volume expansion is given by:

\[ \beta = \frac{1}{T_b} \]  \hspace{1cm} (30)

where:

- \( T_b \) = bulk air temperature in the convective zone of the cavity [K]

The bulk air temperature is given by:

\[ T_b = \frac{T_c + T_\infty}{2} \]  \hspace{1cm} (31)

The projected vertical height of the cavity is given by:

\[ L_a = D_a\cos \theta \]  \hspace{1cm} (32)

where:

- \( D_a \) = the cavity aperture diameter [m]
- \( \theta \) = receiver angle [degrees]
2. Convective Energy Loss Within the Receiver

The Clausing model for convective heat loss within the cavity is given by:

\[ q_c = hA_t(T_t - T_b) + hA_w(T_w - T_b) + hA_s(T_s - T_b) \]  \hspace{1cm} (33)

where:

- \( h \) = average heat transfer coefficient [W/m\(^2\)-K]
- \( A_t \) = tube surface area in convective zone [m\(^2\)]
- \( T_t \) = average tube surface temperature [K]
- \( A_w \) = refractory surface area of cavity in convective zone [m\(^2\)]
- \( T_w \) = average refractory surface temperature [K]
- \( A_s \) = area of interface plane between convective zone and stagnation zone [m\(^2\)]
- \( T_s \) = average temperature of interface plane [K]

The average heat transfer coefficient is determined from the Nusselt number and is given by:

\[ h = \frac{\text{Nu}}{L_a} k \]  \hspace{1cm} (34)

where:

- \( \text{Nu} \) = Nusselt number
- \( k \) = kinematic viscosity of air at the bulk fluid temperature [W/m-K]

For small receivers with a Grashof number of around \( 2.6 \times 10^9 \) the Nusselt number is given by:

\[ \text{Nu} = 0.10 (\text{Gr Pr})^{1/3} \]  \hspace{1cm} (35)
where:

\[ Pr = \text{Prandtl number} = 0.7 \text{ (air)} \]

\[ Gr = \text{Grashof number} \]

The Grashof number is given by:

\[ Gr = \frac{g\beta}{v^2} (T_w - T_\infty)L_a^3 \]  \hspace{1cm} (36)

where:

\[ v = \text{kinematic viscosity of air at the film temperature \,[m^2/s]} \]

The film temperature is given by:

\[ T_f = \frac{T_w + T_b}{2} \]  \hspace{1cm} (37)

For the Grashof number expression the coefficient of expansion is given by:

\[ \beta = \frac{1}{T_f} \]  \hspace{1cm} (38)

Clausing assumes the temperature of the shear plane to be equal to the tube surface temperature in the convective zone (i.e. \( T_s = T_f \)). The cavity convective heat loss is then given by:

\[ q_c = hA_t(T_t - T_b) + hA_w(T_w - T_b) + hA_s(T_t - T_b) \]  \hspace{1cm} (39)

In this work, the refractory surface temperature is assumed to be 100°F cooler than the tube surface temperature. This temperature difference is typical for measured values at the end
plate refractory surface and a heated tube surface near the end plate.

The convective heat transfer areas within the convective zone will vary in size with changes in receiver angle. The expressions for the convective heat transfer areas as a function of receiver angle area are developed in the Zone Area Formulas section.

3. Radiative Energy Loss Through the Aperture

An approximation for the radiative energy loss from the cavity through the aperture, as presented by Clausing, is given by:

\[ q_r = A_a \varepsilon \sigma \left( \frac{A_c}{A_c + A_h} (T_w^4 - T_a^4) + \frac{A_h}{A_c + A_h} (T_m^4 - T_a^4) \right) \]  \hspace{1cm} (40)

where:

\[ \varepsilon = \text{emittance of the cavity} \]

\[ \sigma = \text{Stefan-Boltzmann constant [W/m}^2\cdot\text{K}^4] \]

If the aperture is assumed to radiate as a black body then the emissivity of the cavity is equal to one; especially when the ratio of aperture size to cavity volume is small.

4. Conductive Energy Loss From the Receiver

The Clausing model for the conductive heat loss through the cavity walls is given by:

\[ q_k = \frac{k}{t} \left[ A_h (T_m - T_a) + A_w (T_w - T_a) \right] \]  \hspace{1cm} (41)
where:

\[ k = \text{thermal conductivity of the cavity walls [W/mK]} \]
\[ t = \text{thickness of the cavity walls [m]} \]

For this study:

\[ k = 0.04756 \text{ [W/mK]} \]
\[ t = 0.0889 \text{ [m]} = (3.5 \text{ [in.]}) \]

5. Zone Area Formulas

The receiver cavity is divided into two zones (Fig. 23). The boundary between the zones is formed by a horizontal plane cutting through the cavity at the upper lip of the aperture. The upper zone is assumed stagnant while the lower zone has active convective currents. The area in zone 1 is represented by the internal surface area of the receiver above the horizontal plane. The area in zone 2 is represented by the internal surface area of the receiver below the horizontal plane. The areas of zone 1 and zone 2 vary with receiver angle for a given receiver geometry.

Figure 23. Cavity zones areas
The receiver internal geometry is divided into five sections representing the hot and cold surfaces in the receiver (Fig. 24). The hot surfaces are actively heated. The cold surfaces represent the refractory surfaces. Section 1 is the circular plate at the end of the frustum. Section 2 is the frustum portion of the tube bundle. Section 3 is the cylindrical portion of the tube bundle. Section 4 is the short refractory portion of the cylindrical section. Section 5 is the refractory ring that forms the aperture.

As the receiver is rotated through various angles, each section of the internal receiver geometry may be divided by the horizontal plane that cuts through the upper inside edge of section 5. The formulas defining the portion of the area of each section that is in zone 1 for a given receiver angle range are derived in Appendix 7.

6. Shear Plane Area

The shear plane area is the area of the horizontal plane within the cavity (Fig. 25). The shear plane area is divided into two sections. The first section is formed by the horizontal plane cutting through the cylindrical portion of the receiver cavity. Not all of the horizontal plane in the cylindrical portion participates in the convective heat loss. The sides of the aperture reduce the effective shear plane area by restricting flow along the horizontal plane at the sides of the cavity near the aperture. The shear plane expands from the upper lip of
the aperture in the horizontal plane. The second section is formed where the horizontal plane cuts the frustum portion of the receiver cavity. The formulas that describe the shear plane area in the specified portion of the cavity for a given receiver angle are derived in Appendix 7.

![Diagram of shear plane](image)

Figure 25. View looking down showing the effective shear plane area

**7. Clausing Model Analysis**

A BASIC computer language program was written to solve the convective heat transfer equations (Appendix 8). The mean operating temperature in the program is taken as the tube surface temperature in the convective zone of the receiver. The program calculates the convective energy loss from the receiver cavity for receivers at operating temperatures of 300°F, 400°F, 500°F, and 600°F, receiver angles from 0 to 90° at 15° increments, and aperture diameters of 6 inches, 12 inches, 18 inches, and 26 inches. The results of the Clausing heat loss analysis are presented in Appendix 9. The results of the Clausing model analysis are shown in Figure 26.
D. Siebers and Kraabel Model

Siebers and Kraabel present a simple model for the convective heat transfer from a solar cavity receiver. They emphasize that the model has a large degree of uncertainty due to the lack of sufficient data on cavity receivers. The model was developed for a large central receiver cavity operations on-flux. This model is based primarily on the results of experimental studies from cubical cavities.

The following are the equations used to determine the convective loss from a solar cavity receiver. For natural convection the Nusselt number is given by:

$$Nu_L = 0.088Gr_L^{1/3} \left(\frac{T_w}{T_{sw}}\right)^{0.18}$$ (42)
for $10^5 \leq \text{Gr}_L \leq 10^2$

where:

- $\text{Gr}_L =$ Grashoff number
- $T_w =$ average interior cavity wall temperature [K]
- $T_\infty =$ ambient temperature [K]
- $[\_L =$ the projected vertical height of the receiver aperture [m]

An approximation for the average interior surface area of the cavity is given by:

$$\overline{T_w} = \frac{T_h A_h + T_c A_c}{A_{\text{total}}}$$  \hspace{1cm} (43)

where:

- $T_h =$ average operating temperature of the system [°C]
- $T_c =$ average refractory surface temperature in the cavity [°C]
- $T_c = T_h - 56$°C
- $A_h =$ heated surface area in the cavity [m$^2$]
- $A_c =$ refractory surface area in the cavity [m$^2$]

The $1/3$ exponent on the Grashof number results in a heat transfer coefficient that is independent of cavity dimensions. All fluid properties are evaluated at $T_\infty$. The natural convective heat transfer coefficient is given by:

$$h_{nc,0} = 0.81 \left( \overline{T_w} - T_\infty \right)^{0.426}$$  \hspace{1cm} (44)

where:

- $[\_nc =$ natural convection
\[ h = \text{no lip heat transfer coefficient} \]

The convective heat loss energy is given by:

\[ Q_{\text{conv}} = \bar{h}_{nc,0} A (T_w - T_\infty) \quad (45) \]

where:

- \( A \) = the total interior surface area of the cavity receiver [m\(^2\)]
- \( \bar{T}_w \) = the average receiver heated surface temperature [°C]

Siebers and Kraabel account for aperture effects by multiplying the natural convective heat transfer coefficient by an area ratio factor. The natural convective heat transfer coefficient including the effects of the aperture lip is given by:

\[ \bar{h}_{nc} = h_{nc,0} \left( \frac{A_1}{A_2} \right) \left( \frac{A_3}{A_1} \right)^n \quad (46) \]

where:

- \( n = 0.63 \) for \( 0^\circ \leq \phi \leq 30^\circ \)
- \( n = 0.8 \) for \( 30^\circ \leq \phi \leq 90^\circ \)
- \( A_1 \) = total interior cavity surface area [m\(^2\)] (Fig. 27)
- \( A_3 \) = interior cavity surface area below the horizontal plane [m\(^2\)] (Fig. 27)
- \( A_2 = A_1 \) minus the area of the lower lip [m\(^2\)] (Fig. 27)
Figure 27. Siebers and Kraabel cavity areas

The refractory surfaces are assumed to be 56°C cooler than the mean operating temperature of the receiver. The formulas for areas $A_1$ and $A_2$ are given as follows:

$$A_1 = \pi (R_e + R_o) \sqrt{L_1^2 + (R_e - R_o)^2} + 2\pi R_o L_{th} + \pi R_o^2 + \pi (R_e^2 - R_o^2) + 2\pi R_e L_{in} \quad (47)$$

$$A_2 = A_1 - R_e \cos^{-1} \left( \frac{R_e}{R_o} \right) + R_o \sqrt{(R_e^2 - R_o^2)} \quad (48)$$

The formula for $A_3$ depends on the particular receiver angle. The expressions developed for the variation of the cavity internal zone areas as a function of receiver angle can be found in Appendix 7.

The Grashof number is given by:

$$Gr_L = g \beta (T_w - T_{in}) \frac{L_2^3}{v^2} \quad (49)$$
where:

\[
g = \text{gravitational constant, } 9.81 \text{ m/s}^2
\]

\[
L = \text{cavity diameter [m]}
\]

\[
u = \text{kinematic viscosity [m}^2\text{/s]}
\]

\[
\beta = \text{coefficient of volumetric expansion [K}^{-1}\text{]}
\]

The uncertainty analysis provided by Siebers and Kraabel is presented in Table 2.

### Table 2
Siebers and Kraabel Uncertainty Analysis

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>(A_1)</td>
<td>(\pm 10%)</td>
</tr>
<tr>
<td>(A_2)</td>
<td>(\pm 10%)</td>
</tr>
<tr>
<td>(A_3)</td>
<td>(\pm 10%)</td>
</tr>
<tr>
<td>(A_{an})</td>
<td>(\pm 5%)</td>
</tr>
<tr>
<td>(T_w)</td>
<td>(\pm 10%)</td>
</tr>
<tr>
<td>(T_{\infty})</td>
<td>(\pm 2%)</td>
</tr>
<tr>
<td>natural convection correlation</td>
<td>(\pm 20%)</td>
</tr>
</tbody>
</table>

A computer program was used to solve for the Siebers and Kraabel convective heat loss for various aperture diameters, receiver operating temperatures, and receiver angle. A listing of the computer program is in Appendix 10. The results of the convective heat loss analysis using the Siebers and Kraabel model are presented in Appendix 11. The Siebers and Kraabel predictions for the receiver convective heat loss variation with receiver angle are presented in Figure 28.
With increasing aperture diameter there is a decrease in convective heat loss (Fig. 13 & 14).

The effect of receiver angle on the convective heat loss is more pronounced on larger aperture diameters.

The conduction heat loss forms approximately 65% of the total heat loss for all operating temperatures when the aperture diameter is small and the receiver is placed at a typical operating angle of 45° (Fig. 21).

The percent conduction is reduced and the percent radiative increased with increases in the aperture. With an aperture greater than 12 inches, the percent conduction of the total heat
loss is constant at about 38%. With the 6 inch diameter the percent conduction of the total heat loss is reduced to 25%.

Of primary interest is how well the various models compare with the experimental results. Of the six models examined, only the LeQuere, Penot and Mirenayat model provides for negative receiver angles.

A. Comparison of Previous Models with Experimental Data

The convective heat loss values predicted by the LeQuere, Penot and Mirenayat model are compared with the experimental results of Phase One and Phase Two (Fig. 29). For an ideal correlation, all the data points would fall on the equal value line. An ideal correlation occurs when the predicted results equal the experimental results. All the convective heat loss values predicted by the LeQuere, Penot and Mirenayat model are lower than the experimental results. The large degree of data scattering for higher heat loss values makes it difficult to apply a simple correction factor to the model.

The Koenig and Marvin model for convective heat loss demonstrates more agreement with experimental result than the LeQuere, Penot and Mirenayat model (Fig. 30). The Koenig and Marvin model yields higher convective heat loss values, as compared to the experimental results. The data shows increasing scatter for higher heat loss values. The model only works for positive receiver angles.
Figure 29. LeQuere, Penot, Mirenayat Convective Heat Loss Model Correlation
The Clausing model provides the best fit of all the previous models examined (Fig. 31).

This model predicts only heat loss values for positive receiver angles. The Clausing model is considerably more complicated than any of the other models. The Clausing model overestimates the convective heat loss for lower heat loss conditions and underestimates the convective heat loss for higher heat loss conditions.
Figure 31. Clausing Convective Heat Loss Model Correlation

Figure 32. Siebers and Kraabel Convective Heat Loss Model Correlation
The Siebers and Kraabel model over estimates the convective heat loss as compared with the experimental results (Fig. 32). This model also shows considerable scatter for all heat loss conditions.

B. Stine and McDonald Correlation

The Stine and McDonald model is an extension of the Siebers and Kraabel model to include the effects of varying receiver aperture size and receiver angle (8). The complex set of area determinations are not used in the Stine and McDonald model. The Stine and McDonald correlation for the Nusselt number is given as follows:

\[
\text{Nu}_L = 0.088 \frac{\text{Gr}_L^{1/4} \left( \frac{T_w}{T_\infty} \right)^{0.18} (\cos \phi)^{2.47}}{\left( \frac{d}{L} \right)^s}
\]  

(50)

and

\[
s = 1.12 - 0.98 \left( \frac{d}{L} \right)
\]  

(51)

where:

- \( d \) = aperture diameter [m]
- \( \text{Gr}_L \) = Grashof number based on length \( L \)
- \( L \) = average internal dimension of cavity [m]
- \( \text{Nu}_L \) = Nusselt number based on length \( L \)
- \( T_\infty \) = ambient temperature [K]
- \( T_w \) = average internal wall temperature [K]
- \( f \) = tilt angle of cavity
  
  (\( f = 90^\circ \) is aperture-down, \( f = 0^\circ \) is aperture-sideways)

The aspect ratio term, \( d/L \), accounts for the combined effects of internal surface area and aperture flow area. The effect of the receiver aspect diminishes with increase in aperture size, with the exponent 's'. The Stine and McDonald model predictions compare well with experimental results (Fig. 33). This model can only be applied for positive receiver angles.
The computer program used to generate the Stine and McDonald convective heat loss values is provided in Appendix 12. The data output from the program is presented in Appendix 13.

VIII. ANALYTICAL RADIATIVE HEAT LOSS

In this section the equations used to predict the thermal radiative heat loss through the aperture of the cavity solar receiver are developed. The receiver cavity surfaces are assumed to radiate as gray bodies. The internal geometry is simplified to aid in the formulation of the shape factor expressions.

A. Internal Geometry
The internal receiver surfaces were divided into five main sections (Fig. 34). The sections are defined as either hot or cold. The hot sections were those whose walls were formed by the heat transfer tubing. The hot sections were divided into an integer number of flat, concentric, isothermal bands. The number of bands in each section was determined by the number of turns the heat transfer tubing made in that section. The frustum section has 23 bands and the hot cylindrical section has 15 bands. The width of each band is equal to the surface length of each section divided by the number of bands in each section. The actual spacing between adjacent tubes was not considered significant.

![Figure 34. Receiver internal surface sections.](image)

1. Nomenclature

The following list defines the nomenclature used in the thermal radiative heat loss formulas.

- \( r_e \) = end plate radius [12.7 cm]
- \( r_c \) = cavity radius [33.0 cm]
- \( r_a \) = aperture radius [7.6, 15.2, 22.9, 33.0 cm]
- \( l_c \) = length of frustum section [29.2 cm]
- \( l_m \) = length of hot cylindrical section [25.4 cm]
- \( l_a \) = length of cold cylindrical section [14.0 cm]
- \( l_b \) = width of hot isothermal bands [1.7 cm]
N_c = number of bands in frustum section (23)

N_m = number of bands in hot cylindrical section (15)

\[ l_c = \left( l_b \cdot N_c \right)^2 - \left( r_c - r_e \right)^2 \right)^{\frac{1}{2}} \]  

(54)

B. Assumptions

A number of assumptions are necessary to simplify the thermal radiative heat loss calculations. The assumptions made for this analysis are listed as follows:

1. Each band is isothermal based on a linear interpolation between the inlet temperature at the narrow end of the frustum section to the outlet temperature at the bottom end of the hot cylindrical section.
2. Each band is considered as a flat surface.
3. Each tube band is diffuse and gray with emissivity, \( \varepsilon = 0.85 \).
4. The incident and reflected energy flux is uniform over each area.
5. Each band is adjacent to the next (i.e., no gaps between bands).
6. Each refractory surface has an emissivity of 0.70.

C. Shape Factors

All formulas are developed from the basic disc-to-disc shape factor formula.\(^{(20)}\) The N by N coefficient matrix of the surface energy balance equation (Eqn. 64) requires \( N^2 \) shape factors. The equations are solved using a digital computer. Equation 64 is conservation of energy from all the surfaces. Shape factors describe the geometric relationship between surfaces. The derivation of the shape factor formulas used in the coefficient matrix are presented in Appendix 14.
D. Thermal Radiative Heat Loss Equations

The general equation for thermal radiative heat loss from the receiver through the aperture for internal black body surfaces is given by:

\[ Q_a = -\sigma T_e^4 A_e F_{A_e} - A_a - \sigma T_s^4 A_s F_{A_s} - A_a - \sigma A_{n_m} \sum_{n_m=1}^{N_m} T_{n_m}^4 F_{n_m} - A_a - \sigma \sum_{n_c=1}^{N_c} T_{n_c}^4 A_{n_c} F_{n_c} - A_a \]

where:

\[ ]_s = \text{annuls} \]
\[ ]_e = \text{end plate} \]
\[ ]_a = \text{aperture} \]
\[ ]_{n_m} = \text{cylindrical section} \]
\[ ]_{n_c} = \text{frustum section} \]

As the aperture size to cavity volume ratio decreases, the radiative characteristics of the receiver cavity approach those of a black body emitter. To account for the various aperture sizes studied, the diffuse gray surface formulas were used. Using the net radiative method, the radiative heat loss for a cavity with diffuse gray surfaces is given by:

\[ Q_k = q_k A_k = (q_{o,k} - q_{i,k}) A_k \] (56)

\[ q_{o,k} = \varepsilon_k \sigma T_k^4 + \rho_k q_{i,k} \] (57)

\[ q_{o,k} = \varepsilon_k \sigma T_k^4 + (1 - \varepsilon_k) q_{i,k} \] (58)

where:

\[ q_o \] is the outgoing radiant energy flux (radiosity) [W/m²]
\[ q_i \] is the incoming radiant energy flux [W/m²]
For the aperture:

\[ q_{o,\text{opening}} = 0 \]

\[ T_{\text{opening}} = 0 \]

\[ \varepsilon_{\text{opening}} = 1 \]

For the cavity:

\[ \sum_{j=1}^{N} [\delta_{kj} - (1 - \varepsilon_k) F_{k \to j}] q_{o,j} = \varepsilon_k \sigma T_k^4 \]

(59)

where

\[ N = \text{total number of surfaces} \]

\[ F_{k \to j} = \text{shape factor for surface } k \text{ to surface } j \]

\[ \delta_{kj} = \text{Kronecker delta} \]

\[ \delta_{kj} = \begin{cases} 1 & \text{when } k = j \\ 0 & \text{when } k \neq j \end{cases} \]

therefore:

\[ Q_k = A_k \frac{\varepsilon_k}{(1 - \varepsilon_k)} (\sigma T_k^4 - q_{o,k}) \]

(60)

and

\[ q_{i,k} = \sum_{j=1}^{N} F_{k \to j} q_{o,j} \]

(61)

letting

\[ a_{kj} = \delta_{kj} - (1 - \varepsilon_k) F_{k \to j} \]

(62)

and

\[ C_k = \varepsilon_k \sigma T_k^4 \]

(63)
then

\[
\begin{bmatrix}
a_{11} & \ldots & a_{1j} & \ldots & a_{1N} \\
a_{k1} & \ldots & a_{kj} & \ldots & a_{kN} \\
a_{N1} & \ldots & a_{Nj} & \ldots & a_{NN}
\end{bmatrix}
\begin{bmatrix}
q_{o,1} \\
q_{o,k} \\
q_{o,N}
\end{bmatrix}
= 
\begin{bmatrix}
C_1 \\
C_k \\
C_N
\end{bmatrix}
\] (64)

The solutions for \(q_{o,k}\) are accomplished using a BASIC computer language program (Appendix 15). The heated surfaces are assumed to have an emissivity of 0.85 based on the paint coating specifications. The refractory surfaces are assumed to have an emissivity of 0.70.

E. Assumed Cavity Temperature Distribution

The axial temperature distribution along the heating surface sections is assumed to vary linearly from the top of the frustum section to the bottom of the cylindrical section. The temperature of the top band of the frustum section is equal to the receiver inlet temperature. The temperature of the bottom band of the cylindrical section is equal to the receiver outlet temperature. The inlet and outlet temperature values used in the computer program are assumed to be plus and minus 7.5°F of the operating temperature, respectively.
**Table 3**

**Inlet and Outlet temperatures**

<table>
<thead>
<tr>
<th>Test</th>
<th>Phase</th>
<th>Aperture Diameter [in]</th>
<th>Inlet [°F]</th>
<th>Outlet [°F]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>18</td>
<td>297.7</td>
<td>289.0</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>18</td>
<td>402.4</td>
<td>389.6</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>18</td>
<td>515.6</td>
<td>497.1</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>18</td>
<td>603.9</td>
<td>581.3</td>
</tr>
<tr>
<td>2</td>
<td>6</td>
<td>6</td>
<td>395.2</td>
<td>389.9</td>
</tr>
<tr>
<td>2</td>
<td>6</td>
<td>6</td>
<td>609.3</td>
<td>595.9</td>
</tr>
<tr>
<td>2</td>
<td>12</td>
<td>12</td>
<td>429.4</td>
<td>419.1</td>
</tr>
<tr>
<td>2</td>
<td>12</td>
<td>12</td>
<td>611.3</td>
<td>592.9</td>
</tr>
<tr>
<td>2</td>
<td>18</td>
<td>18</td>
<td>429.3</td>
<td>415.7</td>
</tr>
<tr>
<td>2</td>
<td>18</td>
<td>18</td>
<td>600.3</td>
<td>576.2</td>
</tr>
<tr>
<td>2</td>
<td>26*</td>
<td>26*</td>
<td>414.3</td>
<td>399.5</td>
</tr>
<tr>
<td>2</td>
<td>26*</td>
<td>26*</td>
<td>602.6</td>
<td>570.4</td>
</tr>
</tbody>
</table>

* There is no annulus therefore the aperture diameter is equal to the cavity diameter.

**F. Comparison with Measurements**

The radiometer and the analytical methods are compared with the experimental method, $Q_{\text{unplugged}} - Q_{\text{plugged}}$, for determining the radiative heat loss from the cavity through the aperture. The experimentally determined radiative heat loss was the difference between the open and plugged total receiver heat loss when the receiver aperture was down. A correction was made to account for the heat loss through the aperture plug. The heat loss through the plug must be added to the experimentally determined radiative heat loss data to get the total radiative heat loss from the receiver. The conductive heat loss through the plug is given by:

$$Q_{\text{plug}} = \frac{k_{\text{plug}} \Delta T A_{\text{plug}}}{\Delta x} \quad (65)$$

where:
\[ k_{plug} = \text{thermal conductivity of the plug}[W/m-K] \]
\[ \Delta T = \text{temperature difference between the inner and outer surface}[K] \]
\[ A_{plug} = \text{mean surface area of the plug}[m] \]
\[ \Delta x = \text{distance over which } \Delta T \text{ occurs (the thickness of the plug)}[m] \]

The plugs were fabricated from one inch thick Cera Form® boards. The boards have a mean thermal conductivity of 0.32 Btu-in/hr-sq. ft.-°F (0.0462 W/m-K).

Log-log scales are used to provide linear constant percent difference lines.

1. **Radiometer Method**

The radiometer determined heat loss values compared well with the experimentally determined radiative heat loss, \(Q_{\text{unplugged}}\) minus \(Q_{\text{plugged}}\), with differences within ±20% (Fig. 35).

![Figure 35. Experimentally determined radiometer method correlation](image-url)
2. Analytical Method

The analytical method (Fig. 36) did not compare well with the experimental method for determining the radiative heat loss from the receiver. A number of possible reasons for the discrepancies have been discussed previously.

![Analytically determined radiative correlation](image)

Figure 36. Analytically determined radiative correlation

IX. INSTRUMENTATION CALIBRATION

A. Flow Meter Calibration

The flow measurement apparatus consists of three basic parts: the turbine flow meter, the inductive pick-off, and the pulse rate counter (Fig. 37). The turbine flow meter rotates at a specific rate for a given fluid type and volumetric fluid flow rate. The rate of rotation is linearly proportional to the fluid flow rate within the specified range. The flow meter must
be calibrated for a specific fluid viscosity, temperature, and flow rate range for accurate measurement.

The inductive pick-off is positioned above the turbine flow meter. When a turbine blade passes the inductive pick-off, an electromagnetically induced pulse signal is sent to the pulse rate converter (PRC). The PRC changes the pulse rate signal to voltage or current outputs. The current or voltage signal is then read by the data acquisition computer. The voltage signal should be used when the distance from the PRC to the computer input terminal is less than ten feet. For distances greater than ten feet the current signal should be used, as long leads result in substantial voltage drops. Voltage drops may significantly skew the voltage signal. The current signal is not affected by the voltage drop. The current signal requires a precision resistor across the computer input terminals. The resistor effectively converts the current signal to a voltage input at the computer terminals. Since precision resistors are expensive, economy requires the voltage signal should be used whenever possible.

Voltage signal output was used during testing for the reasons stated above. The computer displays the equivalent fluid volumetric flow rate based on the voltage input. The slope and offset values of the flow rate versus voltage linear function were inputted into the computer.
The heat transfer fluid flow rate is measured with three turbine type flow meters in series. Three flow meters are used for measurement redundancy. One of these flow meters was factory calibrated. The factory calibration specifications sheet is Appendix 16. The calibration curve for the factory calibrated flow meter is presented in Figure 38. The equation of the volumetric flow rate as a function of flow meter output frequency for the factory calibrated flow meter is:

\[
\dot{\nu} = 0.0039578 + 0.001489f 
\]

(65)

where:

\( f \) = the flow meter output frequency [Hz]

\( \dot{\nu} \) = volumetric flow rate [gpm]
The pulse rate converters (PRCs) were calibrated in-house according to the manufacturer's procedure \(^{(13)}\). The calibration points for the PRC used with the factory calibrated flow meter are presented in Table 4.

### Table 4
Pulse Rate Converter Calibration Points

<table>
<thead>
<tr>
<th>frequency [Hz]</th>
<th>output voltage [volts]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1600</td>
<td>8.0</td>
</tr>
<tr>
<td>2000</td>
<td>10.0</td>
</tr>
</tbody>
</table>
The linear relationship for the frequency as a function of output voltage, as determined by the calibration points, is given as:

\[ f = 200 \, E \]  \hspace{1cm} (66)

where:

\( f \) = is the frequency input to the PRC [Hz]
\( E \) = is the output signal from the PRC [volts]

Substituting the frequency equation of the PRC calibration into the flow rate equation of the factory calibrated flow meter yields an expression for the volumetric flow rate as a function of voltage as follows:

\[ \dot{v} = 0.0039578 + 0.001489(200 \, E) \]  \hspace{1cm} (67)

which reduces to:

\[ \dot{v} = 0.0039578 + 0.2978E \]  \hspace{1cm} (68)

which can be approximated as:

\[ \dot{v} = 0.2978E \]  \hspace{1cm} (69)

Because \( E > 3.36 \) volts at typical flow rates the error due to this approximation is less than:

\[
\frac{0.0039578}{(0.2978)(3.36)}
\]

The remaining two flow meters were calibrated against the factory calibrated flow meter.

Three meters were used for redundancy in the event one meter fails during testing.

1. Flow Meters Calibration

The outputs from the three flow meters were compared for various flow rates. The voltage outputs of the three flow meters were recorded (Appendix 17) (Fig. 39). The heat transfer fluid temperature was maintained at 300°F. Corrective slope and offset values were determined for each of the two uncalibrated flow meters. Applying the linear corrections
will bring the two uncalibrated flow meter voltage outputs into agreement with the factory calibrated flow meter output.

\[
\text{Cal Flow} = -2.9070 + 2.9616 \text{Flow}_1 \quad R^2 = 0.999
\]

\[
\text{Cal Flow} = -3.2878 + 3.1808 \text{Flow}_2 \quad R^2 = 1.000
\]

![Graph showing correlation curves for in-house calibrated flow meters](image)

Figure 39. In-house calibrated flow meters correlation curves

The difference between the corrected readings and the factory calibrated readings are compared (Fig. 40). The minimum flow rate for all testing never went below one gallon per minute or 3.36 volts. This flow rate is well within the manufacture specified flow range. The maximum difference between the backup flow meters and the factory calibrated flow meter is ±0.15 volts or ±0.045 gallons per minute. Temperature effects are negligible \(^{(14)}\).
Substituting the linear correction expression for the in-house calibrated flow meters into the volumetric flow rate expression of the factory calibrated flow meter yields expressions for the volumetric flow rates as a function of voltage inputs to the computer. The resulting expressions are given as follows:

\[ \dot{v}_{\text{flow 1}} = 0.2978(-2.9070 + 2.9616E_{\text{flow 1}}) \]  
\[ \dot{v}_{\text{flow 2}} = 0.2978(-3.2878 + 3.1808E_{\text{flow 2}}) \]

which reduce to:

\[ \dot{v}_{\text{flow 1}} = -0.8657 + 0.88196E_{\text{flow 1}} \]  
\[ \dot{v}_{\text{flow 2}} = -0.9791 + 0.94724E_{\text{flow 2}} \]

The scale and offset values from these expressions were inputted into the computer. The computer then displays the flow rate for each flow meter in gallons per minute (gpm).
B. Thermocouple Calibration

The total receiver heat loss, $Q_T$, was given by the product of the mass flow rate, the specific heat of the heat transfer fluid and the temperature difference between the fluid inlet and outlet temperatures ($\Delta T$). Of the three variables, the largest error in the heat loss was due to the measurement of $\Delta T$. Two redundant methods were used to measure $\Delta T$ for these experiments. One method of determining $\Delta T$ was by subtracting the fluid outlet temperature from the fluid inlet temperature. The other method utilized a direct temperature difference measurement from two thermocouples, one in each of the fluid inlet and outlet lines. This differential thermocouple connection avoids inaccuracies due to reference junction compensation but still requires knowledge of the absolute temperature values. To reduce the error introduced to the total heat loss calculation by the thermocouple readings it was necessary to calibrate the inlet, outlet, and delta temperature thermocouples used in the receiver. Standard thermocouple probes have a maximum error of $\pm2.2^\circ C$ or $0.75\%$, whichever is greater (15). Using two absolute temperature measurements from standard thermocouple probes would result in a $\Delta T$ error of $\pm3.11^\circ C$ ($\pm5.60^\circ F$) or $1.06\%$ whichever is greater. With temperature differences as low as $2.78^\circ C$ ($5^\circ F$) recorded, the standard thermocouples did not yield values within an acceptable error.

A single factory calibrated K-type thermocouple probe was purchased and all other probes were calibrated against it. Only one calibrated probe could be purchased due to budget constraints. The calibrated probe has an accuracy of $\pm0.2^\circ F$ after linear correction is applied (Fig. 41). The three point calibration data for the factory calibrated thermocouple is provided in Table 7. The calibration is certified traceable to the U.S. National Bureau of Standards (Appendix 18).
A linear variation of the actual temperature as a function of the indicated temperature is given by:

$$T_{\text{actual}} = -2.4256 + 1.006T_{\text{indicated}}$$  \hspace{1cm} (74)

with a correlation coefficient, $R=1.00$

Table 5 shows the error in the calibrated probe after linear correction is applied.
Table 5
Factory Calibrated thermocouple probe error

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>300.56</td>
<td>300.06</td>
<td>299.94</td>
<td>0.12</td>
</tr>
<tr>
<td>499.39</td>
<td>499.77</td>
<td>499.96</td>
<td>0.19</td>
</tr>
<tr>
<td>699.09</td>
<td>701.00</td>
<td>700.86</td>
<td>0.14</td>
</tr>
</tbody>
</table>

1. Thermocouple Calibration Apparatus

A thermocouple calibrating device was fabricated (Fig. 42). The calibrator consisted of a heat source, a heat sink, and an insulated cover. The heat sink was formed from a solid brass cylinder with three holes drilled in one end to accommodate thermocouple probes and a single hole in the other end to accommodate the heat source (Fig. 43). The soldering iron used was an Ungar CI-45, 0.38 A 120 V AC/DC. The soldering iron tip was modified for a snug fit in the heat sink (Fig. 44). The soldering iron was connected to a variable AC power supply. The brass block was insulated to control the rate of heat loss. The thermocouple leads were connected to a data acquisition system.
Figure 42. Thermocouple calibrator

Figure 43. Thermocouple calibrator heat sink
2. Thermocouple Calibration Procedure

Three separate tests were required to accomplish calibration of the two absolute and the two differential thermocouples. In the first test, the absolute inlet and outlet TC probes were inserted in the heat sink along with the calibrated TC probe. The heat source was plugged into a variable AC power supply. The variable AC supply was adjusted until a maximum temperature of approximately 700°F was indicated for the calibrated TC. The power supply was then removed and the system allowed to cool slowly. The temperature time histories for the absolute temperature inlet and outlet thermocouples were recorded. The heat sink was assumed to be isothermal at all TC junctions for any time. Correction factors for the TC’s were obtained by comparing the TC outputs to that of the calibrated TC output.

3. Thermocouple Calibration Results

The temperature time history curves for the absolute temperature inlet and outlet thermocouples are presented in Figure 45. The relationship for the inlet thermocouple is quite linear as indicated in Fig. 46. The linear relationship between the inlet TC reading and the calibrated TC reading is given by;

\[ T_{\text{cal}} = -.2217 + 1.0027 \text{ Tin} \]  

(75)
The relationship for the outlet thermocouple is quite linear as indicated in Fig. 47. The linear relationship between the outlet TC reading and the calibrated TC reading is given by:

\[ T_{\text{cal}} = -0.1850 + 1.0047 \times \text{Tout} \]  

(76)

Figure 45. Inlet and Outlet thermocouples calibration histories
Figure 46. Factory Calibrated versus inlet thermocouple readings

Figure 47. Factory Calibrated versus outlet thermocouple readings
These equations are used to correct the outlet TC readings to the calibrated TC readings. The difference between the calibrated and inlet TC readings are compared before and after the linearized correction factors were applied (Fig. 48). The maximum difference between the calibrated TC reading and the inlet TC reading before the correction is applied is 2.6°F. After application of the correction factors the maximum difference is 1.02°F. The maximum difference between the calibrated TC readings and the outlet TC readings before the linear correction is applied is 4.2°F. After the linear correction is applied the maximum difference is 1.44°F (Fig. 49).

Figure 48. Inlet thermocouple errors
Two thermocouples can be arranged to provide a voltage output proportional to the temperature difference of the two junctions (Fig. 50). When both thermocouple junctions are at the same temperature the voltage output is zero.

\[
V = \alpha (T_1 - T_2)
\]

Figure 49. Outlet thermocouple errors

Figure 50. Differential thermocouple connection
The two thermocouple probes used for the direct temperature difference measurement were inserted in the thermocouple calibrator along with the calibrated thermocouple. The unit was then heated and cooled as per the procedure used for calibrating the inlet and outlet absolute temperature thermocouple probes. Ideally, the net voltage output from the temperature differential connection should be zero regardless of the calibrator temperature. The temperature and micro volt output histories for the calibration procedure are presented in Figure 51. Comparing the differential thermocouple's output with the calibrated thermocouple reading indicates no linear correlation (Fig. 52). The step function of the voltage output is an indication of the minimum computer analog to digital converter (ADC) resolution.

The maximum error in the differential voltage output is 25.45 μV which corresponds to a temperature difference error of 1.147 °F.

---

Figure 51. Differential thermocouple readings history
The absolute mean temperature must also be known to convert the millivolt output of the differential thermocouples into an equivalent temperature difference. The voltage to temperature difference conversion factor is determined from a polynomial function of the mean temperature (Fig. 53). The conversion factor polynomial was derived from the voltage-temperature tables for K-type thermocouples.\(^{(14)}\)

No calibration of the differential thermocouples was performed with a difference in temperature at the junctions.
C. Radiometer Calibration

The Hy-Cal® Hy-Therm® Pyrheliometer P-8400-B was used to measure the thermal radiative heat loss from the receiver cavity (Fig. 54). The spectral range of the radiometer without its quartz window is from 0.2 to 30 microns. The radiometer consists of a thermopile on top of a heat sink. The thermopile converts the temperature gradient across the pile to a proportional current signal. The heat sink is water cooled aluminum base. The exposed end of the thermopile is coated with fused colloidal graphite providing a minimum absorptivity of 0.9. The radiometer outputs 5 mV per solar constant (0.13980 Watts/cm²). The calibration specifications are provided in Appendix 19.

The radiometer was supplied with a bezel mounted quartz window. Because the radiative being measured is in the infra-red wave length region, preliminary testing showed that the
quartz window excessively attenuated the heat flux to the radiometer resulting in an insufficient signal output. When the window was removed, the radiometer became overly sensitive to localized convective heating and cooling which resulted in a fluctuating output signal. For low heat flux level measurements, a window that is virtually transparent to the infrared radiative is desirable. Various window materials were considered and tested for replacement of the quartz window.

Figure 54. Radiometer section view

1. Radiometer Window Evaluations

A bezel was fabricated to accommodate various film windows for testing (Fig. 55). The film was pulled snugly over the bezel. The film was secured in place with a rubber band. The bezel fits over the radiometer. The film window is close enough to the thermopile wafer providing the thermopile with almost 180° of view.

Figure 55. Film window bezel
The test stand consisted of an inverted hot plate positioned concentrically above and facing the radiometer (Fig. 56). A K-type thermocouple probe is immersed in the coolant return catch basin for the radiometer. The base of the radiometer is assumed to be at the same temperature as the water in the catch basin. The hot plate is inverted and positioned above the radiometer to prevent convective heating of the radiometer. The hot plate temperature is controlled using a variable AC power supply. The radiometer can be positioned at various vertical distances from the hot plate. The surface of the hot plate has nine K-type thermocouples welded to it.

![Figure 56. Radiometer calibration test stand](image)

The hot plate surface was coated with high emissivity Pyromark® paint. The manufacturer’s specifications for the Pyromark® paint are provided in Appendix 20. The average black-body normal emittance of the painted surface is a function of the surface temperature and can be as low as 0.867 at 600K (Fig. 57). A fifth order polynomial curve fit provides an equation for paint surface emittance as a function of surface temperature.
The parameters for the window evaluation test plan are shown in Table 6. At the beginning and end of each test, background measurements were recorded by removing the radiometer from the test stand and placing it above the hot plate on an insulated pad. For each of the plate and background thermal radiative measurements the readings were recorded after a two minute stabilization period. The variable AC power supply was adjusted until the average plate temperature was close to the target temperature. The thermocouples were distributed over the plate surface (Fig. 58).
Table 6
Radiometer window test parameters

<table>
<thead>
<tr>
<th>Test Window</th>
<th>Target Temperature</th>
<th>Vertical Displacement</th>
</tr>
</thead>
<tbody>
<tr>
<td>Saran Wrap</td>
<td>300°F</td>
<td>0.32r</td>
</tr>
<tr>
<td>quartz</td>
<td>&quot;</td>
<td>&quot;</td>
</tr>
<tr>
<td>Glad Wrap</td>
<td>&quot;</td>
<td>&quot;</td>
</tr>
<tr>
<td>none</td>
<td>&quot;</td>
<td>&quot;</td>
</tr>
<tr>
<td>Saran Wrap</td>
<td>400°F</td>
<td>0.75r</td>
</tr>
<tr>
<td>quartz</td>
<td>&quot;</td>
<td>&quot;</td>
</tr>
<tr>
<td>Glad Wrap</td>
<td>&quot;</td>
<td>&quot;</td>
</tr>
<tr>
<td>none</td>
<td>&quot;</td>
<td>&quot;</td>
</tr>
<tr>
<td>Saran Wrap</td>
<td>500°F</td>
<td>1.23r</td>
</tr>
<tr>
<td>quartz</td>
<td>&quot;</td>
<td>&quot;</td>
</tr>
<tr>
<td>Glad Wrap</td>
<td>&quot;</td>
<td>&quot;</td>
</tr>
<tr>
<td>none</td>
<td>&quot;</td>
<td>&quot;</td>
</tr>
<tr>
<td>Saran Wrap</td>
<td>600°F</td>
<td>1.80r</td>
</tr>
<tr>
<td>quartz</td>
<td>&quot;</td>
<td>&quot;</td>
</tr>
<tr>
<td>Glad Wrap</td>
<td>&quot;</td>
<td>&quot;</td>
</tr>
<tr>
<td>none</td>
<td>&quot;</td>
<td>&quot;</td>
</tr>
<tr>
<td>Saran Wrap</td>
<td>700°F</td>
<td>2.5r</td>
</tr>
<tr>
<td>quartz</td>
<td>&quot;</td>
<td>&quot;</td>
</tr>
<tr>
<td>Glad Wrap</td>
<td>&quot;</td>
<td>&quot;</td>
</tr>
<tr>
<td>none</td>
<td>&quot;</td>
<td>&quot;</td>
</tr>
</tbody>
</table>

where \( r = \) radius of the hot plate

81
For an average area temperature only thermocouples five through nine were considered.

The average area temperature is given by:

\[ T_{\text{average}} = \frac{T_5 + T_6 + T_7 + T_8 + T_9}{5} \]  

(77)

The window transmittance, \( \tau_{\text{window}} \), is given by:

\[ \tau_{\text{window}} = \frac{q_{\text{radio}}}{q_{\text{radio/plate}}} \]  

(78)

where

- \( q_{\text{radio}} \) = the thermal radiative heat flux read by the radiometer
- \( q_{\text{plate/radio}} \) = the thermal radiative heat flux incident on the radiometer from the hot plate

The radiative leaving the hot plate and incident on the radiometer is given by:

\[ q_{\text{radio/plate}} = q_{\text{plate}} F_{\text{plate-radio}} \]  

(79)

where

- \( q_{\text{plate}} \) = the thermal radiative heat flux leaving the hot plate
- \( F_{\text{plate-radio}} \) = the shape factor from the hot plate to the radiometer

The radiative heat flux leaving the hot plate is given by:
\[ q_{\text{plate}} = \sigma \varepsilon (T_{\text{plate}}^4 - T_{\text{ambient}}^4) \pi r_{\text{plate}}^2 \]  

(80)

where

\[ \sigma = 5.729 \times 10^{-8} \text{ [W/(m}^2\text{K}^4)] \] Stefan-Boltzmann constant

\[ r_{\text{plate}} = 8.9 \text{ [cm]} \] the radius of the hot plate

\[ \varepsilon = \text{emissivity of the plate surface.} \]

The shape factor from the hot plate to the radiometer is given by:

\[ F_{\text{plate-radio}} = \frac{1}{2} \left( \frac{1 + h^2 + r_{\text{radio}}^2}{r_{\text{plate}}^2} - \sqrt{\left[ \frac{1 + h^2 + r_{\text{radio}}^2}{r_{\text{plate}}^2} \right]^2 - 4 \left( \frac{r_{\text{radio}}^2}{r_{\text{plate}}^2} \right)^2} \right) \]  

(81)

where

\[ h = \text{vertical displacement of the hot plate and radiometer.} \]

\[ r_{\text{radio}} = \text{radius of the radiometer} = 0.5625 \text{ [in.]} \]

When the diameter of the radiometer approaches one inch, as is the case here, a simplified formula may be used. The simplified shape factor formula is given as follows:

\[ F_{\text{plate-radio}} = \frac{r_{\text{radio}}^2}{h^2 + r_{\text{plate}}^2} \]  

(82)

All real gas effects have been ignored (absorption, scattering etc.).

The results of the test are in Appendix 21. The quartz window transmittance varies linearly with source temperature (Fig. 59). The transmittance for the Grad Wrap and Saran Wrap windows are nearly constant over the testing temperature range. The average transmittance for the no window condition is 0.99. Ideally, the transmittance for the no window
condition should be unity. The error in the no window transmittance is well within the instrument's error. Real gas effects may also account for a small loss in the incident flux on the radiometer from the hot plate.

![Figure 59. Radiometer windows transmittance](image)

The average transmittance for the Saran Wrap window was determined to be 0.87. The manufacturer specifies a transmittance of 0.88 for the infrared portion of the spectrum. Technical information for Saran Wrap® films is provided in Appendix 22. Although the Glad Wrap had similar transmission characteristics, the Saran Wrap was selected as the radiometer window, since no manufacturer specifications were available for the Glad Wrap. In addition to selecting a radiometer window, it was also necessary to determine effects of radiometer positioning relative to the heat source.
2. Radiometer Positioning

The radiometer positioning sensitivity must be considered for accurate heat flux measurements. The parameters for the radiometer positioning sensitivity test plan are presented in Table 7. At the end of each test the background measurements were taken by removing the radiometer from the test stand and placing it above the hot plate on an insulated pad. For each of the plate and background thermal radiative measurements the radiometer readings were recorded after a two minute stabilization period.

<table>
<thead>
<tr>
<th>Hot Plate Temp</th>
<th>Offset Distance</th>
<th>Vertical Displacement</th>
</tr>
</thead>
<tbody>
<tr>
<td>400°F</td>
<td>0 to 2r step 0.5r</td>
<td>0.75r</td>
</tr>
<tr>
<td>700°F</td>
<td>0 to 6r step 1r</td>
<td>2.5r</td>
</tr>
<tr>
<td>400°F</td>
<td>0</td>
<td>1 to 6r step 1r</td>
</tr>
<tr>
<td>700°F</td>
<td>0</td>
<td>2 to 12r step 2r</td>
</tr>
</tbody>
</table>

where \( r = \) radius of the hot plate

The shape factor from the hot plate to the radiometer is given by:

![Diagram of radiometer setup](image)
\[ F_{\text{plate-radio}} = \frac{1}{2} \left( 1 - \frac{1 + \left( \frac{h}{a} \right)^2 - \left( \frac{r_{\text{plate}}}{a} \right)^2}{\sqrt{\left( 1 + \left( \frac{h}{a} \right)^2 - \left( \frac{r_{\text{plate}}}{a} \right)^2 \right)^2 - 4 \left( \frac{r_{\text{radio}}}{r_{\text{plate}}} \right)^2}} \right) \left( \frac{r_{\text{radio}}}{r_{\text{plate}}} \right)^2 \] (83)

where

- \( h \) = vertical displacement of the hot plate and radiometer.
- \( r_{\text{radio}} \) = radius of the radiometer = 1.43 cm
- \( r_{\text{plate}} \) = radius of the hot plate = 8.9 cm
- \( h \) = vertical displacement [in.]
- \( a \) = horizontal displacement [in.]

The results of the displacement sensitivity test are tabled in Appendix 23. The effects of vertical and horizontal displacement of the radiometer from the heat source are presented in Figure 60. Ideally, all values should be equal to one. The radiometer is especially sensitive to horizontal displacement (i.e. off axis readings).

![Figure 60. Radiometer displacement effects](image)

The results indicate that the radiometer should be placed as close as possible and in line
with the heat source. Extreme care was taken to assure the radiometer was centered on the receiver axis during the test phase. An elaborate string and plumb bob arrangement was used to center the radiometer.

X. ERROR ANALYSIS

This error analysis determines error in the results based on test method and instruments used. The error analysis also serves as one form of evaluating the test method. The error analysis does not account for human errors or systematic errors. The general form for the error analysis of a given function, \( z \), where \( z = z(x_i) \) is given as follows:\(^{(16)}\)

\[
\sigma_z = \sqrt{\sum_i \left( \left( \frac{\partial z}{\partial x_i} \right)^2 \sigma_{x_i}^2 \right)} 
\]

(84)

The general formula for the total heat lost from the cavity is given by:

\[
Q_{\text{total}} = \rho \dot{v} c_p \Delta T. 
\]

(85)

The percent error in the total heat loss is given by:

\[
\frac{\sigma_{Q_{\text{total}}}}{Q_{\text{total}}} = \sqrt{\left(\frac{\sigma_p}{\rho}\right)^2 + \left(\frac{\sigma_{\dot{v}}}{\dot{v}}\right)^2 + \left(\frac{\sigma_{c_p}}{c_p}\right)^2 + \left(\frac{\sigma_{\Delta T}}{\Delta T}\right)^2} 
\]

(86)

The density of the heat transfer fluid is a function of the temperature and is approximated by\(^{(17)}\):

\[
\rho = 60.6 - 0.0324 \ T_{\text{inlet}} + 9.84 \times 10^{-6} \ T_{\text{inlet}}^2 - 1.79 \times 10^{-8} \ T_{\text{inlet}}^3 \ [\text{lb/ft}^3] 
\]

(87)

The error in the heat transfer fluid density is approximated by:
\[
\sigma_p = \sqrt{(-0.0324 + 1.968e^{-5}T_{\text{inlet}} - 5.37e^{-8}T_{\text{inlet}}^2)^2\sigma_{T_{\text{inlet}}}^2}
\]

The inlet temperature is used for density calculation. The inlet thermocouple used is the one closest to the flow meters.

The heat capacity of the heat transfer fluid is a function of the temperature and is approximated by\(^{(17)}\):

\[
c_p = 0.3690 + 2.267e^{-4}T_{\text{mean}} [\text{Btu/lb} \ ^{\circ}\text{F}]
\]

The error in the heat transfer fluid heat capacity is approximated by:

\[
\sigma_{c_p} = 2.267e^{-4}\sigma_{T_{\text{mean}}}
\]

**A. Flow Measurement Error Analysis**

The HTF flow rate is given by:

\[
\dot{v} = 0.2978E
\]

where:

- \(E\) = is the output signal from the pulse rate converter (PRC) [volts]
- \(\dot{v}\) = volumetric flow rate [gpm]

The manufacturer's specified linearity of the PRC is ±0.4% of full scale. The calibrated flow meter is accurate to within ±0.5% of the reading in the flow rate range of 0.8 to 2.5 gallons per minute and with a fluid viscosity in the range of 0.4 to 2.0 centistokes.\(^{(18)}\)

The output of the PRC to the computer is from 0 to 10 volts. The accuracy of the computer for the 10 volt input range is the larger of ±1% of the reading or ±0.2% of the range (the range is 11 volts for the -1 to 10 volt input ).\(^{(19)}\) The combined error for the factory calibrated flow meter reading is given by:
\[
\sigma_{\text{flow}} = \sqrt{b^2 \sigma_E^2}
\]

(92)

\[
\sigma_E = \sqrt{(\sigma_{\text{PRC}})^2 + (\sigma_{\text{flow meter}})^2 + (\sigma_{\text{computer}})^2}
\]

(93)

where:

- \(b\) = the slope of the flow rate versus voltage output for the PRC-flow meter combination
  
  \(b = 0.2978 \, \text{[gpm/volt]}\)
  
  \(\beta_{\text{PRC}}\) = the error in the pulse rate converter

\[
\frac{\sigma_{\text{PRC}}}{E_{\text{scale PRC}}} = \pm 0.4\% \text{ of full scale}
\]

\(E_{\text{scale PRC}} = 10 \, \text{volts}\)

\(\beta_{\text{flow meter}}\) = the error in the flow meter

\[
\frac{\sigma_{\text{flow meter}}}{E_{\text{reading}}} = 0.5\% \text{ of reading}
\]

\(\beta_{\text{computer}}\) = the error in the voltage signal ADC

\[
\frac{\sigma_{\text{computer}}}{E_{\text{reading or E_scale computer}}} = \pm 1.\% \text{ of reading or } \pm 0.2\% \text{ of the range}
\]

\(E_{\text{scale computer}} = 11 \, \text{volts (from -1 to + 10 volts)}\)

The maximum error on flow rate measurement is 1.6 % or 0.0208 gpm.

B. Temperature Measurement Error Analysis

The accuracy of the computer must also be taken into consideration. The specified accuracy of the data acquisition system is given as \(\pm 1.44^\circ\text{F} \, (\pm 0.8 \, ^\circ\text{C})\) with a resolution of 0.18 \(\circ\text{F}\) (0.1\(^\circ\text{C}\)) for K-type thermocouples. The accuracy of the differential thermocouple connection is specified as \(\pm 20\mu\text{V}\) for the \(\pm 25\text{mV}\) range setting. This corresponds to a
temperature difference accuracy of ±0.90°F (±0.5°C). Calibration tests indicate an error of ±25.45 μV which corresponds to a temperature difference error of ±1.14°F.

The error of the temperature readings must include the combined effects of the error of the computer, calibrated probe, the absolute temperature probes, and the computer micro volt readings. The error of the absolute temperature thermocouples is given by:

\[ \sigma_{\text{Tin/cal}} = \sqrt{\sigma_{\text{Tin/cal}}^2 + \sigma_{\text{cal}}^2 + \sigma_{\text{computer TC}}^2} \] (94)

and

\[ \sigma_{\text{Tot/out}} = \sqrt{\sigma_{\text{Tot/out}}^2 + \sigma_{\text{cal}}^2 + \sigma_{\text{computer TC}}^2} \] (95)

where

\[ \sigma_{\text{Tin/cal}} = \text{error of the inlet probe as compared with the calibrated probe.} \]
\[ \sigma_{\text{Tin/cal}} = ±1.02 °F \]
\[ \sigma_{\text{Tot/out}} = \text{error of the outlet probe as compared with the calibrated probe.} \]
\[ \sigma_{\text{Tot/out}} = ±1.44 °F \]
\[ \sigma_{\text{computer TC}} = \text{error of the computer ADC thermocouple channels.} \]
\[ \sigma_{\text{computer TC}} = ±1.0 °F \]

The mean temperature is required in converting the micro volt signal from the differential thermocouple connection to an equivalent temperature difference. The mean temperature is calculated from the absolute temperature readings of the inlet and outlet thermocouples as follows:

\[ T_{\text{mean}} = \frac{T_{\text{in}} + T_{\text{out}}}{2} \] (96)

The error in the mean temperature is given by:

\[ \sigma_{T_{\text{mean}}} = \sqrt{\sigma_{T_{\text{in}}}^2 + \sigma_{T_{\text{out}}}^2} \] (97)

The temperature difference from the differential thermocouple reading is given by:

\[ \Delta T = \frac{\mu V}{k} \] (98)
where \( k \) is the micro volt to temperature difference conversion factor.

The conversion factor, \( k \), is a function of the mean temperature (Fig. 49). The error in the measured temperature difference is given by:

\[
\sigma_{\Delta T} = \sqrt{\left(\frac{\partial \Delta T}{\partial \mu V} \sigma_{\mu V}\right)^2 + \left(\frac{\partial \Delta T}{\partial k} \sigma_k\right)^2}
\]

(99)

which reduces to:

\[
\sigma_{\Delta T} = \sqrt{\left(\frac{\sigma_{\mu V}}{k}\right)^2 + \left(\frac{\mu V}{k^2} \sigma_k\right)^2}
\]

(100)

The error in the micro volt reading is dependent on the error in the differential thermocouple output as well as the error in the computer ADC. The error in the micro volt reading is given by:

\[
\sigma_{\mu V} = \sqrt{\sigma_{\Delta T\mu V-cal} + \sigma_{computer \mu V}}
\]

(101)

where

\( \sigma_{\Delta T\mu V-cal} = \) error in differential thermocouple as compared with the calibrated thermocouple.

\( \sigma_{\Delta T\mu V-cal} = \pm 25.45 \mu V \)

\( \sigma_{computer \mu V} = \) error in the computer microvolt reading.

\( \sigma_{computer \mu V} = \pm 20 \mu V \)

The micro volt to temperature difference conversion factor, \( k \), is determined from a ninth order polynomial of the mean temperature at the inlet and outlet junctions. The function for \( k \) is given as follows:

\[
k = (m0) + (m1)T_{mean} + (m2)T^2_{mean} + (m3)T^3_{mean} + (m4)T^4_{mean} + (m5)T^5_{mean} + (m6)T^6_{mean} + (m7)T^7_{mean} + (m8)T^8_{mean} + (m9)T^9_{mean}
\]

(102)

where

\( m0 = -5.9996574248 \)

\( m1 = 0.57623140669 \)

\( m2 = -0.0050328211032 \)

\( m3 = 2.6366840630e^{-5} \)
\[m_4 = -9.2911149803 \times 10^{-8}\]
\[m_5 = 2.2361762393 \times 10^{-10}\]
\[m_6 = -3.5624990721 \times 10^{-13}\]
\[m_7 = 3.5424764505 \times 10^{-16}\]
\[m_8 = -1.9716528398 \times 10^{-19}\]
\[m_9 = 4.6528339298 \times 10^{-23}\]

The error in \( k \) is given by:
\[
\sigma_k = \sqrt{\left(\frac{\partial k}{\partial T_{\text{mean}}} \sigma_{T_{\text{mean}}}\right)^2}
\] (103)

where:
\[
\frac{\partial k}{\partial T_{\text{mean}}} = (m_1) + 2(m_2)T_{\text{mean}} + 3(m_3)T_{\text{mean}}^2 + 4(m_4)T_{\text{mean}}^3 + 5(m_5)T_{\text{mean}}^4
\]
\[
+ 6(m_6)T_{\text{mean}}^5 + 7(m_7)T_{\text{mean}}^6 + 8(m_8)T_{\text{mean}}^7 + 9(m_9)T_{\text{mean}}^8
\] (104)

C. Normalization Error Analysis

The heat loss is normalized using the following formula:
\[
Q_{\text{normalized}} = Q_{\text{measured}} \left(\frac{T_{\text{target}} - T_{\text{ambient standard}}}{T_{\text{measured}} - T_{\text{ambient measured}}}\right)
\] (105)

The percent error in the normalized heat loss is given by:
\[
\frac{\sigma Q_{\text{normalized}}}{Q_{\text{normalized}}} = \sqrt{\left(\frac{\sigma Q_{\text{measured}}}{Q_{\text{measured}}}\right)^2 + \left(\frac{\sigma N}{N}\right)^2}
\] (106)

where
\[
N = \frac{T_{\text{target}} - T_{\text{ambient standard}}}{T_{\text{measured}} - T_{\text{ambient measured}}}
\] (107)

and
\[
\frac{\sigma N}{N} = \sqrt{\frac{\sigma_{T_{\text{measured}}}^2 + \sigma_{T_{\text{ambient measured}}}^2}{(T_{\text{measured}} - T_{\text{ambient measured}})^2}}
\] (108)
The convective heat loss is given by:

$$\frac{\sigma_N}{N} = \sqrt{\frac{\sigma_{T_{\text{meas}}}^2 + \sigma_{T_{\text{ambient measured}}}^2}{(T_{\text{meas}} - T_{\text{ambient measured}})^2}} \quad (109)$$

The error in the convective heat loss is given by:

$$Q_{\text{convective}} = Q_{\text{total}} - Q_{\text{total 90°}} \quad (110)$$

The conductive heat loss is given by:

$$Q_{\text{conductive}} = Q_{\text{total 90° plugged}} \quad (111)$$

The error in the conductive heat loss is given by:

$$\sigma_{Q_{\text{conductive}}} = \sigma_{Q_{\text{total 90° plugged}}} \quad (112)$$

The radiative heat loss is given by:

$$Q_{\text{radiative}} = Q_{\text{total 90° unplugged}} - Q_{\text{conductive}} \quad (113)$$

The error in the radiative heat loss is given by:

$$\sigma_{Q_{\text{radiative}}} = \sqrt{\sigma_{Q_{\text{total 90° unplugged}}}^2 + \sigma_{Q_{\text{conductive}}}^2} \quad (114)$$

The maximum error in the normalized total heat loss is 30.63%. This error occurs with the aperture plugged and the receiver in the +90° position (Fig. 61). The total heat loss with the aperture plugged is equivalent to the conductive heat loss.
Figure 61. Error in total heat loss for various operating temperatures

Figure 62. Error in total heat loss for various aperture diameter operating at 400°F
The errors in the convective and radiative heat losses are higher due to summation variables, each of which contains an error (Equations 110, 111, 113). The maximum error in the convective heat loss is 35.70%. The maximum convective heat loss error occurs at a receiver operating temperature of 400°F with a 6 inch aperture and the receiver in the +90° position (Fig. 64).
Figure 64. Error in convective heat loss for various operating temperatures
Figure 65. Error in convective heat loss for various aperture diameter operating at 400°F

Figure 66. Error in convective heat loss for various aperture diameter operating at 600°F
Figure 67. Error in radiative heat loss from Phase 1 test

Figure 68. Error in radiative heat loss from Phase 2 test
The error in the radiative heat loss measurement increases with decrease in receiver operating temperature and receiver aperture diameter (Fig. 67 & 68). The maximum error in the radiative heat loss is 38.5% with a six inch aperture and a 400°F operating temperature.

XI. CONCLUSIONS:

The results of experimental testing of the heat loss from a solar cavity receiver for various aperture diameters and operating temperatures have been presented. With an increase in aperture size there was an increase in the convective heat loss. The effect of aperture size on the convective heat loss decreased with increases in aperture size. Decreasing the aperture diameter from 18 inches to 6 inches, reduce the convective losses by 60%. The 18 inch aperture presently used for this type of receiver has little or no effect on the free convective heat loss from the receiver.

Conduction was the primary mode of heat loss from the receiver for very small apertures (less than 12 inch diameter).

A low cost radiometer can be used to determine the radiative heat loss from a cavity within ±20% of experimentally determined radiative heat loss.

XII. RECOMMENDATIONS

Further investigation into cavity temperature distribution and internal surface emissivity and how each affects analytically determined radiative heat loss. Inquiries into convective heat loss from a cavity is necessary to consider how various wind condition will effect heat loss characteristics.
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<th>Flow SRI error</th>
<th>Flow SRI corr</th>
<th>Flow SRI corr error</th>
<th>Flow SRI corr err</th>
<th>T in</th>
<th>T out</th>
<th>T in</th>
<th>T out</th>
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<td>85.7</td>
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<td>1.27</td>
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<td>0.0016</td>
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<td>85.7</td>
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Appended 1: Phase One Temperature and Angle Test Results
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<th>Normalized Heat Loss</th>
<th>Low Heat Loss Error</th>
<th>Low Heat Loss Cost</th>
<th>Heat Loss Cost</th>
<th>Heat Loss Cost &amp; Radiation Cost</th>
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<th>Heat Loss Cost x Alpha</th>
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<th>Heat Loss Cost, Alpha &lt; 0.001</th>
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**Appendix 1:** Phase One: Temperature and Angle Test Results
Appendix 2: Phase Two; Aperture Size Test Result

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<th>Test Result</th>
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<tr>
<td>Sylphem Density (kN/m³)</td>
<td>error Density (%)</td>
</tr>
<tr>
<td>-------------------------</td>
<td>-------------------</td>
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<tr>
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Appendix 2: Phase Two: Aperture Size Test Result
Appendix 2: Phase Two; Aperture Size Test Result

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<td>P2</td>
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<tr>
<td>P3</td>
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<tr>
<td>P4</td>
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<td>Result 4</td>
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<tr>
<td>P5</td>
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<td>P6</td>
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<tr>
<td>P7</td>
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<td>Result 7</td>
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APPENDIX 3: LeQuere, Penot and Mirenayat Model Computer Program Listing

REM
REM P. LeQuere, F. Penot, & M. Mirenayat
REM
REM The LeQuere, Penot, & Mirenayat model is used here to predict the convective
losses from
REM a solar cavity receiver operating at various temperatures and receiver angles.
REM
PRINT "P. LeQuere, F. Penot, & M. Mirenayat Model for Predicting Convective Heat Loss"
PRINT
PRINT
REM
REM
REM ************************** Receiver Geometry **************************
REM
Re=.127 :REM end plate radius [m]
Rc=.33 :REM cavity radius [m]
Ro=.45 :REM receiver outside radius [m]
Lf=.292 :REM frustum length [m]
Lh=.254 :REM cylinder length hot [m]
Lc=.14 :REM cylinder length cold [m]
REM
REM ************************** Constants **************************
REM
pi=4*ATN(1)
g=9.810001 :REM gravitational acceleration [m/sec^2]
SB=5.6696E-08 :REM Stefan Boltzmann const. [W/m^2 K^4]
Cp=1006.86 :REM specific heat capacity of air at Ta [J/kg K]
Pf=1.19406 :REM density of air at Ta [kg/m^3]
e=.9 :REM emittance of cavity
kl=.04756 :REM insulation conductance [W/m-K] [.33B/h/ft^2/in]
t=.0899 :REM thickness of insulation [m] [3.5 in]
Ta=70 :REM ambient temperature [F]
Ta=(Ta+459.67)/1.8 :REM ambient temperature [K]
REM
REM *************** Angle constants ***************
DIM a(13),b(13)
FOR I=1 TO 13
READ a(I)
DATA .057,.047,.054,.046,.045,.048,.046,.092,.081,.064,.060,.068 5,.033,0
NEXT I
FOR I=1 TO 13
READ b(I)
DATA .353,.36,.37,.369,.368,.33,.331,.332,.316,.292,.302,0
NEXT I
REM
REM open clipboard file for transferring data to spread sheet
OPEN "CLIP:" FOR OUTPUT AS #1
100:
REM
REM CLS
PRINT "End Plate Radius [m] = ";Re
PRINT "Cavity Radius [m] = ";Rc
PRINT "Frustum Length [m] = ";Lf

106
PRINT "Hot Cylindrical Section Length [m] = ";Lh
PRINT "Cold Cylindrical Section Length [m] = ";Lc
PRINT "Ambient Temperature [K] = ";Ta
PRINT
REM **************************** Write to the clipboard ****************************
WRITE#1,",","End Plate Radius [m] = ",Re
WRITE#1,",","Cavity Radius [m] = ",Rc
WRITE#1,",","Frustum Length [m] = ",Lf
WRITE#1,",","Hot Cylindrical Section Length [m] = ",Lh
WRITE#1,",","Cold Cylindrical Section Length [m] = ",Lc
WRITE#1,",","Ambient Temperature [K] = ",Ta
WRITE#1,
REM **************************** Receiver Aperture Radius Loop ****************************
FOR I=1 TO 4
CLS
READ Ra
DATA .0762,.1524,.2286,.329
Da=2*Ra :REM aperture diameter [m]
Aa=pi*Ra^2 :REM aperture area [m^2]
REM **************************** Area Constants ****************************
REM In the following section Ah and Ar are calculated.
REM Ah is the total interior heated cavity surface area based on the tube bundle geometry.
REM At is the total cavity area.
Ah=pi*(Re+Rc)*(Lf^2+(Rc-Re)^2)^.5+2*pi*Rc*Lh
Ar=pi*Re^2+pi*(Rc^2-Ra^2)+2*pi*Rc*Lc
At=Ah+Ar
Ao=pi*((Rc+Ro)/2)^2+2*pi*((Rc+Ro)/2)+pi*(Rc^2-Ra^2)
REM **************************** Print Header ****************************
CLS
PRINT "Aperture Radius [m] = ";Ra
PRINT "Total Cavity Area [m^2] = ";
PRINT USING "###.####";At
PRINT "Total Heated Cavity Area [m^2] = ";
PRINT USING "###.####";Ah
PRINT "Total Refractory Cavity Area [m^2] = ";
PRINT USING "###.####";Ar
REM **************************** write header to clipboard ****************************
WRITE#1,","Aperture Radius [m] = ",Ra
WRITE#1,","Total Cavity Area [m^2] = ",At
WRITE#1,","Total Heated Cavity Area [m^2] = ",Ah
WRITE#1,","Total Refractory Cavity Area [m^2] = ",Ar
REM **************************** Operating Temperature Loop ****************************
REM
FOR Tmf=300 TO 600 STEP 100
WRITE#1, Rem The operating temperature is converted from F to K
Tm=(Tmf+459.67)/1.8
Twf=Tmf-100 :REM The refractory surfaces are assumed to be 100°F cooler
REM than the heated tube surfaces.
\[ Tw = \frac{(Tw_f + 459.67)}{1.8} \]

CLS

REM ******************** Print Table Header ***************
PRINT "T mean [K] = ";
PRINT USING "####.#"; Tm;
PRINT " [°F] = ";
PRINT USING "####."); Tmf
PRINT
PRINT ;TAB(3);" Angle";TAB(12);" Q conv";TAB(24);" Nu";TAB(35);" Gr"
PRINT ;TAB(1);" [degrees]";TAB(13);"[Watts]"
PRINT
REM **************************** write table header to clipboard ************
WRITE#1,","T mean [K] = ",Tm
WRITE#1,", [°F] = ",Tmf
WRITE#1,
WRITE#1," Angle"," Q conv"," Nu ",; Gr"
WRITE#1," [degrees]","[Watts]"
WRITE#1,
REM **************************** Beginning of Angle Loop ****************************
n=0
FOR phi = -90 TO 90 STEP 15
    z = phi*phi/180 REM convert angle to radian measure
REM **************************** Angle Function ****************************
n = n+1
REM ****************************
Tcav = (Tm*Ah + Tw*Ar)/At : REM area average cavity temperature [K]
L = 2*Ra
C = 1.1547E+19*Ta^4.4187 : REM gB/v^2.
k = .0071749261015++.000064030639041#*Ta
REM ******************** Heat Loss Calculations ********************
Gr = C*(Tcav-Ta)*L^3 : REM Grashof number
Pr = 7.814008749-.00037306809395#*Ta+5.2131644352D-07*Ta^2-2.
1272705278D-10*Ta^3
REM Pr = Prandtl number
Nu = a(n)*Gr^n : REM Nusselt number
h = Nu*k/L : REM heat transfer coefficient [W/m^2 K]
REM **************************** heat loss calculations ********************
Qc = h*At*(Tcav-Ta) : REM convective
REM **************************** Output Loop Results ********************
PRINT ;TAB(3);phi;
PRINT TAB(12);
PRINT USING "####.#"; Qc;
PRINT TAB(22);
PRINT USING "####.#"; Nu;
PRINT TAB(32);
PRINT USING "####.#####"; Gr
REM **************************** write to clipboard ****************************
WRITE#1, phi, Qc, Nu, Gr
REM **************************** End of Angle Loop ****************************
NEXT phi
REM ****************************
REM  *************** End of Temperature Loop ***************
   NEXT Tmf
REM  ********************** End of Aperture Radius Loop ******
REM  NEXl
REM  ********************** End of Aperture Radius Loop ******
CLOSE#1
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#### T mean [°F] vs [°C]

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#### Aperture Radius [m/m²] = 0.238

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Koenig and Marvin Model

The Koenig and Marvin method is used here to predict the convective losses from a cavity solar receiver operating at various temperatures and receiver angles.

```
PRINT " Koenig and Marvin Model for Predicting Convective Heat Loss"
PRINT
PRINT ******************* Receiver Geometry ********************
REM
Re=.127 :REM end plate radius [m]
Rc=.33 :REM cavity radius [m]
Ro=.45 :REM receiver outside radius [m]
Lf=.292 :REM frustum length [m]
Lh=.254 :REM cylinder length hot [m]
Lc=.14 :REM cylinder length cold [m]
REM
pi=4*ATN(1)' REM gravitational acceleration [m/sec^2]
g=9.810001 :REM Stefan Boltzmann const. [W/m^2 K^4]
SB=.6696E-08 :REM specific heat capacity of air at Ta [J/kg K]
Pf=1.19406 :REM density of air at Ta [kg/m^3]
e=.9 :REM emittance
ki=.04756 :REM thickness of insulation [m]
t=.0889 :REM insulation conductance [W/m-K] [.33B/h/ft^2/in]
Ta=70 :REM ambient temperature [°F]
Ta=(Ta+459.67)/1.8 :REM ambient temperature [K]
REM
OPEN "CLIP:" FOR OUTPUT AS #1
100:
REM Write to the clipboard
PRINT * End Plate Radius [m] = ";Re
PRINT * Cavity Radius [m] = ";Rc
PRINT * Frustum Length [m] = ";Lf
PRINT * Hot Cylindrical Section Length [m] = ";Lh
PRINT * Cold Cylindrical Section Length [m] = ";Lc
PRINT * Ambient Temperature [K] = ";Ta
REM
CLOSE
PRINT
PRINT ************** Print Constants **************
REM
PRINT "End Plate Radius [m] = ";Re
PRINT "Cavity Radius [m] = ";Rc
PRINT "Frustum Length [m] = ";Lf
PRINT "Hot Cylindrical Section Length [m] = ";Lh
PRINT "Cold Cylindrical Section Length [m] = ";Lc
PRINT "Ambient Temperature [K] = ";Ta
REM
WRITE#1, "End Plate Radius [m] = ";Re
WRITE#1, "Cavity Radius [m] = ";Rc
WRITE#1, "Frustum Length [m] = ";Lf
WRITE#1, "Hot Cylindrical Section Length [m] = ";Lh
WRITE#1, "Cold Cylindrical Section Length [m] = ";Lc
WRITE#1, "Ambient Temperature [K] = ";Ta
```

APPENDIX 5: Koenig and Marvin Model Computer Program Listing
**Receiver Aperture Radius Loop***************

```
FOR I=1 TO 4
CLS
READ Ra
DATA .0762,.1524,.2286,.329
Da=2*Ra :REM aperture diameter [m]
Aa=pi*Ra^2 :REM aperture area [m^2]
REM
```

In the following section Ah and Ar are calculated.
REM Ah is the total interior heated cavity surface area based on the tube bundle geometry.
REM Ar is the total interior refractory cavity surface.
REM At is the total cavity area.

```
Ah=pi*(Re+Rc)*(Lf^2+(Rc-Re)^2)^.5+2*pi*Rc*Lh
Ar=pi*Re^2+pi*(Rc^2-Ra^2)+2*pi*Rc*Lc
At=Ah+Ar
Ao=pi*((Rc+Ro)/2)^2+2*pi*((Rc+Ro)/2)+pi*(Rc^2-Ra^2)
```

**Print Header**********************

```
PRINT "Aperture Radius [m] = ";Ra
PRINT " Total Cavity Area [m^2] = ";
PRINT USING "###.####";At
PRINT " Total Heated Cavity Area [m^2] = ";
PRINT USING "###.####";Ah
PRINT " Total Refractory Cavity Area [m^2] = ";
PRINT USING "###.####";Ar
REM
```

**Operating Temperature Loop***************

```
FOR Tmf=300 TO 600 STEP 100
CLS
Tm=(Tmf+459.67)/1.8
Twf=Tmf-100 :REM The refractory surfaces are assumed to be 100°F cooler than the heated tube surfaces.
Tw=(Twf+459.67)/1.8
CLS
REM
```

**Print Table Header**********************

```
PRINT "T mean [Q]= ";Tm;
PRINT USING "####.#";Tmf
PRINT ;TAB(3);" AngleW;TAB(12);"Q convW;TAB(24);"Q totalW;TAB(34);"% ConvW;
PRINT TAB(45);" Nu ;TAB(56);" GrW;TAB(66);"P(a)W;TAB(76);"kW;TAB(86);"gB/v^2W;TAB(96);"PrW;
```

113
PRINT ;TAB(1);" [degrees]";TAB(13);"[Watts]";TAB(24);"[Watts]";TAB(36);"%"
PRINT
REM **************************** write table header to clipboard **************
REM
WRITE#1,"","T mean [K] = ",Tm
WRITE#1,"","[°F] = ",Tmf
WRITE1,#1,
WRITE1,#1," Angle","Q conv","Q total","% Conv"," Nu ","," Gr","P(n)
WRITE1,#1," [degrees]","[Watts]","[Watts]","%","","",""
WRITE1,#1,
REM **************************** Beginning of Angle Loop *********************
FOR a=0 TO 90 STEP 15
z=pi*a/180 :REM convert angle to radian measure
REM **************************** Angle Function ****************************
IF a>45 THEN 121
P=(COS(z))^3.2
GOTO 124
121 :
IF a=90 THEN 122
P=.707*(COS(z))^2.2
GOTO 124
122 :
P=0
124 :
REM **************************** Heat Loss Calculations *********************
REM
Tcav=(Tm*Ah+Tw*Ar)/At :REM area average cavity temperature [K]
Lcc=Rc/Ra
L=2*.5*Rc
Tp=(11*Tcav+3*Ta)/16 :REM air properties temperature [K]
C=1.1547E+19*TP^-4.4187 :REM gB/v^2.
k=.007174261015+.0000640306390411*TP
REM **************************** Heat Loss Calculations *********************
Gr=C*(Tcav-Ta)^3 :REM Grashof number
Pr=.7814008749-.00037306809395*TP+.52131644352D-07*TP^2-2.
1272705278D-10*TP^3
REM Pr = Prandtl number
C1=.52
a1=1.75
Nu=C1*P*Lcc^a1*(Gr*Pr)^.25 :REM Nusselt number
h=Nu*k/L :REM heat transfer coefficient [W/m^2 K]
REM **************************** heat loss calculations *********************
REM
Qr=pi*Ra^2*e*SB*(Tcav^4-Ta^4) :REM radiative
Qc=h*At*(Tcav-Ta) :REM convective
Qk=ki*Ao*(Tcav-Ta)/t :REM conductive
Qt=Qr+Qc+Qk
PQc=Qc/Qt*100
REM **************************** Output Loop Results ********************
PRINT ;TAB(3);a;
PRINT TAB(12);
PRINT USING "####.#";Qc;
PRINT;TAB(24);
PRINT USING "####.#";Qt;
PRINT TAB(34);
PRINT USING "##.#";PQc;

114
PRINT TAB(44);
PRINT USING "###.##";Nu;
PRINT TAB(54);
PRINT USING "###.###";Gr;
PRINT TAB(64);
PRINT USING "#.####";P;
PRINT TAB(74);
PRINT USING "#.####";k;
PRINT TAB(84);
PRINT USING "##.##";C;
PRINT TAB(94);
PRINT USING "#.####";Pr;
PRINT TAB(104);
PRINT USING "###.##";T;
REMM************** write to clipboard  ********************
WRITE#1,a,Qc,Qt,PQc,Nu,Gr,P
REMM~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~End of Angle Loop  ********************
NEXT a
REMM~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~Output Radiation & Conduction  ********************
PRINT*Radiation (Watts) = ";Qr
PRINT*Conduction (Watts) = ";Qk
REMM~~~~~~~~~~~~~~Write radiation & conduction to clipboard  ********************
WRITE#1,"",""Radiation (Watts) = ";Qr
WRITE#1,"",""Conduction (Watts) = ";Qk
REMM~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~End of Temperature Loop  ********************
NEXT Tmf
REMM~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~End of Aperture Radius Loop  ********************
NEXT I
REMM~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~
CLOSE#1
END
Appendix 6: Koenig and Marvin Model Heat Loss

### Table 1: Heat Loss Data

<table>
<thead>
<tr>
<th>Angle (degree)</th>
<th>Q_{conv} (Watt)</th>
<th>Q_{total} (Watt)</th>
<th>% Conv</th>
<th>Nu</th>
<th>Gr</th>
<th>Fp0</th>
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<tbody>
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### Table 2: Heat Loss Data

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<th>Q_{conv} (Watt)</th>
<th>Q_{total} (Watt)</th>
<th>% Conv</th>
<th>Nu</th>
<th>Gr</th>
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<tbody>
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### Table 3: Heat Loss Data

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### Table 4: Heat Loss Data

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### Table 5: Heat Loss Data

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<th>Q_{total} (Watt)</th>
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<th>Nu</th>
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### Appendix 6: Koenig and Marvin Model Heat Loss

<table>
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<tr>
<th>Angle (degrees)</th>
<th>Qconv (W/m²)</th>
<th>Qrad (W/m²)</th>
<th>Qcond (W/m²)</th>
<th>Qloss (W/m²)</th>
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Radiation (W/m²) = \( \frac{Q_{rad}}{A} \)

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<th>Qconv (W/m²)</th>
<th>Qrad (W/m²)</th>
<th>Qcond (W/m²)</th>
<th>Qloss (W/m²)</th>
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Radiation (W/m²) = \( \frac{Q_{rad}}{A} \)

<table>
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<th>Angle (degrees)</th>
<th>Qconv (W/m²)</th>
<th>Qrad (W/m²)</th>
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Radiation (W/m²) = \( \frac{Q_{rad}}{A} \)

<table>
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<th>Angle (degrees)</th>
<th>Qconv (W/m²)</th>
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Radiation (W/m²) = \( \frac{Q_{rad}}{A} \)

<table>
<thead>
<tr>
<th>Angle (degrees)</th>
<th>Qconv (W/m²)</th>
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Radiation (W/m²) = \( \frac{Q_{rad}}{A} \)
Appendix 6: Koenig and Marvin Model Heat Loss

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118
## Appendix 6: Koenig and Marvin Model Heat Loss

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**Radiation (Watts):** 171.8

**Conduction (Watts):** 178.7

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**Radiation (Watts):** 172.7

**Conduction (Watts):** 165.8

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**Radiation (Watts):** 133.9

**Conduction (Watts):** 137.9

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**Radiation (Watts):** 192.3

**Conduction (Watts):** 440.0
Appendix 7: Zone and Shear Plane Area Formulas

1. Zone Area Formulas

The receiver cavity is divided into two zones (Fig. 23). The boundary between the zones is formed by a horizontal plane cutting through the cavity at the upper lip of the aperture. The upper zone is assumed stagnate while the lower zone has active convective currents. Zone 1 area represents the internal surface area of the receiver above the horizontal plane. The zone 2 area represents the internal surface area of the receiver below the horizontal plane. The zone 1 and zone 2 areas vary with receiver angle for a given receiver geometry. The following formulas describe the zone 1 and zone 2 surface areas of the receiver cavity.

![Diagram of cavity zones](image)

Figure 23. Cavity zones areas.
The receiver internal geometry is divided into five sections representing the hot and cold surfaces in the receiver (Fig. 24). The hot surfaces are actively heated. The cold surfaces represent the refractory surfaces. Section 1 is the circular plate at the end of the frustum. Section 2 is the frustum portion of the tube bundle. Section 3 is the cylindrical portion of the tube bundle. Section 4 is the short refractory portion of the cylindrical section. Section 5 is the refractory ring that forms the aperture.

![Cavity sections diagram](image)

Figure 24. Cavity sections.

As the receiver is rotated through various angles, each section of the internal receiver geometry may be divided by the horizontal plane that cuts through the upper inside edge of section 5. The critical angles represent limits for the various algebraic expression of the zone areas (Fig. 69). The following formulas define the portion of the area of each section that is in zone 1 for a given receiver angle range. The remaining surface area of each section in zone 2 is determined by subtracting the zone 1 area from the total surface area for that section.
where:

$$\phi_1 = \tan^{-1} \left[ \frac{R_a - R_e}{\lambda_{cc} + \lambda_{hc} + \lambda_f} \right]$$  \hspace{1cm} (115)$$

$$\phi_2 = \tan^{-1} \left[ \frac{R_a + R_e}{\lambda_{cc} + \lambda_{hc} + \lambda_f} \right]$$  \hspace{1cm} (116)$$

$$\phi_3 = \tan^{-1} \left[ \frac{R_a + R_e}{\lambda_{cc} + \lambda_{hc}} \right]$$  \hspace{1cm} (117)$$

$$\phi_4 = \tan^{-1} \left[ \frac{R_a + R_e}{\lambda_{cc}} \right]$$  \hspace{1cm} (118)$$

where:

$$R_e = \text{radius of the end plate}$$
(a) section 1

Range \( 0 \leq \theta \leq \theta_1 \)

\[
\text{Area} = 0
\]

Range \( \theta_1 \leq \theta \leq \theta_2 \)

\[
\text{Area} = R_e^2 \cos^{-1} \left[ \frac{R_a - (\kappa_{cc} + \kappa_{hc} + \kappa_f) \tan \theta}{R_e} \right] + \left[ (\kappa_{cc} + \kappa_{hc} + \kappa_f) \tan \theta - R_a \right]
\]

\[\left\{ 2R_e \left[ (\kappa_{cc} + \kappa_{hc} + \kappa_f) \tan \theta - R_a + R_e \right] - \left[ (\kappa_{cc} + \kappa_{hc} + \kappa_f) \tan \theta - R_a + R_e \right]^2 \right\}^{1/2}\]

Range \( \theta_2 \leq \theta \leq \frac{\pi}{2} \)

\[
\text{Area} = \pi R_e^2
\]

(b) section 2

Range \( 0 \leq \theta \leq \theta_1 \)

\[
\text{Area} = \left\{ \frac{\left[ (\kappa_{cc} - \kappa_f)R_a - (\kappa_{cc} + \kappa_{hc}) \right] \sin \theta}{\sin \left[ \frac{\pi}{2} - \theta + \tan^{-1} \left( \frac{R_e - R_a}{\kappa_f} \right) \right]} + \frac{(R_e - R_a)}{\sin \left[ \tan^{-1} \left( \frac{R_e - R_a}{\kappa_f} \right) \right]} \right\}
\]
\[ R_e \cos^{-1} \left[ \frac{R_a - (\ell_{cc} + \ell_{hc}) \tan \varnothing}{R_e} \right] \]  

Range \( \varnothing_1 \leq \varnothing \leq \varnothing_2 \)

\[ \text{Area} = \sqrt{\ell_f^2 + (R_c - R_e)^2} \left\{ R_e \cos^{-1} \left[ \frac{R_a - (\ell_{cc} + \ell_{hc}) \tan \varnothing}{R_e} \right] \right. + R_e \cos^{-1} \left[ \frac{R_a - (\ell_{cc} + \ell_{hc}) \tan \varnothing}{R_c} \right] \} \]  

Range \( \varnothing_2 \leq \varnothing \leq \varnothing_3 \)

\[ \text{Area} = \left\{ \pi R_e + \pi \left\{ R_e + a \sin \left[ \tan^{-1} \left( \frac{R_c - R_e}{\ell_f} \right) \right] \right\} \right\} a + \left\{ R_e \cos^{-1} \left[ \frac{R_a - (\ell_{cc} + \ell_{hc}) \tan \varnothing}{R_c} \right] \right. + \pi \left\{ R_e + a \sin \left[ \tan^{-1} \left( \frac{R_c - R_e}{\ell_f} \right) \right] \right\} \} \]  

\[ \left[ \sqrt{(R_c - R_e)^2 + \ell_f^2} - a \right] \]  

(123)

where:

\[ a = \frac{\sqrt{(R_e + R_a)^2 + (\ell_{cc} + \ell_{hc} + \ell_f)^2} \sin \left[ \varnothing - \tan^{-1} \left( \frac{R_a + R_e}{\ell_{cc} + \ell_{hc} + \ell_f} \right) \right]}{\sin \left[ \pi - \varnothing - \tan^{-1} \left( \frac{R_c - R_e}{\ell_f} \right) \right]} \]  

Range \( \varnothing_3 \leq \varnothing \leq \frac{\pi}{2} \)

\[ \text{Area} = \pi (R_c + R_e) \sqrt{(R_c - R_e)^2 + \ell_f^2} \]  

(125)

(c) section 3

Range \( 0 \leq \varnothing \leq \varnothing_3 \)
\[ \text{Area} = R_c \cdot \lambda_{hc} \left\{ \cos^{-1} \left[ \frac{R_a - \lambda_{cc} \tan \varnothing}{R_c} \right] + \cos^{-1} \left[ \frac{R_a - (\lambda_{cc} + \lambda_{hc}) \tan \varnothing}{R_c} \right] \right\} \]

Range \( \varnothing_3 \leq \varnothing \leq \varnothing_4 \) \hspace{1cm} (126)

\[ \text{Area} = 2 \pi R_c \left[ \lambda_{cc} + \lambda_{hc} - \frac{(R_a + R_c)}{\tan \varnothing} \right] + R_c \left[ \frac{(R_a + R_c)}{\tan \varnothing} - \lambda_{cc} \right] \left\{ \pi + \cos^{-1} \left[ \frac{R_a - \lambda_{cc} \tan \varnothing}{R_c} \right] \right\} \]

Range \( \varnothing_4 \leq \varnothing \leq \frac{\pi}{2} \) \hspace{1cm} (127)

\[ \text{Area} = 2 \pi R_c \cdot \lambda_{hc} \] \hspace{1cm} (128)

(d) section 4

Range \( 0 \leq \varnothing \leq \varnothing_4 \)

\[ \text{Area} = R_c \cdot \lambda_{cc} \left\{ \cos^{-1} \left[ \frac{R_a}{R_c} \right] + \cos^{-1} \left[ \frac{R_a - \lambda_{cc} \tan \varnothing}{R_c} \right] \right\} \]

Range \( \varnothing_4 \leq \varnothing \leq \frac{\pi}{2} \) \hspace{1cm} (129)

\[ \text{Area} = 2 \pi R_c \left[ \lambda_{cc} - \frac{(R_a + R_c)}{\tan \varnothing} \right] + R_c \left\{ \pi + \cos^{-1} \left[ \frac{R_a}{R_c} \right] \right\} \left( \frac{R_a + R_c}{\tan \varnothing} \right) \]

\[ \text{Area} = 2 \pi R_c \left[ \lambda_{cc} - \frac{(R_a + R_c)}{\tan \varnothing} \right] + R_c \left\{ \pi + \cos^{-1} \left[ \frac{R_a}{R_c} \right] \right\} \left( \frac{R_a + R_c}{\tan \varnothing} \right) \]

(e) section 5

Range \( 0 \leq \varnothing \leq \frac{\pi}{2} \)

\[ \text{Area} = R_c^2 \cos^{-1} \left[ \frac{R_a}{R_c} \right] - R_a \sqrt{R_c^2 - R_a^2} \]

Range \( \varnothing = \frac{\pi}{2} \) \hspace{1cm} (131)
The shear plane area is the area of the horizontal plane within the cavity (Fig. 25). The shear plane area is divided into two sections. The first section is formed by the horizontal plane cutting through the cylindrical portion of the receiver cavity. Not all of the horizontal plane in the cylindrical portion participates in the convective heat loss. The sides of the aperture reduce the effective shear plane area by restricting flow along the horizontal plane at the sides of the cavity near the aperture. The shear plane expands parabolically from the upper lip of the aperture in the horizontal plane. The second section is formed where the horizontal plane cuts the frustum portion of the receiver cavity. The following formulas describe the shear plane area in the specified portion of the cavity for a given receiver angle.

\[
\text{Area} = \pi \left( R_c^2 - R_a^2 \right)
\]

(132)

2. Shear Plane Area

Figure 25. View looking down showing the effective shear plane area.
The angles in figure 70 are defined as follows:

\[ \phi_1 = \tan^{-1} \left[ \frac{R_a}{\ell_{cc} + \ell_{hc}} \right] \]  \hspace{1cm} (133)

\[ \phi_2 = \tan^{-1} \left[ \frac{R_a + R_c}{\ell_{cc} + \ell_{hc}} \right] \]  \hspace{1cm} (134)

Range \[ 0 \leq \phi \leq \phi_1 \]

\[ D = \frac{(\ell_{cc} + \ell_{hc})}{\cos \phi} \]  \hspace{1cm} (135)

\[ L = 2\sqrt{R_e^2 - R_a^2} \]  \hspace{1cm} (136)

\[ L_c = 2\sqrt{R_a^2 - (\ell_{cc} + \ell_{hc}) \tan \phi} \]  \hspace{1cm} (137)
\[
\frac{\text{Area}}{2D} = \frac{\frac{3}{2D} + \frac{3}{2D}}{2D} = \frac{3}{2D}
\]

\[
\phi \sin \frac{R}{\phi} = D
\]

\[
0 = I
\]

\[
I = 2R
\]

\[
\frac{\phi \sin \frac{R}{\phi}}{R} = D
\]

\[
L = 2R
\]

\[
\frac{2 \sqrt{R^2 + 2L}}{2} \leq \phi \leq 2 \phi
\]

\[
\text{Range}
\]

\[
\frac{\text{Area}}{2D} = \frac{\frac{3}{2D} + \frac{3}{2D}}{2D} = \frac{3}{2D}
\]

\[
\phi \sin \left( \frac{R}{\phi} + \frac{R}{\phi} \right) = D
\]

\[
I = 2R
\]

\[
\phi \sin \frac{R}{\phi} = D
\]

\[
I = 2R
\]

\[
\frac{2 \sqrt{R^2 + 2L}}{2} = L
\]

\[
\phi \leq \phi \leq \phi
\]

\[
\text{Range}
\]

\[
\frac{\text{Area}}{2D} = \frac{\frac{3}{2D} + \frac{3}{2D}}{2D} = \frac{3}{2D}
\]
The angles in figure 71 are defined as follows:

\[
\theta_1 = \tan^{-1} \left[ \frac{R_a - R_e}{\ell_{cc} + \ell_{hc} + \ell_f} \right] \tag{152}
\]

\[
\theta_2 = \tan^{-1} \left[ \frac{R_a + R_e}{\ell_{cc} + \ell_{hc} + \ell_f} \right] \tag{153}
\]

\[
\theta_3 = \tan^{-1} \left[ \frac{R_a + R_e}{\ell_{cc} + \ell_{hc}} \right] \tag{154}
\]

Range \[0 \leq \theta \leq \theta_1\]

\[
L = 2\sqrt{R_a^2 - \left[ R_a - (\ell_{cc} + \ell_{hc}) \tan\theta \right]^2} \tag{155}
\]

\[
D = \ell_f \left[ R_c + (\ell_{cc} + \ell_{hc}) \tan\theta - R_a \right] \frac{R_e - R_a}{(R_c - R_e) \cos \theta - \ell_f \sin \theta} \tag{156}
\]
\[
\text{Area}_{\text{shear}} = \frac{2D}{3} \left[ \frac{L^2 + LL_e + L_e^2}{L + L_e} \right]
\]  

Range \( \phi_1 \leq \phi \leq \phi_2 \)

\[
L = 2\sqrt{R_c^2 - [R_a - (L_{cc} + L_{hc}) \tan \phi]^2}
\]  
\[
L_e = 2\sqrt{R_c^2 - [R_a - (L_{cc} + L_{hc} + L_f) \tan \phi]^2}
\]

\[
D = \frac{L_f}{\cos \phi}
\]

\[
\text{Area}_{\text{shear}} = \frac{2D}{3} \left[ \frac{L^2 + LL_e + L_e^2}{L + L_e} \right]
\]  

Range \( \phi_2 \leq \phi \leq \phi_3 \)

\[
L = 2\sqrt{R_c^2 - [R_a - (L_{cc} + L_{hc}) \tan \phi]^2}
\]

\[
L_e = 0
\]

\[
D = \frac{L_f [R_a - (L_{cc} + L_{hc}) \tan \phi + R_a]}{(R_c - R_e) \cos \phi + L_f \sin \phi}
\]

\[
\text{Area}_{\text{shear}} = \frac{2D}{3} \left[ \frac{L^2 + LL_e + L_e^2}{L + L_e} \right]
\]

Range \( \phi_3 \leq \phi \leq \frac{\pi}{2} \)

\[
\text{Area}_{\text{shear}} = 0
\]

The total shear area in the cavity at any one angle is the sum of the shear areas of the cylindrical section and the frustum section.
Appendix 8: Clausing's Model Computer Program Listing

```
REMCLAUSING'S METHOD
REM5 sections program w/ shear plane area
REM
REM The Clausing method is used here to predict the convective, radiative,
REM and conductive losses from a cavity solar receiver operating at various
REM temperatures and receiver angles.
REM
PRINT * Clausing's Method of Predicting Heat Losses*
PRINT
REM
REM
REM temperatures and receiver angles.
REM
PA=127 :REM end plate radius [m]
Rc=.33 :REM cavity radius [m]
Lf=.292 :REM frustum length [m]
Lh=.254 :REM cylinder length hot [m]
Lc=.14 :REM cylinder length cold [m]
REM
REM End Plate Radius [m] = ;Re
REM Cavity Radius [m] = ;Rc
REM Frustrum Length [m] = ;Lf
REM Hot Cylindrical Section Length [m] = ;Lh
REM Cold Cylindrical Section Length [m] = ;Lc
```

131
**Ambient Temperature** [K] = Ta

**End Plate Radius** [m] = Re

**Cavity Radius** [m] = Rc

**Frustrum Length** [m] = Lf

**Hot Cylindrical Section Length** [m] = Lh

**Cold Cylindrical Section Length** [m] = Lc

**Ambient Temperature** [K] = Ta

---

**FOR** I = 1 TO 4

**IF** Pflag = 1 THEN 58

LPRINT CHR$(12)

**READ** Ra

DATA .0762,.1524,.2286,.329

Da=2*Ra :REM aperture diameter [m]

Aa=pi*Ra^2 :REM aperture area [m^2]

**ANGLE LIMITS**

In this section the 3 angle limits, z1, z2, z3, and z4 are calculated.

z1=ATN((Ra-Re)/(Lf+Lc+Lh)) :pl=z1*180/pi

z2=ATN((Ra+Re)/(Lf+Lc+Lh)) :p2=z2*180/pi

z3=ATN((Ra+Rc)/(Lc+Lh)) :p3=z3*180/pi

z4=ATN((Ra+Rc)/(Lc+Lh)) :p4=z4*180/pi

z5=ATN(Ra/(Lc+Lh)) :p5=z5*180/pi

---

**AREA CONSTANTS**

In the following section Ah and Ac are calculated.

Ah is the total interior heated cavity surface area based on the tube bundle geometry.

Ar is the total interior refractory cavity surface.

Aha is the interior heated cavity surface area in zone 2, the convective zone.

Ara is the interior refractory cavity surface area in zone 2, the convective zone.

AT is the total cavity area.

Ah=pi*(Re+Rc)*((Lf^2+Rc-Re)^2)^.5+2*pi*Rc*Lh

Ar=pi*Re^2+pi*(Rc^2-Ra^2)+2*pi*Rc*Lc

AT=Ah+Ar

---

**Print Header**

CLS

LPRINT "Aperture Radius [m] = Ra"

LPRINT "Total Cavity Area [m^2] = "

LPRINT USING "###.####";AT

LPRINT "Total Heated Cavity Area [m^2] = "

LPRINT USING "###.####";Ah
LPRINT "Total Refractory Cavity Area [m^2] = ";
LPRINT USING "#####.#####"; Ar

60 :
REM  ****************** write header to clipboard ******************
WRITE#1, ";"; "Aperture Radius [m] = ", Ra
WRITE#1, ";"; "Total Cavity Area [m^2] = ", AT
WRITE#1, ";"; "Total Heated Cavity Area [m^2] = ", Ah
WRITE#1, ";"; "Total Refractory Cavity Area [m^2] = ", Ar
REM  ****************** Operating Temperature Loop ******************
REM
FOR Tmf=300 TO 600 STEP 100
WRITE#1,
REM  The operating temperature is converted from F to K
Tm=Tmf+459.67/1.8
TWF=Tmf-100
REM  The refractory surfaces are assumed to be 100°F cooler
REM  than the heated tube surfaces.
Tw=(TWF+459.67)/1.8
CLS
REM
IF Pflag=1 THEN 62
REM  **************** Print Table Header ****************
LPRINT "T mean [K] = ";
LPRINT USING "####ió#"; Tm;
LPRINT "[°F] = ";
LPRINT USING "#####.#####"; Tmf
LPRINT
LPRINT ;TAB(3); "Angle";TAB(12); "Tube Area";TAB(22); "Refrac. Area";TAB(34); "Q conv";
LPRINT;TAB(44); "Q total"; TAB(54); "% Conv"; TAB(65); "Nu "; TAB(76); "Gr"; TAB(87); "Ashear";
LPRINT ;TAB(1); "[degrees]";TAB(13); "[m^2]";TAB(23); "[m^2]";TAB(34); "[Watts]" ;
LPRINT;TAB(44); "[Watts]";TAB(56); "%";TAB(88); "[m^2]"
LPRINT
REM  ****************** write table header to clipboard ***********
WRITE#1, ";"; "T mean [K] = "; Tm
WRITE#1, ";"; "[°F] = "; Tmf
WRITE#1,
REM  ****************** Beginning of Angle Loop *****************
FOR A=0 TO 90 STEP 15
z=pi*A/180
REM  The receiver cavity is divided into 5 sections to accommodate the
REM  zone area calculations. The sections are defined as follows;
REM  section 1 = end plate
REM  section 2 = frustum
REM  section 3 = hot cylinder
REM  section 4 = cold cylinder
REM  section 5 = ring

133
AZ1 = area of section 1 in zone 1
AZ2 = area of section 2 in zone 1
AZ3 = area of section 3 in zone 1
AZ4 = area of section 4 in zone 1
AZ5 = area of section 5 in zone 1

**********SECTION 1************

IF z>z1 THEN 101
  AZ1=0
  GOTO 201

101 :  
  IF z>z2 THEN 102
    x=(Ra-(Lf+Lc+Lh)*TAN(z))/Rc
    cx=-ATN(x/SQR(-x*x+1))+1.5708
    m=(Lf+Lc+Lh)*TAN(z)-Ra+Re
    AZ1=Re^2*cx+((Lf+Lc+Lh)*TAN(z)-Ra)*SQR(2*Re*m-m^2)
    GOTO 201
  102 :
  201 :

**********SECTION 2************

IF z>z1 THEN 202
  x=(Ra-(Lc+Lh)*TAN(z))/Rc
  cx=-ATN(x/SQR(-x*x+1))+1.5708
  m=ATN((Rc-Re)/Lf)
  AZ2=((Rc-Re)/Lf)*((Rc-Re)/Lf)+(Lc+Lh)*SIN(z)/SIN(pi/2-z-m)
  AZ2=(AZ2+(Rc-Re)/SIN(m))*Rc*cx
  GOTO 251

202 :  
  IF z>z3 THEN 203
    x1=(Ra-(Lc+Lf+Lh)*TAN(z))/Re
    cx1=-ATN(x1/SQR(-x1*x1+1))+1.5708
    x2=(Ra-(Lc+Lh)*TAN(z))/Rc
    cx2=-ATN(x2/SQR(-x2*x2+1))+1.5708
    AZ2=SQR(Lf^2+(Rc-Re)^2)*(Rc*cx1+Rc*cx2)
    GOTO 251

203 :  
  IF z>z3 THEN 204
    m=ATN((Rc-Re)/Lf)
    n=ATN((Ra+Re)/(Lc+Lf+Lh))
    l=SQR((Lc+Lf+Lh)*2+(Ra+Re)^2)*SIN(z-n)/SIN(pi-z-m)
    x=(Ra-(Lc+Lh)*TAN(z))/Rc
    cx=-ATN(x/SQR(-x*x+1))+1.5708
    AZ2=(pi*Re+pi*(Re+l*SIN(m)))*l
    AZ2=AZ2+(Rc*cx+pi*(Re+l*SIN(m)))*(SQR(Lf^2+(Rc-Re)^2)-l)
    GOTO 251

204 :
  AZ2=pi*(Re+Rc)*SQR(Lf^2+(Rc-Re)^2)

251 :

**********SECTION 3************

IF z>z4 THEN 253
IF z>z3 THEN 252
  x1=(Ra-Lc*TAN(z))/Rc
  x2=(Ra-(Lc+Lh)*TAN(z))/Rc
  cx1=-ATN(x1/SQR(-x1*x1+1))+1.5708
  cx2=-ATN(x2/SQR(-x2*x2+1))+1.5708
AZ3=Rc*(cx1+cx2)*Lh
GOTO 301
252 :
  m=(Ra+Rc)/TAN(z)
  x=(Ra-Lc*TAN(z))/Rc
  cx=-ATN(x/SQR(-x^2+1))+1.5708
  AZ3=2*pi*Rc*(Lh+Lc-m)+Rc*(m-Lc)*(pi+cx)
GOTO 301
253 :
  AZ3=2*pi*Rc*Lh
301 :
REM ****************** SECTION 4 **********************
IF z>z4 THEN 302
  x1=Ra/Rc
  x2=(Ra-Lc*TAN(z))/Rc
  cx1=-ATN(x1/SQR(-x1^2+1))+1.5708
  cx2=-ATN(x2/SQR(-x2^2+2)+1)+1.5708
  AZ4=Rc*(cx1+cx2)*Lc
GOTO 401
302 :
  x=Ra/Rc
  cx=-ATN(x/SQR(-x*x+1)) + 1.5708
  m=(Ra+Rc)/TAN(z)
  AZ4=2*pi*Rc*(Lc-m)+Rc*m*(pi+cx)
401 :
REM ****************** SECTION 5 **********************
REM ************** Shear Area Calculations **************
REM ************** Cylindrical Shear Area Section **************
800 :
  IF z>z5 THEN 810
  L1=0
  Le=2*(Ra-(Ra-Lc+Lh)*TAN(z))^2)^.5
  D=(Lc+Lh)/COS(pi*A/180)
  Acshear=2*D*(L1^2 +L1*Le+Le^2)/(3*(L1+Le))
GOTO 850
810 :
  IF z>z3 THEN 820
  L1=0
  Le=2*Rc
  D=Ra/SIN(z)
  L2=2*Rc
  Le2=2*(Ra-(Ra-(Lc+Lh)*TAN(z)))^2)^.5
  D2=((Lc+Lh)*TAN(z)-Ra)/SIN(z)
  135
\[ A_{\text{shear}} = 2D^* \left( L_1^2 + L_1^*L_e + L_e^2 \right) / \left( 3(L_1 + L_e) \right) + 2D^* \left( L_2^2 + L_2^*L_e + L_e^2 \right) / \left( 3(L_2 + L_e) \right) \]

GOTO 850

820 :
L1 = 0
Le = 2*Rc
D = Ra/SIN(z)
L2 = 2*Rc
Le2 = 0
D2 = Rc/SIN(z)
Acshear = 2*D*(L1^2 + L1^*Le + Le^2)/(3*(L1 + Le)) + 2*D*(L2^2 + L2^*Le + Le^2)/(3*(L2 + Le))

850 :

---------- Frustum Shear Area Section ----------

REM

IF z > z1 THEN 860
L1 = 2*(Rc^2 - (Ra - (Lc + Lh)^TAN(z))^2)^.5
Le = 0
D = Lf*(Rc + (Lc + Lh)*TAN(z) - Ra)/((Rc - Re)^COS(z) - Lf^SIN(z))
Afshear = 2*D*(L1^2 + L1^*Le + Le^2)/(3*(L1 + Le))
GOTO 890

860 :
IF z > z2 THEN 870
L1 = 2*(Rc^2 - (Ra - (Lc + Lh)^TAN(z))^2)^.5
Le = 2*(Re^2 - (Ra - (Lc + Lh + Lf)^TAN(z))^2)^.5
D = Lf/COS(z)
Afshear = 2*D*(L1^2 + L1^*Le + Le^2)/(3*(L1 + Le))
GOTO 890

870 :
IF z > z3 THEN 880
L1 = 2*(Rc^2 - (Ra - (Lc + Lh)^TAN(z))^2)^.5
Le = 0
D = Lf*(Rc - (Lc + Lh)*TAN(z) + Ra)/(COS(z)*Rc - Re + Lf^SIN(z))
Afshear = 2*D*(L1^2 + L1^*Le + Le^2)/(3*(L1 + Le))
GOTO 890

880 :
Afshear = 0

890 : Ashear = Acshear + Afshear

REM

Heat Loss Calculations

La = Da*COS(z) :REM projected length of aperture [m]
IF La < 0 THEN La = 0
T1 = .08*(Tm-Ta) + 273 :REM first guess temp of air leaving the aperture
XP = 200 :REM
350 :
Tc = T1
GOTO Temp
370 :
Q1 = DQ
380 :
Tc = XP
GOTO Temp
400 :
Q2 = DQ
Tx = (T1*Q2 - XP*Q1)/(Q2 - Q1)
Tc = Tx

136
Temp:

\[ \text{Tb} = \frac{(Tc + Ta)}{2} \] : REM bulk temp inside cavity [K]

\[ B1 = \frac{1}{\text{Tb}} \] : REM coefficient of volume expansion [1/K]

\[ \text{Tf} = \frac{(Tm + \text{Tb})}{2} \] : REM film temp [K]

\[ B2 = \frac{1}{\text{Tf}} \] : REM coefficient of volume expansion [1/K]

\[ \text{Pa} = 352.95/\text{Tf} \] : REM absolute viscosity [kg/m-sec]

\[ V = \frac{1}{\text{Pa}} \] : REM kinematic viscosity [m^2/sec]

\[ \text{B1} = \frac{1}{\text{Tf}} \] : REM film temp from [K] to [°F]

\[ k = 0.00679 + 3.5353E-05 \times \text{Tf} \] : REM thermal conductivity [W/m-K]

\[ \text{Vb} = \sqrt{g \times B1 \times \text{Tv} \times \text{La}} \] : REM characteristic velocity due to buoyancy [m/sec]

\[ \text{Va} = 0.5 \times \text{Vb} \] : REM average velocity [m/sec]

\[ \text{Qc} = (\text{Pr} \times 0.5 \times \text{Aa} \times \text{Va} \times \text{Cp} \times (\text{Tc} - \text{Ta})) \] : REM heat transfer through aperture [W]

\[ \text{Gr} = g \times B2 \times (\text{Tm} - \text{Tb}) \times \text{La}^3 / \text{V}^2 \] : REM Grashof number

\[ \text{Pr} = 7 \] : REM Prandtl number

\[ \text{Nu} = 1.1 \times (\text{Gr} \times \text{Pr})^{0.333} \] : REM Nusselt number

\[ h = \frac{\text{Nu} \times k}{\text{Da}} \] : REM heat transfer coefficient [W/m^2 K]

\[ \text{Qi} = h \times \text{Aa} \times (\text{Tm} - \text{Tb}) + h \times \text{Ara} \times (\text{Tw} - \text{Tb}) + h \times \text{Ashear} \times (\text{Tm} - \text{Tb}) \] : REM heat transfer within the aperture [W]

\[ \text{DQ} = \text{Qi} - \text{Qc} \]

IF \( \text{Tc} = \text{T1} \) THEN 370
IF \( \text{Tc} = \text{XP} \) THEN 400
IF \( \text{ABS(DQ)} < 0.1 \) THEN 740
IF \( \text{DQ} < 0 \) THEN GOTO 720

\( \text{XP} = \text{Tx} \)
GOTO 380

720:
\( \text{T1} = \text{Tx} \)
GOTO 350

740:
\[ \text{Qr} = Aa \times e \times Sb \times ((Ac / (Ah + Ac)) \times (Tw^4 - Ta^4) + (Ah / (Ah + Ac)) \times (Tm^4 - Ta^4)) \] : REM radiative loss [W]

\[ \text{Qk} = (k \times \text{I} / \text{t}) \times (Ah \times (\text{Tm} - \text{Ta}) + Ar \times (\text{Tw} - \text{Ta})) \]

\[ \text{QT} = \text{Qc} + \text{Qr} + \text{Qk} \] : REM total heat loss from receiver

\[ \text{PQc} = 100 \times \text{Qc} / \text{QT} \] : REM %convective

REM

*************** Output Loop Results ***************

REM

LPRINT ;TAB(3);A;
LPRINT TAB(12);
LPRINT USING "###.###";Aha;
LPRINT TAB(22);
LPRINT USING "###.###";Ara;
LPRINT TAB(32);
LPRINT USING "####.#";Qc;
LPRINT;TAB(42);
LPRINT USING "####.#";QT;
LPRINT TAB(54);
LPRINT USING "###.#";PQc;
LPRINT TAB(64);
LPRINT USING "###.#";Nu;
LPRINT TAB(74);
LPRINT USING "####.#";Gr;
LPRINT TAB(88);
LPRINT USING "##.##";Ashear
66:
REM ************************ write to clipboard ************************
WRITE#1,A,Aha,Ara,Qc,QT,PQc,Nu,Gr,Ashear
REM ******************************** End of Angle Loop ************************
NEXTA
REM ************************************************************
IF Pflag=1 THEN 68
REM ************************** Output Radiation & Conduction ********
LPRINT*Radiation (Watts) = ";,Qr
LPRINT*Conduction (Watts) = ";,Qk
68:
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WRITE#1,"","Radiation (Watts) = ";,Qr
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REM **************************** End of Temperature Loop ******
NEXT Tmf
REM *************************************************
REM **************************** End of Aperture Radius Loop ******
NEXT I
REM *************************************************
CLOSE#1
END
### Appendix 9: Claussung Model Heat Loss Data

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### Appendix 9 Clauing Model Heat Loss Data

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140
Appendix 9 Clauing Model Heat Loss Data

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- Conduction (Watt) = 910.0

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- Radiation (Watt) = 414.5
- Conduction (Watt) = 137.2

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- Conduction (Watt) = 218.3

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- Radiation (Watt) = 1042.3
- Conduction (Watt) = 229.7

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## Appendix 9 Clauising Model Heat Loss Data

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**Conduction (Watt) = 9.42**

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**Radiation (Watt) = 858.5**
**Conduction (Watt) = 125.2**

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**Radiation (Watt) = 1413.2**
**Conduction (Watt) = 166.2**

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**Radiation (Watt) = 271.7**
**Conduction (Watt) = 201.2**
The Siebers & Kraabel method is used here to predict the convective
losses from a solar cavity receiver operating at various temperatures and
receiver angles.

The program allows some variations in receiver geometry. These
variables are inputted in this section of the program. The characteristic
length, as called for in the reference, is simply the cavity diameter
given here as 'C'.

**Receiver Geometry**

\[
\begin{align*}
R_e &= 0.127 & \text{end plate radius [m]} \\
R_c &= 0.33 & \text{cavity radius [m]} \\
L_f &= 0.292 & \text{frustum length [m]} \\
L_h &= 0.254 & \text{cylinder length hot [m]} \\
L_c &= 0.14 & \text{cylinder length cold [m]} \\
C_l &= 2R_c & \text{characteristic length [m]}
\end{align*}
\]

**Constants**

\[
\begin{align*}
\pi &= 4\times ATN(1) \\
T_a &= 70 & \text{ambient temperature [°F]}
\end{align*}
\]

**Receiver Aperture Radius Loop**

\[
\begin{align*}
r_4 &= 0.0762, 0.1524, 0.2286, 0.329 \\
D_a &= 2r \\
p_1 &= \frac{\pi}{180} z_1 \\
p_2 &= \frac{\pi}{180} z_2 \\
p_3 &= \frac{\pi}{180} z_3 \\
p_4 &= \frac{\pi}{180} z_4
\end{align*}
\]


REM ************************************************ Area Constants ************************************************
REM In the following section area 1, area 2, and area 3 are calculated.
REM A1 is the total interior cavity surface area.
REM A2 is the total interior cavity surface area minus the lower lip.
REM A3 is the interior cavity surface area below the horizontal plane cutting through the receiver at the top of the aperture.
REM
Ah = π*(Re + Rc)*((Lf*2 + (Rc - Re)*Rc*2)^0.5 + 2*π*Re*Lh
Ar = π*Re^2 + π*(Rc^2 - Ra^2) + 2*π*Rc*Lc
A1 = Ah + Ar
A2 = A1 - Ra^2*(ATN((Ra/Rc)/SQRT(-Ra/Rc)*(Ra/Rc + 1))) + 1.5708 + Ra

REM Area 3 is a function of the receiver angle and is determined by calculating the total area in zone 1, above the horizontal plane, and then subtracting this value from the total area.
REM
PRINT "Siebers & Kraabel Method"
PRINT
PRINT *End Plate Radius [m] = ";Re
PRINT *Cavity Radius [m] = ";Rc
PRINT *Aperture Radius [m] = ";Ra
PRINT *Frustum Length [m] = ";Lf
PRINT *Hot Cylindrical Section Length [m] = ";Lh
PRINT *Cold Cylindrical Section Length [m] = ";Lc
PRINT *Tamb = ";Ta
PRINT "Total Area [m^2] = ";A1
REM
WRITE#1,
WRITE#1,. . . ,"Aperture Radius [m] = ";Ra
WRITE#1,. . . ,"Total Area [m^2] = ";A1
REM
FOR Th=300 TO 600 STEP 100
REM mean system operating temperature of receiver [°F]
TR=Th-100
TW=(Th*Ah+Tr*Ar)/A1
CLS
REM ************************************************ Air Properties ************************************************
REM The value of 'k' calculated here is the product of the gravitational constant times the coefficient of volumetric expansion divided by the kinematic viscosity squared. (i.e., g βv^2 [1/Km^3]) The equation for 'k' is based on data from Table A-1, p. 388, Kays & Crawford, "Convective Heat and Mass Transfer", second edition, McGraw-Hill.
PRINT
k=2.651E+08-21860001*Ta+7935.4726*Ta^2-13.3076*Ta^3+.0082*T a^4
Gr=k*(Tw-Ta)/1.8*C^3 :REM Grashoff number
Nu=.088*Gr^*(1/3)*((Tw+459.67)/(Ta+459.67))^1.18 :REM Nusselt number
hc=.81*((Tw-Ta)/1.8)^0.426
REM hc=natural convection no lip heat transfer coefficient
REM

REM *************** Print Table Header ***************
PRINT "Nusselt Number =",Nu
PRINT "Grashoff Number =",Gr
PRINT "T mean [°F] =",Th
PRINT ;TAB(3);" Angle";TAB(11);"Sec 1";TAB(19);"Sec 2";TAB(27);"Sec 3";
PRINT;TAB(35);"Sec 4";TAB(43);"Sec 5";TAB(50);"Heat Loss";TAB(61);" h ";
PRINT;TAB(72);"Total Zone 1"
PRINT ;TAB(1);" [degrees]";TAB(11);"[m²]";TAB(19);"[m²]";TAB(27);"[m²]";
PRINT;TAB(35);"[m²]";TAB(43);"[m²]";TAB(51);"[Watts]";TAB(60);"[Watts/K-m²]";
PRINT;TAB(73);"[m²]"
PRINT
REM ************* WRITE HEADER TO CLIPBOARD *************
WRITE#1,"" "Nusselt Number =",Nu
WRITE#1,"" "Grashoff Number =",Gr
WRITE#1,"T mean =",Th
WRITE#1,
WRITE#1,"Angle","Sec 1","Sec 2","Sec 3","Sec 4","Sec 5","Heat Loss","h","Total Zone 1"
WRITE#1,"[degrees]","[m²]","[m²]","[m²]","[m²]","[m²]","[m²]","[Watts]","[Watts/K-m²]","[m²]
WRITE#1,
REM ********************* Beginning of Angle Loop *********************
FOR A=0 TO 90 STEP 15
z=p*A/180
REM **************** Zone Areas ****************
REM The receiver cavity is divided into 5 section to accommodate the
REM zone area calculations. The sections are defined as follows;
REM section 1 = end plate
REM section 2 = frustum
REM section 3 = hot cylinder
REM section 4 = cold cylinder
REM section 5 = ring
REM AZ1 = area of section 1 in zone 1
REM AZ2 = area of section 2 in zone 1
REM AZ3 = area of section 3 in zone 1
REM AZ4 = area of section 4 in zone 1
REM AZ5 = area of section 5 in zone 1
REM ***************************SECTION 1****************************
  IF z>z1 THEN 101
    AZ1=0
  GOTO 201
101:
  IF z>z2 THEN 102
    x=(Ra-(Lf+Lc+Lh)*TAN(z))/Rc
cx=-ATN(x/SQR(-x*x+1))+1.5708
m=(Lf+Lc+Lh)*TAN(z)-Ra+Re
    AZ1=Re^2*cx+((Lf+Lc+Lh)*TAN(z)-Ra)*SQR(2*Re*m-m^2)
  GOTO 201
102:
  AZ1=pi*Re^2
201:
IF z>z1 THEN 202
    x=(Ra-(Lc+Lh)*TAN(z))/Rc
    cx=ATN(x/SQR(-x*x+1))+1.5708
    m=ATN((Rc-Re)/Lf)
    AZ2=((Rc-Re)/Lf)*(Rc-Ra)+Lc+Lh)*SIN(z)/SIN(pi/2-z-m)
    AZ2=(AZ2+(Rc-Ra)/SIN(Re-m))*Rc*cx
    GOTO 301

202 :
IF z>z2 THEN 203
    x1=(Ra-(Lc+Lf+Lh)*TAN(z))/Re
    cx1=ATN(x1/SQR(-x1*x1+1))+1.5708
    x2=(Ra-(Lc+Lh)*TAN(z))/Rc
    cx2=ATN(x2/SQR(-x2*x2+1))+1.5708
    AZ2=SQR(Lf^2+(Rc-Re)^2)*(Re*cxl +Rc*cx2)
    GOTO 301

203 :
IF z>z3 THEN 204
    m=ATN((Ra+Re)/(Lc+Lf+Lh))
    l=SQR((Lc+Lf+Lh)^2+(Ra+Re)^2)*SIN(z-n)/SIN(pi-z-m)
    x=(Ra-(Lc+Lh)*TAN(z))/Rc
    cx=ATN(x/SQR(-x*x+1))+1.5708
    AZ2=(pi*Re+pi*(Re+l*SIN(m)))*l
    AZ2=AZ2+(Rc*cx+pi*(Re+l*SIN(m)))*(SQR(Lf^2+(Rc-Re)^2)-l)
    GOTO 301

204 :
    AZ2=pi*(Re+Rc)*SQR(Lf^2+(Rc-Re)^2)

301 :
REM ............................................ SECTION 3 ............................................

IF z>z3 THEN 302
    x1=(Ra-Lc*TAN(z))/Rc
    x2=(Ra-(Lc+Lh)*TAN(z))/Rc
    cx1=ATN(x1/SQR(-x1*x1+1))+1.5708
    cx2=ATN(x2/SQR(-x2*x2+1))+1.5708
    AZ3=Rc*(cx1+cx2)*Lh
    GOTO 401

302 :
IF z>z4 THEN 303
    m=(Ra+Rc)/TAN(z)
    x=(Ra-Lc*TAN(z))/Rc
    cx=ATN(x/SQR(-x*x+1))+1.5708
    AZ3=2*pi*Rc*(Lh+Lc-m)+Rc*(m-Lc)*(pi+cx)
    GOTO 401

303 :
    AZ3=2*pi*Rc*Lh

401 :
REM ............................................ SECTION 4 ............................................

IF z>z4 THEN 402
    x1=Ra/Rc
    x2=(Ra-Lc*TAN(z))/Rc
    cx1=ATN(x1/SQR(-x1*x1+1))+1.5708
    cx2=ATN(x2/SQR(-x2*x2+1))+1.5708
    AZ4=Rc*(cx1+cx2)*Lc
    GOTO 501

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402:
  x=Ra/Rc
  cx=-ATN(x/SQR(-x*x+1))+1.5708
  m=(Ra+Rc)/TAN(z)
  AZ4=2*pi*Rc*(Lc-m)+Rc*m*(pi+cx)

501:
  REM *************** SECTION 5 *******************************
  zm=pi/2
  IF z<zm THEN 502
  AZ5=pi*(Rc^2-Ra^2)
  GOTO 600

502:
  x=Ra/Rc
  cx=-ATN(x/SQR(-x*x+1))+1.5708
  AZ5=Rc^2*cx-Ra*SQR(Rc^2-Ra^2)

600:
  REM *************** Calculate Area 3 ***********************
  AZT=A1+AZ2+AZ3+AZ4+AZ5
  A3=A1-AZT

REX  *************** Heat Loss Calculation **********************
  zz=pi*30/180
  IF z>zz THEN 601
  n=.63
  GOTO 700

601:
  n=.8

700:
  h=hc*(A1/A2)*(A3/A1)^n
  q=h*A1*(Tw-Ta)/1.8

REM *************** PRINTER Output ***********************
REM
PRINT;TAB(4);
PRINT USING"###.#";A;
PRINT;TAB(9);
PRINT USING "###.####";AZ1;
PRINT;TAB(17);
PRINT USING "###.####";AZ2;
PRINT;TAB(25);
PRINT USING "###.####";AZ3;
PRINT;TAB(33);
PRINT USING "###.####";AZ4;
PRINT;TAB(41);
PRINT USING "###.####";AZ5;
PRINT;TAB(50);
PRINT USING "###.####";q;
PRINT;TAB(60);
PRINT USING "###.####";h;
PRINT;TAB(70);
PRINT USING "###.####";AZT
REM *************** OUTPUT TO CLIPBOARD ***********************
WRITE #1,A,AZ1,AZ2,AZ3,AZ4,AZ5,q,h,AZT
NEXT A
NEXT Th
NEXT rad
PRINT "bye!"
CLOSE #1
END
## Appendix 11 - Siebers and Kraabel Model Heat Loss

<table>
<thead>
<tr>
<th>Angle (degrees)</th>
<th>Sec 1</th>
<th>Sec 2</th>
<th>Sec 3</th>
<th>Sec 4</th>
<th>Sec 5</th>
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<th>Total Zone 1 (kW)</th>
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### T mean = 100

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### Appendix 11: Siebers and Kraabel Model Heat Loss

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<th>Angle (Degrees)</th>
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<th>Case 5</th>
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<th>k</th>
<th>Total Zone 1 (kW/m²)</th>
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## Appendix 11 Siebers and Kraabel Model Heat Loss

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**Total Heat Loss (Watts) = 11494.4**

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**Total Heat Loss (Watts) = 11494.4**

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**Total Heat Loss (Watts) = 11494.4**
### Appendix 11 Siebers and Kraabel Model Heat Loss

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<th>Angle (Degrees)</th>
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**Siebers and Kraabel Model Heat Loss**

- **Nusselt Number:** 1.14650
- **Grashof Number:** 1.14650
- **1 mean = 400**

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**Siebers and Kraabel Model Heat Loss**

- **Nusselt Number:** 204.54
- **Grashof Number:** 9.05504
- **1 mean = 500**

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**Siebers and Kraabel Model Heat Loss**

- **Nusselt Number:** 726.17
- **Grashof Number:** 1.185510
- **1 mean = 600**
APPENDIX 12: Stine and McDonald Model Computer Program Listing

REM
REM **************************** Stine & McDonald Method
REM
REM ****************************
REM
PRINT "Stine & McDonald Method of Predicting Convective Losses"
PRINT
PRINT
REM **************************** Receiver Geometry ****************************
REM
Re=.127
Rc=.33
Lf=.292
Lh=.254
Lc=.14
L=2*Rc
REM
************************* Constants *************************
pi=4*ATN(1)
Ta=70 :Tak=(Ta+459.67)/1.8
REM ambient temperature [°F]
REM
OPEN "CLIP:" FOR OUTPUT AS #1
REM
WRITE#1, "Stine & McDonald Method"
WRITE#1,"End Plate Radius [m] = ",Re
WRITE#1,"Cavity Radius [m] = ",Rc
WRITE#1,"Frustum Length [m] = ",Lf
WRITE#1,"Hot Cylindrical Section Length [m] = ",Lh
WRITE#1,"Cold Cylindrical Section Length [m] = ",Lc
WRITE#1,"T amb [°F] = ",Ta
REM
************************ Receiver Aperture Radius Loop ****************************
REM
FOR rad=1 TO 4
READRa
DATA .0762, .1524, .2286, .329
Da=2*Ra
REM
************************ Area Constants ****************************
REM
Ah=pi*(Re+Rc)*(Lf^2+(Rc-Re)^2)^.5+2*pi*Rc*Lh
Ar=pi*Re^2+pi*(Rc^2-Ra^2)+2*pi*Rc*Lc
At=Ah+Ar
REM
************************ Print Header ****************************
REM
PRINT "Stine & McDonald Method"
PRINT
PRINT "End Plate Radius [m] = ";Re
PRINT "Cavity Radius [m] = ";Rc
PRINT "Aperture Radius [m] = ";Ra
PRINT "Frustum Length [m] = ";Lf
PRINT "Hot Cylindrical Section Length [m] = ";Lh
PRINT "Cold Cylindrical Section Length [m] = ";Lc
PRINT "T amb [°F] = ";Ta; " [k] = ";Tak
PRINT " Total Area [m^2] = ";At

PRINT
REM                     write header to clipboard
WRITE#1,                   
WRITE#1,"","Aperture Radius [m] = ",Ra
WRITE#1,""," Total Area [m^2] = ",At
REM                     Operating Temperature Loop
REM
FOR Th=300 TO 600 STEP 100
REM mean system operating temperature of receiver [°F]
Tr=Th-100
Tw=(Th*Ah+Tr*Ar)/At
CLS
REM
REM The value of 'k' calculated here is the product of the gravitational
REM constant times the coefficient of volumetric expansion divided by
REM the kinematic viscosity squared. (i.e., g β/v^2 [1/K-m^3]) The
REM equation for 'k' is based on data from Table A-1, p. 388, Kays & Crawford,
PRINT
gbv=1.1547E+19*Tak^-4.4187
Gr=gbv*(Th-Ta)/1.8*L^3 :REM Grashoff number
k=.0071749261015#+.000064030639041#*Tak :REM thermal conductivity of air
REM
REM                     Print Table Header
PRINT "Grashoff Number = ";Gr
PRINT "T mean [°F] = ";Th
PRINT ;TAB(3);" Angle";TAB(11);"Nu";TAB(19);"Heat Loss";TAB(27);" h "
PRINT ;TAB(1);" [degrees]";TAB(11);";TAB(19);"[Watts]";TAB(27);"[Watts/K-m^2]".
PRINT
REM                     Write header to clipboard
WRITE#1,","Grashoff Number = ",Gr
WRITE#1,"T mean = ",Th
WRITE#1,                   
WRITE#1,"Angle","Nu","Heat Loss","h"
WRITE#1,"[degrees]","","[Watts]","[Watts/K-m^2]"
REM                     Beginning of Angle Loop
REM
FOR a=-90 TO 90 STEP 15
 z=pi*a/180
REM                     Heat Loss Calculation
REM
s=-.982*(Da/L)+1.12
IF a=90 THEN 200
Nu=.088*Gr^(1/3)*((Tw+459.67)/(Ta+459.67))^(.18*(COS(z))^2.47 *(Da/L)^s
GOTO300

154
200:
Nu=0:
300:
:REM Nusselt number
h=Nu*k/L
q=h*At*(Tw-Ta)/1.8
REM
PRINT *************** PRINTER Output ***************
REM
PRINT;TAB(4);
PRINT USING "###.*";a;
PRINT;TAB(9);
PRINT USING "###.*";Nu;
PRINT;TAB(17);
PRINT USING "###.*";q;
PRINT;TAB(25);
PRINT USING "###.*";h
REM *************** OUTPUT TO CLIPBOARD ***************
WRITE #1,a,Nu,q,h
NEXT a
NEXT Th
NEXT rad
PRINT "bye!"
CLOSE #1
END
Appendix 13: Stein and McDonald Model Heat Loss

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T mean = 600

156
## Stein and McDonald Model Heat Loss

### Aperture Radius [m] = 0.1524

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| Grashof Number = 9.79E+9 |

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157
### Appendix 13: Stein and McDonald Model Heat Loss

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# Appendix 13: Stein and McDonald Model Heat Loss

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Grashof Number = $1.21 \times 10^{10}$
Appendix 14: Shape Factors Formulas

All formulas are developed from the basic disc-to-disc shape factor formula (20). The N by N coefficient matrix of the heat loss equation requires N² shape factor equations (Eqn. 1). The shape factor equations are solved using a digital computer. The following section shows the development of the shape factor formulas used in the coefficient matrix.

(a) End Plate (section 1 ↔ aperture):

\[ L = l_c + l_m + l_a \]

let \( R_e = \frac{r_e}{L} \), \( R_a = \frac{r_a}{L} \), and \( X = 1 + \frac{1 + R_e^2}{R_a^2} \)

\[ F_{A_e - A_a} = \frac{A_a}{A_e} F_{A_a - A_e} \]

\[ F_{A_e - A_a} = \frac{A_a}{2A_e} \left[ X - \sqrt{X^2 - 4 \left( \frac{R_e}{R_a} \right)^2} \right] \]

\[ F_{A_e - A_a} = \frac{\frac{r_a^2}{2r_e^2} \left[ 1 + \frac{L^2 + r_e^2}{r_a^2} - \sqrt{\left(1 + \frac{L^2 + r_e^2}{r_a^2} \right)^2 - 4 \left( \frac{r_e}{r_a} \right)^2} \right]} \]

\[ F_{A_e - A_a} = \frac{1}{2r_e^2} \left\{ \left( \frac{1}{l_c} + \frac{1}{l_m} + \frac{1}{l_a} \right)^2 + \frac{r_a^2 + r_e^2}{r_a^2} - \sqrt{\left[ \left( \frac{1}{l_c} + \frac{1}{l_m} + \frac{1}{l_a} \right)^2 + \frac{r_a^2 + r_e^2}{r_a^2} \right]^2 - 4 \left( \frac{r_e}{r_a} \right)^2} \right\} \]
(b) Center Cylinder (section 3 ↔ aperture):

\[ y_2 = l_a + n_m l_b \]

\[ y_1 = l_a + (n_m - 1) l_b \]

\[ F_{A_a - A_{section}} = F_{A_a - x-section_1} - F_{A_a - x-section_2} \]

\[ A_a F_{A_a - A_{section}} = A_{section} F_{A_{section} - A_a} \]

\[ F_{A_{section} - A_a} = \frac{A_a}{A_{section}} F_{A_{section} - A_a} \]

\[ F_{A_{section} - A_a} = \frac{A_a}{A_{section}} \left[ F_{A_a - x-section_1} - F_{A_a - x-section_2} \right] \]

\[ F_{A_{section} - A_a} = \frac{A_a}{A_{section}} \left[ \frac{A_c}{A_a} F_{x-section_1} - A_a - \frac{A_c}{A_a} F_{x-section_2} - A_a \right] \]

\[ F_{A_{section} - A_a} = \frac{A_c}{A_{section}} \left[ F_{x-section_1} - A_a - F_{x-section_2} - A_a \right] \]

\[ F_{A_{section} - A_a} = \frac{r_c}{2l_b} \left[ F_{x-section_1} - A_a - F_{x-section_2} - A_a \right] \]

let \[ R_1 = \frac{r_c}{y}, \quad R_2 = \frac{r_a}{y}, \quad X = 1 + \frac{1 + R_2^2}{R_1^2} \]

then \[ F_{A_{x-section} - A_a} = \frac{1}{2} \left[ X - \sqrt{X^2 - 4 \left( \frac{R_2}{R_1} \right)^2} \right] \]

\[ F_{A_{x-section} - A_a} = \frac{1}{2} \left[ \frac{1 + 1 + R_2^2}{R_1^2} - \sqrt{\left( 1 + \frac{1 + R_2^2}{R_1^2} \right)^2 - 4 \left( \frac{R_2}{R_1} \right)^2} \right] \]

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\[ F_{A_{x\text{-section}_2}} - A_a = \frac{1}{2} \left[ 1 + \frac{1 + \frac{r_a^2}{r_c^2}}{\frac{y_2}{y_2^2}} \sqrt{\left( 1 + \frac{1 + \frac{r_a^2}{r_c^2}}{\frac{y_2}{y_2^2}} \right)^2 - 4 \left( \frac{r_a}{r_c} \right)^2} \right] \]

\[ F_{A_{x\text{-section}_1}} - A_a = \frac{1}{2} \left[ 1 + \frac{y_2^2 + r_a^2}{r_c^2} \right] \sqrt{\left( 1 + \frac{y_2^2 + r_a^2}{r_c^2} \right)^2 - 4 \left( \frac{r_a}{r_c} \right)^2} \]

and similarly:

\[ F_{A_{\text{section}}} - A_a = \frac{r_c}{4r} \left[ \frac{y_1^2 + r_a^2}{r_c^2} \right] \sqrt{\left( 1 + \frac{y_1^2 + r_a^2}{r_c^2} \right)^2 - 4 \left( \frac{r_a}{r_c} \right)^2} \]

\[ + \sqrt{\left( 1 + \frac{y_2^2 + r_a^2}{r_c^2} \right)^2 - 4 \left( \frac{r_a}{r_c} \right)^2} \]

\[ F_{A_{\text{section}} - A_a} = \frac{1}{4r_c} \left[ y_1^2 - y_2^2 - \sqrt{\left( r_1^2 + y_1^2 + r_2^2 \right)^2 - 4 \left( r_{\alpha_a} \right)^2} + \sqrt{\left( r_1^2 + y_2^2 + r_2^2 \right)^2 - 4 \left( r_{\alpha_a} \right)^2} \right] \]

\[ F_{A_{\text{section}} - A_a} = \frac{1}{4r_c} \left[ (l_a + (n_m - 1)b)^2 - (l_a + n_mb)^2 \right] \sqrt{\left( (l_a + (n_m - 1)b)^2 + r_2^2 + r_2^2 \right)^2 - 4 \left( r_{\alpha_a} \right)^2} \]

\[ + \sqrt{(l_a + n_mb)^2 + r_2^2 + r_2^2} - 4 \left( r_{\alpha_a} \right)^2 \]

(c) Frustum \( \text{(section 2 \leftrightarrow aperture):} \)

The view factor for a section of the frustum from \( A_a \) is equal to the view factor from \( A_a \) to the circular cross-sectional area on the bottom of the frustum section minus the view factor from \( A_a \) to the circular cross-section on the top of the frustum section.
\[ F_{A_a} - A_{section} = F_{A_a} - x-section_1 - F_{A_a} - x-section_2 \]

\[ A_{a} F_{A_a} - A_{section} = A_{section} F_{A_{section}} - A_{a} \]

\[ F_{A_{section}} - A_{a} = \frac{A_{a}}{A_{section}} F_{A_a} - A_{section} \]

\[ F_{A_{section}} - A_{a} = \frac{A_{a}}{A_{c}} [ F_{A_a} - x-section_1 - F_{A_a} - x-section_2 ] \]

\[ F_{A_{section}} - A_{a} = \frac{A_{a}}{A_{section}} \left[ \frac{A_{x-section_1}}{A_a} F_{x-section_1} - A_{a} - \frac{A_{x-section_2}}{A_a} F_{x-section_2} - A_{a} \right] \]

\[ F_{A_{section}} - A_{a} = \frac{1}{A_{section}} \left[ A_{x-section_1} F_{x-section_1} - A_{a} - A_{x-section_2} F_{x-section_2} - A_{a} \right] \]

\[ F_{A_{section}} - A_{a} = \frac{1}{(r_1 + r_2)_b} \left[ r_1^2 F_{x-section_1} - A_{a} - r_2^2 F_{x-section_2} - A_{a} \right] \]

\[ \text{let } \quad R_1 = \frac{r_{x-section}}{y}, \quad R_2 = \frac{r_a}{y}, \quad X = 1 + \frac{1 + R_2^2}{R_1^2} \]

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then \( F_{Ax-section} \cdot A_a = \frac{1}{2} \left[ X - \sqrt{X^2 - 4 \left( \frac{R_2}{R_1} \right)^2} \right] \)

\[
F_{Ax-section} \cdot A_a = \frac{1}{2} \left[ 1 + \frac{1 + R_2^2}{R_1^2} - \sqrt{\left( 1 + \frac{1 + R_2^2}{R_1^2} \right)^2 - 4 \left( \frac{R_2}{R_1} \right)^2} \right]
\]

\[
F_{Ax-section} \cdot A_a = \frac{1}{2} \left[ 1 + \frac{h_2 + r_2^2}{h_2^2} - \sqrt{\left( 1 + \frac{h_2 + r_2^2}{h_2^2} \right)^2 - 4 \left( \frac{r_2}{h_2} \right)^2} \right]
\]

and similarly:

\[
F_{Ax-section_1} \cdot A_a = \frac{1}{2} \left[ 1 + \frac{h_1^2 + r_1^2}{r_1^2} - \sqrt{\left( 1 + \frac{h_1^2 + r_1^2}{r_1^2} \right)^2 - 4 \left( \frac{r_1}{r_1} \right)^2} \right]
\]

\[
F_{Ax-section} \cdot A_a = \frac{1}{2l_b(r_1 + r_2)} \left[ \frac{r_1^2 - r_2^2 + h_1^2 - h_2^2 - \sqrt{(r_1^2 + h_1^2 + r_2^2)^2 - 4 \left( r_1r_2 \right)^2}}{r_1} + \sqrt{\left( r_2^2 + h_2^2 + r_2^2 \right)^2 - 4 \left( r_2r_2 \right)^2} \right]
\]

\[
h_1 = l_a + l_m + \left( n_m - 1 \right) \frac{l_c}{N_c} \quad \text{h}_2 = l_a + l_m + n_m \frac{l_c}{N_c}
\]

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\[ r_2 = r_c - n_m \frac{r_e - r_c}{N_c} \quad r_1 = r_c - (n_m - 1) \frac{r_e - r_c}{N_c} \]

\[ F_{nm, A_n} = \frac{1}{2l_b} \left[ \frac{2r_c - (2n_m - 1)(r_e - r_c)}{N_c} \right] \left[ \left[ r_c - \left( n_m - 1 \right) \frac{r_e - r_c}{N_c} \right]^2 - \left[ r_c - n_m \left( \frac{r_e - r_c}{N_c} \right) \right]^2 \right. \]
\[ + \left[ l_a + l_m + (n_m - 1) \frac{r_e}{N_c} \right]^2 - \left[ l_a + l_m + n_m \frac{r_e}{N_c} \right]^2 \]
\]

\[ - \sqrt{\left\{ \left[ r_c - \left( n_m - 1 \right) \frac{r_e - r_c}{N_c} \right]^2 + \left( l_a + l_m + (n_m - 1) \frac{r_e}{N_c} \right)^2 + r_a^2 \right\}^2 - 4 \left\{ \left[ r_c - \left( n_m - 1 \right) \frac{r_e - r_c}{N_c} \right] r_a \right\}^2} \]
\[ + \sqrt{\left\{ \left[ r_c - n_m \frac{r_e - r_c}{N_c} \right]^2 + \left( l_a + l_m + n_m \frac{r_e}{N_c} \right)^2 + r_a^2 \right\}^2 - 4 \left\{ \left[ r_c - n_m \frac{r_e - r_c}{N_c} \right] r_a \right\}^2} \]

(d) Spacer Ring (section 4 ↔ aperture):

\[ F_{A_n} - A_s = 1 - F_{A_n} - A_{x-section} \]

\[ F_{A_s - A_n} = \frac{A_n}{A_s} \cdot F_{A_n - A_s} \]

\[ F_{A_n - A_{x-section}} = \frac{A_{x-section}}{A_a} \cdot F_{A_{x-section} - A_a} \]

\[ F_{A_a} - A_a = \frac{A_a}{A_a} \left[ 1 - \frac{A_{x-section}}{A_a} \cdot F_{A_{x-section} - A_a} \right] \]

\[ F_{A_{x-section} - A_a} = \frac{1}{2} \left[ X - \sqrt{X^2 - 4 \left( \frac{R_a}{R_c} \right)^2} \right] \]

\[ F_{A_a - A_a} = \frac{r_a^2}{2r_{c1}} \left[ 1 - \frac{r_a^2}{2r_a^2} \left[ X - \sqrt{X^2 - 4 \left( \frac{R_a}{R_c} \right)^2} \right] \right] \]
where \( R_a = \frac{r_a}{l_a} \), \( R_c = \frac{r_c}{l_a} \)

\[
X = 1 + \frac{\left(\frac{r_a}{l_a}\right)^2}{\left(\frac{r_c}{l_a}\right)^2}
\]

\[
X = 1 + \frac{l_a^2 + r_a^2}{r_c^2}
\]

therefore:

\[
F_{A_{x\text{-section}} - A_a} = \frac{1}{2} \left[ 1 + \frac{l_a^2 + r_a^2}{r_c^2} - \sqrt{\left(1 + \frac{l_a^2 + r_a^2}{r_c^2}\right)^2 - 4\left(\frac{r_a}{l_a}\right)^2} \right]
\]

\[
F_{A_t - A_a} = \frac{r_a^2}{2r_c l_a} \left[ 1 - \frac{r_a^2}{2r_c^2} \left[ 1 + \frac{l_a^2 + r_a^2}{r_c^2} - \sqrt{\left(1 + \frac{l_a^2 + r_a^2}{r_c^2}\right)^2 - 4\left(\frac{r_a}{l_a}\right)^2} \right] \right]
\]

\[
F_{A_s - A_a} = \frac{r_a^2}{2r_c l_a} \left[ 1 - \frac{1}{2r_a^2} [(r_c^2 + l_a^2 + r_a^2) - \sqrt{(r_c^2 + l_a^2 + r_a^2)^2 - 4(r_c^2 - \frac{r_a^2}{l_a})^2}] \right]
\]

(e) (section i and j \( \leftrightarrow \) section i and j): Shape Factors from a Surface Onto Itself

For the flat surfaces (aperture, annulus, and end plate) \( F_{i\cdot i} = 0 \)

For the cylindrical and frustum sections:

\[
F_{i\cdot i} = 1 - F_{i\cdot x1} - F_{i\cdot x2}
\]

\[
F_{i\cdot x1} = \frac{A_{x1}}{A_i} F_{x1\cdot i} \quad \text{and} \quad F_{i\cdot x2} = \frac{A_{x2}}{A_i} F_{x2\cdot i}
\]

therefore

\[
F_{i\cdot x1} = \frac{A_{x1}}{A_i} F_{x1\cdot i} \quad \text{and} \quad F_{i\cdot x2} = \frac{A_{x2}}{A_i} F_{x2\cdot i}
\]
and

\[ F_{x1-i} = 1 - F_{x1-x2} \quad \text{and} \quad F_{x2-i} = 1 - F_{x2-x1} \]

\[ F_{x2-x1} = \frac{A_{x1}}{A_{x2}} F_{x1-x2} \]

\[ F_{x1-x2} = \frac{A_{x1}}{A_{x2}} \left( 1 - F_{x1-x2} \right) - 2 \frac{A_{x1}}{A_{x2}} F_{x1-x2} \]

\[ F_{x1-x2} = \frac{1}{2} \left[ X - \sqrt{X^2 - 4 \left( \frac{R_2}{R_2} \right)^2} \right] \]

where \( R_1 = \frac{r_1}{h}, \ R_2 = \frac{r_2}{h}, \ \text{and} \ X = 1 + \frac{h^2 + r_2^2}{r_1^2} \)

\[ F_{x1-x2} = \frac{1}{2} \left[ 1 + \frac{h^2 + r_2^2}{r_1^2} - \sqrt{\left[ 1 + \frac{h^2 + r_2^2}{r_1^2} \right]^2 - 4 \left( \frac{r_2}{r_1} \right)^2} \right] \]

\[ F_{x1-x2} = \frac{A_{x1}}{A_{x1}} \frac{A_{x2}}{A_{x1}} \left[ 1 + \frac{h^2 + r_2^2}{r_1^2} - \sqrt{\left[ 1 + \frac{h^2 + r_2^2}{r_1^2} \right]^2 - 4 \left( \frac{r_2}{r_1} \right)^2} \right] \]

\[ F_{x1-x2} = \frac{r_2^2}{(r_1 + r_2) \lambda_b} + \frac{r_1^2}{(r_1 + r_2) \lambda_b} \left[ \frac{h^2 + r_2^2}{r_1^2} - \sqrt{\left[ 1 + \frac{h^2 + r_2^2}{r_1^2} \right]^2 - 4 \left( \frac{r_2}{r_1} \right)^2} \right] \]

For the spacer ring (section 4 \( \leftrightarrow \) section 4):

\( r_1 = r_2 = r_c \quad \text{and} \quad h = l_a \)

For the hot cylinder (section 3 \( \text{band} \ i \leftrightarrow \text{section} \ 3 \( \text{band} \ i \)):

\( r_1 = r_2 = r_c \quad \text{and} \quad h = l_b \)

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For the frustum (section $2_{\text{band } i \leftrightarrow \text{section } 2_{\text{band } j}}$):

$$r_1 = r_c - (r_c - r_e) \frac{n_c - 1}{N_c}, \quad r_2 = r_c - (r_c - r_e) \frac{n_c}{N_c}, \quad \text{and} \quad h = \frac{l_{j_b}}{\sqrt{1 + (r_c - r_e)^2}}$$

**Shape Factors from Bands of the Cylindrical and Frustum Sections to other Bands of the Cylindrical and Frustum Sections (section $i_{\text{band } j \leftrightarrow \text{section } j_{\text{band } k}}$):**

The following shape factor formulas are used between different bands of the frustum section, between different bands of the hot cylindrical section, between bands of the hot cylindrical section and bands of the frustum section, between the spacer ring and bands of the hot cylindrical section, and between the spacer ring and bands of the frustum section.

$$F_{i \cdot j} = F_{i \cdot x2} - F_{i \cdot x1}$$

$$F_{x1 \cdot i} = F_{x1 \cdot x3} - F_{x1 \cdot x4}$$

$$F_{x2 \cdot i} = F_{x2 \cdot x3} - F_{x2 \cdot x4}$$

$$F_{i \cdot x1} = \frac{A_{x1}}{A_i} F_{x1 \cdot i} \quad \text{and} \quad F_{i \cdot x2} = \frac{A_{x2}}{A_i} F_{x2 \cdot i}$$
\[ F_{i-j} = \frac{Ax_2}{A_i} \left[ F_{x2 - x3} - F_{x2 - x4} \right] - \frac{Ax_1}{A_i} \left[ F_{x1 - x3} - F_{x1 - x4} \right] \]

\[ F_{i-j} = \frac{r_2^2}{(r_3 + r_4)l_b} \left[ F_{x2 - x3} - F_{x2 - x4} \right] - \frac{r_1^2}{(r_3 + r_4)l_b} \left[ F_{x1 - x3} - F_{x1 - x4} \right] \]

\[ F_{x_n-x_m} = \frac{1}{2} \left[ 1 + \frac{h_{nm}^2 + r_m^2}{r_n^2} - \sqrt{\left[ 1 + \frac{h_{nm}^2 + r_m^2}{r_n^2} \right] - 4 \left[ \frac{r_m}{r_n} \right]^2} \right] \]

between different bands of the frustum section (section 2\_band i ↔ section 2\_band j):

\[ r_1 = r_c - (r_c - r_e) \frac{n_j}{N_c} \]

\[ r_2 = r_c - (r_c - r_e) \frac{(n_j - 1)}{N_c} \]

\[ r_3 = r_c - (r_c - r_e) \frac{n_i}{N_c} \]

\[ r_4 = r_c - (r_c - r_e) \frac{(n_i - 1)}{N_c} \]

\[ h = \frac{l_{db}}{\sqrt{l_e^2 + (r_c - r_e)^2}} \]

\[ h_{13} = h_{24} = (n_j - n_i)h \]

\[ h_{14} = (n_j - n_i + 1)h \]

\[ h_{23} = (n_j - n_i - 1)h \]

between different bands of the hot cylindrical section (section 3\_band i ↔ section 3\_band j):

\[ r_1 = r_2 = r_3 = r_4 = r_c \]

\[ h_{13} = h_{24} = (n_j - n_i)l_b \]

\[ h_{14} = (n_j - n_i + 1)l_b \]

\[ h_{23} = (n_j - n_i - 1)l_b \]

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between bands of the hot cylindrical section and bands of the frustum section  
(section 2_{band i} \leftrightarrow section 3_{band j}):  

\[ r_1 = r_c - (r_c - r_J) \frac{n_j}{N_c} \]

\[ r_2 = r_c - (r_c - r_J) \frac{(n_j - 1)}{N_c} \]

\[ r_3 = r_4 = r_c \]

\[ h = \frac{l_1 b}{\sqrt{l_c^2 + (r_c - r_J)^2}} \]

\[ h_{13} = h_{24} = l_m - n_j l_b + n_j h \]

\[ h_{14} = l_m - (n_j - 1) l_b + n_j h \]

between the spacer ring and bands of the hot cylindrical section  
(section 4 \leftrightarrow section 3_{band i}):  

\[ F_{\text{spacer} - j} = \frac{r_2^2}{(r_3 + r_4)^2} \left[ F_{x2 - x3} - F_{x21 - x4} \right] - \frac{r_1^2}{(r_3 + r_4)^2} \left[ F_{x1 - x3} - F_{x1 - x4} \right] \]

\[ F_{x n - x m} = \frac{1}{2} \left[ \frac{h_{nm}^2 + r_m^2}{r_n^2} \right]^2 \sqrt{1 + \frac{h_{nm}^2 + r_m^2}{r_n^2}} - 4 \left( \frac{r_m}{r_n} \right)^2 \]

\[ r_1 = r_2 = r_3 = r_4 = r_c \]

\[ h_{13} = n_j l_b \]

\[ h_{14} = n_j l_b + l_s \]

\[ h_{23} = (n_j - 1) l_b \]

\[ h_{24} = (n_j - 1) l_b + l_s \]

between the spacer ring and bands of the frustum section  
(section 4 \leftrightarrow section 2_{band i}):
The following shape factor formulas are used between the end plate and bands of the hot cylindrical section, bands of the frustum section, and the spacer section, between the aperture and the bands of the hot cylindrical section, bands of the frustum section, and the spacer section, and between the annulus and aperture combined and bands of the hot cylindrical section, bands of the frustum section, and the spacer section.

\[ r_1 = r_c - (r_e - r_o) \frac{n_j}{N_c} \]
\[ r_2 = r_c - (r_e - r_o) \frac{(n_j - 1)}{N_c} \]
\[ r_3 = r_4 = r_c \]

\[ h = \frac{l_i l_b}{\sqrt{l_c^2 + (r_c - r_e)^2}} \]
\[ h_{13} = l_m + n_j h \]
\[ h_{14} = l_m + n_j h + l_a \]
\[ h_{23} = l_m + (n_j - 1) h \]
\[ h_{24} = l_m + (n_j - 1) h + l_a \]

Shape Factors from Circular Sections to Bands of the Cylindrical and Frustum Sections:
between the end plate and bands of the hot cylindrical section
(Section 1 ↔ Section 3 and i):

\[ r_1 = r_2 = r_c \]
\[ r_3 = r_e \]
\[ h_{23} = l_c + (N_m - n_j) l_b \]
\[ h_{13} = l_c + (N_m - n_j + 1) l_b \]

between the end plate and bands of the frustum section
(Section 1 ↔ Section 2 and i):

\[ r_1 = r_e + (r_c - r_e) \left[ 1 - \frac{n_j}{N_c} \right] \]
\[ r_2 = r_e + (r_c - r_e) \left[ 1 - \frac{n_j}{N_c} \right] \]
\[ r_3 = r_e \]
\[ h = \frac{l_{\ell_b}}{\sqrt{l_c^2 + (r_c - r_\ell)^2}} \]

\[ h_{23} = (N_c - n_j) h \]

\[ h_{13} = (N_c - n_j + 1) h \]

between the end plate and the spacer section (section 1 ↔ section 4):

\[ r_1 = r_2 = r_3 = r_c \]

\[ r_3 = r_\ell \]

\[ h_{23} = l_c + l_m \]

\[ h_{13} = l_c + l_m + l_a \]

between the annulus and aperture combined and bands of the hot cylindrical section (section 5+ aperture ↔ section 3 band j):

\[ r_1 = r_2 = r_3 = r_c \]

\[ h_{23} = l_a + (n_j - 1) l_b \]

\[ h_{13} = l_a + n_j l_b \]

between the annulus and aperture combined and bands of the frustum section (section 5+ aperture ↔ section 2 band j):

\[ r_1 = r_c - n_j \frac{(r_c - r_\ell)}{N_c} \]

\[ r_2 = r_c - (n_j - 1) \frac{(r_c - r_\ell)}{N_c} \]

\[ r_3 = r_c \]

\[ h = \frac{l_{\ell_b}}{\sqrt{l_c^2 + (r_c - r_\ell)^2}} \]

\[ h_{23} = l_a + l_m + (n_j - 1) h \]

\[ h_{13} = l_a + l_m + n_j h \]

between the annulus and aperture combined and the spacer section (section 5+ aperture ↔ section 4):

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between the aperture and bands of the hot cylindrical section
(aperture ↔ section 3 bands i):

\[ r_1 = r_2 = r_3 = r_c \]
\[ h_{23} = 0 \]
\[ h_{13} = l_a \]

between the aperture and bands of the frustum section
(aperture ↔ section 2 bands i):

\[ r_1 = r_c - \frac{n_j (r_c - r_d)}{N_c} \]
\[ r_2 = r_c - (n_j - 1) \frac{(r_c - r_d)}{N_c} \]
\[ r_3 = r_a \]
\[ h = \frac{l_d l_b}{\sqrt{l_c^2 + (r_c - r_d)^2}} \]
\[ h_{23} = l_a + l_m + (n_j - 1) h \]
\[ h_{13} = l_a + l_m + n_j h \]

between the aperture and the spacer section
(aperture ↔ section 4):
The following shape factor formulas are used between the end plate and the aperture: and between the end plate and the aperture and annulus combined.

\[
F_{i,j} = \frac{1}{2} \left[ 1 + \frac{h^2 + r_j^2}{r_i^2} \right] \sqrt{1 + \frac{h^2 + r_j^2}{r_i^2} - 4 \left( \frac{r_j}{r_i} \right)^2}
\]

between the end plate and the aperture
(\text{section } 1 \leftrightarrow \text{ aperture}):

\[
\begin{align*}
  r_i &= r_e \\
  r_j &= r_a \\
  h &= l_a + l_m + l_c
\end{align*}
\]

between the end plate and the aperture and annulus combined
(\text{section } 1 \leftrightarrow \text{ aperture + section } 5):

\[
\begin{align*}
  r_i &= r_e \\
  r_j &= r_a \\
  h &= l_a + l_m + l_c
\end{align*}
\]
\[ r_i = r_e \]
\[ r_j = r_c \]
\[ h = l_a + l_m + l_c \]

For shape factors from the spacer section, bands of the hot cylindrical section, and bands of the frustum section to the annulus section the following relationship is used:

\[ F_i - \text{annulus} = F_i - (\text{annulus} + \text{aperture}) - F_i - \text{aperture} \]
Appendix 15: Analytical Thermal Radiation Heat Loss Program Listing

REM
REM
REM
REM THEORETICAL THERMAL RADIATION HEAT LOSS PROGRAM
REM
REM This program predicts the thermal radiative heat loss from
REM the cavity solar receiver using the net radiation method.
REM
REM DIFFUSE GRAY BODY VERSION
REM This section of the program is used to verify the thermal radiation
REM shape factor formulas of the solar cavity receiver.
REM
REM ****************************************************
REM nomenclature
REM *************
REM re = end plate radius
REM rc = cavity radius
REM ra = aperture radius
REM lc = length of frustum section
REM lm = length of hot cylindrical section
REM la = length of cold cylindrical section
REM lbc = width of hot isothermal bands in frustum section
REM lbm = width of hot isothermal bands in hot cylindrical section
REM Nc = number of bands in frustum section
REM Nm = number of bands in hot cylindrical section
REM
REM constants
REM *************
S=5.729*10^-8 ;REM Stephan-Boltzmann constant W/(m^2 K^4)
pi=3.14
REM ****************************************************
REM CAVITY GEOMETRY
REM ****************************************************
re=.254/2
rc=.33
lc=.292
la=.14
lm=.686-lc-la
Nm=15!
lbm=lm/Nm
Nc=23!
lbc=SQR((rc-re)^2+lc^2)/Nc
hc=lc*lbc/SQR((rc-re)^2+lc^2)
DIM rad(12),Top(12)
FOR n=1 TO 12
READ rad(n)
DATA 0.2286,0.2286,0.2286,0.2286,0.2286,0.2286,0.0762,0.0762,0.1524,0.1524,0.2286,2
FOR n=1 TO 12
READ Top(n)
DATA 300,400,500,600,400,600,400,600,400,600,400,600
NEXT n
Tdiff=20 :REM assumed temperature difference between inlet and outlet
REM This section determines the total number of elements that make
REM up the internal surface of the cavity receiver.
REM
NT=Nc+Nm+4
DIM F(NT,NT),sum(NT),A(NT,NT),E(NT),T(NT),C(NT),q(NT),G(NT,NT+1)
,M(NT+1)
REM F(I,J) is the shape factor matrix
REM sum(NT) is a shape factor verification array
REM A(NT,NT) is the coefficient matrix
REM E(NT) is the emissivity array
REM T(NT) is the temperature array
REM C(NT) is the constant array
REM G(NT,NT+1) is the augmented Gaussian matrix
REM q(NT) is the outgoing radiant energy flux (radiosity)
REM
REM This section defines the numbering of elements that make
REM up the internal surface of the cavity receiver.
REM
aperture = 1
annulus = 2
spacer ring = 3
end plate = 4
hot cylindrical section is numbered 5 thru Nm+4
frustum section is numbered Nm+5 thru Nc + 6
REM
REM EMISSIVITY ARRAY INPUT SECTION
REM
E(1)=0 :REM emissivity of the aperture
E(2)=.7 :REM emissivity of the annulus
E(3)=.7 :REM emissivity of the spacer section
E(4)=.7 :REM emissivity of the end plate
FOR n=5 TO NT
E(n)=.85
NEXT n
REM
REM BEGINNING OF APERTURE RADIUS VARIATION LOOP
REM
numT=4
FOR samp=1 TO 12
ra=rad(samp)
REM SHAPE FACTORS CALCULATION SECTION
REM
REM Shape factors for each element onto itself *****
REM
REM For flat surfaces Fi-i = 0 ************
\[
F(1,1) = 0 \\
F(2,2) = 0 \\
F(4,4) = 0 \\
\text{REM aperture, annulus, and end plate}
\]

\[
\text{************ spacer section ************}
\]

\[
ri = rc \\
rj = rc \\
h = la
\]

GOSUB shape

\[
F(3,3) = 1 - \frac{ri^2}{(ri + rj)/(1-F)} - \frac{rj^2}{(ri + rj)/(1-F)}
\]

\[
\text{REM hot cylindrical section ******************}
\]

\[
ri = rc \\
rj = rc \\
h = lbm
\]

GOSUB shape

FOR n = 1 TO Nm

\[
k = n + 4 \\
F(k,k) = 1 - \frac{ri^2}{(ri + rj)/(lbm)} - \frac{rj^2}{(ri + rj)/(lbm)}
\]

NEXT n

\[
\text{REM frustum section ******************}
\]

\[
h = hc \\
\text{FOR n = 1 TO Nc}
\]

\[
ri = rc - (rc-re)*n/Nc \\
rj = rc - (rc-re)*(n-1)/Nc
\]

GOSUB shape

\[
k = Nm + 4 + n \\
F(k,k) = 1 - \frac{ri^2}{(ri + rj)/(lbm)} - \frac{rj^2}{(ri + rj)/(lbm)}
\]

NEXT n

\[
\text{Shape factor from elements of the cylindrical and frustum sections}
\]

\[
\text{REM to other elements of the cylindrical and frustum sections}
\]

\[
\text{REM between different elements of the frustum section *******}
\]

FOR M = 1 TO Nc-1

FOR n = M + 1 TO Nc

\[
rl = rc - (rc-re)*n/Nc \\
r2 = rc - (rc-re)*(n-1)/Nc \\
r3 = rc - (rc-re)*M/Nc \\
r4 = rc - (rc-re)*(M-1)/Nc \\
h13 = (n-M)*hc \\
h24 = h13 \\
h14 = (n-M+1)*hc \\
h23 = (n-M-1)*hc \\
h = h13 \\
ri = rl \\
rj = r3
\]

GOSUB shape

\[
F13 = FF \\
h = h23 \\
ri = r2 \\
rj = r3
\]

GOSUB shape

\[
F23 = FF \\
h = h14 \\
ri = r1 \\
rj = r4
\]

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GOSUB shape
F14=FF
h=h24
ri=r2
rj=r4
GOSUB shape
F24=FF
i=Nm+4+M
j=Nm+4+n
F(i,j)=r2^2/((r3+r4)*lbc)*(F23-F24)-r1^2/((r3+r4)*lbc)*(F13-F14)
F(j,i)=(r3+r4)/(r1+r2)*F(i,j)
NEXT n
NEXT M
REM *** between different elements of the hot cylindrical section ****
FOR M=1 TO Nm-1
FOR n=M+1 TO Nm
r1=rc
r2=rc
r3=rc
r4=rc
h13=(n-M)*lbm
h24=h13
h14=(n-M+1)*lbm
h23=(n-M-1)*lbm
h=h13
ri=r1
rj=r3
GOSUB shape
F13=FF
h=h23
ri=r2
rj=r3
GOSUB shape
F23=FF
h=h14
ri=r1
rj=r4
GOSUB shape
F14=FF
h=h24
ri=r2
rj=r4
GOSUB shape
F24=FF
i=4+M
j=4+n
F(i,j)=r2^2/((r3+r4)*lbm)*(F23-F24)-r1^2/((r3+r4)*lbm)*(F13-F14)
F(j,i)=(r3+r4)/(r1+r2)*F(i,j)
NEXT n
NEXT M
REM *** Shape factors between elements of the hot cylindrical ***
REM *** section and elements of the frustum section ****
REM
FOR M=1 TO Nm
FOR n=1 TO Nc
    r1 = rc - (rc - re)*n/Nc
    r2 = rc - (rc - re)*(n-1)/Nc
    r3 = rc
    r4 = rc
    h13 = lm-M*lbm+n*hc
    h24 = lm-(M-1)*lbm+(n-1)*hc
    h14 = lm-(M-1)*lbm+n*hc
    h23 = lm-M*lbm+(n-1)*hc
    h = h13
    ri = r1
    rj = r3
    GOSUB shape
    F13 = FF
    h = h23
    ri = r2
    rj = r3
    GOSUB shape
    F23 = FF
    h = h14
    ri = r1
    rj = r4
    GOSUB shape
    F14 = FF
    h = h24
    ri = r2
    rj = r4
    GOSUB shape
    F24 = FF
i = M+4
j = n+4+Nm
F(i,j) = r2^2/((r3+r4)*lbm)*(F23-F24)-r1^2/((r3+r4)*lbm)*(F13-F14)
F(j,i) = (r3+r4)*lbm/((r1+r2)*lbc)*F(i,j)
NEXT n
NEXT M
REM
REM *** Shape factors between the spacer section and elements ***
REM *** of the hot cylindrical section*****
REM
FOR n=1 TO Nm
    r1 = rc
    r2 = rc
    r3 = rc
    r4 = rc
    h13 = n*lbm
    h24 = (n-1)*lbm+la
    h14 = n*lbm+la
    h23 = (n-1)*lbm
    h = h13
    ri = r1
    rj = r3
    GOSUB shape
    F13 = FF
    h = h23
    ri = r2
rj=r3
GOSUB shape
F23=FF
h=h14
ri=r1
rj=r4
GOSUB shape
F14=FF
h=h24
ri=r2
rj=r4
GOSUB shape
F24=FF
j=3
i=3
F(i,j)=r2^2/((r3+r4)*la)*(F23-F24)-r1^2/((r3+r4)*la)*(F13-F14)
F(j,i)=(r3+r4)*la/((r1+r2)*lbm)*F(i,j)
NEXT n
REM
REM *** Shape factors between the spacer section and elements ***
REM *** of the frustum section ***
REM
FOR n=1 TO Nc
r1=rc-(rc-re)*n/Nc
r2=rc-(rc-re)*(n-1)/Nc
r3=rc
r4=rc
h13=lm+n*hc
h24=lm+la+(n-1)*hc
h14=lm+la+n*hc
h23=lm+(n-1)*hc
h=h13
ri=r1
rj=r3
GOSUB shape
F13=FF
h=h23
ri=r2
rj=r3
GOSUB shape
F23=FF
h=h14
ri=r1
rj=r4
GOSUB shape
F14=FF
h=h24
ri=r2
rj=r4
GOSUB shape
F24=FF
j=n+4+Nm
F(i,j)=r2^2/((r3+r4)*la)*(F23-F24)-r1^2/((r3+r4)*la)*(F13-F14)
F(j,i)=(r3+r4)*la/((r1+r2)*lbm)*F(i,j)
REM Shape factors between circular sections and the spacer section, elements of the hot cylindrical section, and elements of the frustum section.
REM
REM **between the end plate and elements of the hot cylindrical section**

r3 = re
r1 = rc
r2 = rc
FOR n = 1 TO Nm
h23 = lc + lm - n*lbm
h13 = lc + lm - (n-1)*lbm
ri = r3
rj = r1
h = h23
GOSUB shape
F31 = FF
ri = r3
rj = r2
h = h23
GOSUB shape
F32 = FF
i = 4
j = 4+n
F(i,j) = F32 - F31
F(j,i) = r3^2/((r1 + r2)*lbm)*F(i,j)
NEXT n
REM **between the end plate and elements of the frustum section**

r3 = re
r1 = re + (rc - re)*((1- (n-1)/Nc)
r2 = re + (rc - re)*((1-n/Nc)

h23 = lc - n*hc
h13 = lc - (n-1)*hc

ri = r3
rj = r1
h = h23
GOSUB shape
F31 = FF
ri = r3
rj = r2
h = h23
GOSUB shape
F32 = FF
i = 4
j = 4+Nm+n
F(i,j) = F32 - F31
F(j,i) = r3^2/((r1 + r2)*lbc)*F(i,j)
NEXT n
REM **between the end plate and the spacer section**

r3 = re
r1 = rc
r2 = rc
h23 = lc + lm
\[ h_{13} = l_c + l_m + l_a \]
\[ r_i = r_3 \]
\[ r_j = r_1 \]
\[ h = h_{13} \]

GOSUB shape

\[ F_{31} = FF \]
\[ r_i = r_3 \]
\[ r_j = r_2 \]
\[ h = h_{23} \]

GOSUB shape

\[ F_{32} = FF \]
\[ i = 4 \]
\[ j = 3 \]

\[ F(i,j) = F_{32} - F_{31} \]
\[ F(j,i) = r_3^2 \left( (r_1 + r_2) \right) F(i,j) \]

REM **between the aperture and elements of the hot cylindrical section**

\[ r_3 = r_a \]
\[ r_1 = r_c \]
\[ r_2 = r_c \]

FOR n = 1 TO Nm
  \[ h_{23} = l_a + (n - 1) \cdot l_b m \]
  \[ h_{13} = l_a + n \cdot l_b m \]
  \[ r_j = r_3 \]
  \[ r_j = r_1 \]
  \[ h = h_{13} \]

GOSUB shape

\[ F_{31} = FF \]
\[ r_i = r_3 \]
\[ r_j = r_2 \]
\[ h = h_{23} \]

GOSUB shape

\[ F_{32} = FF \]
\[ i = 1 \]
\[ j = 4 + n \]

\[ F(i,j) = F_{32} - F_{31} \]
\[ F(j,i) = r_3^2 \left( (r_1 + r_2) \right) F(i,j) \]

NEXT n

REM **between the aperture and elements of the frustum section**

\[ r_3 = r_a \]

FOR n = 1 TO Nc
  \[ r_1 = r_c - n \cdot (r_c - r_e) / N_c \]
  \[ r_2 = r_c - (n - 1) \cdot (r_c - r_e) / N_c \]
  \[ h_{23} = l_a + l_m + (n - 1) \cdot h_c \]
  \[ h_{13} = l_a + l_m + n \cdot h_c \]
  \[ r_j = r_3 \]
  \[ r_j = r_1 \]
  \[ h = h_{13} \]

GOSUB shape

\[ F_{31} = FF \]
\[ r_i = r_3 \]
\[ r_j = r_2 \]
\[ h = h_{23} \]

GOSUB shape

\[ F_{32} = FF \]
\[ i = 1 \]
\[ j = 4 + n + Nm \]

\[ F(i,j) = F32 - F31 \]

\[ F(j,i) = r3A2 / ((r1 + r2) \cdot Ibc) \cdot F(i,j) \]

NEXT n:

REM **between the aperture and the spacer section**

ri = ra
rj = rc
h = la
GOSUB shape

i = 1
j = 3

\[ F(i,j) = 1 - FF \]

\[ F(j,i) = r1A2 / (2 \cdot rj \cdot la) \cdot F(i,j) \]

REM***************************************************************

REM Shape factors for the annulus section are determined by the differences
REM between shape factors of the annulus and aperture combined with
REM an element and the aperture with an element.
REM***************************************************************

REM **between the annulus and elements of the hot cylindrical section**

r3 = rc
r1 = rc
r2 = rc

FOR n = 1 TO Nm

\[ h23 = la + (n-1) \cdot lbm \]

\[ h13 = la + n \cdot lbm \]

ri = r3
rj = r1
h = h13
GOSUB shape

F31 = FF
ri = r3
rj = r2
h = h23
GOSUB shape

F32 = FF
i = 2
j = 4 + n

\[ F(i,j) = (rc^2 \cdot (F32 - F31) - ra^2 \cdot F(1,j)) / (rc^2 - ra^2) \]

\[ F(j,i) = (rc^2 - ra^2) / ((r1 + r2) \cdot Ibm) \cdot F(i,j) \]

NEXT n

REM **between the annulus and elements of the frustum section**

r3 = rc

FOR n = 1 TO Nc

\[ r1 = rc - n \cdot (rc - re) / Nc \]

\[ r2 = rc - (n-1) \cdot (rc - re) / Nc \]

\[ h23 = la + lm + (n-1) \cdot hc \]

\[ h13 = la + lm + n \cdot hc \]

ri = r3
rj = r1
h = h13
GOSUB shape

F31 = FF
ri = r3
rj = r2
h = h23
GOSUB shape
F32=FF
i=2
j=4+n+Nm
F(i,j) = (rc^2*(F32-F31)-ra^2*F(1,j))/(rc^2-ra^2)
F(j,i) = (rc^2-ra^2)/((r1+r2)*lc)*F(i,j)
NEXT n
REM **between the annulus and the aperture**
r1=rc
r2=rc
r3=ra
ri=r1
rj=r1
h=la
GOSUB shape
F12=FF
ri=r1
rj=r3
GOSUB shape
F13=FF
i=2
j=3
F(i,j) = 1-rc^2/(rc^2-ra^2)*(F12-F13)
F(j,i) = (rc^2-ra^2)/((r1+r2)*la)*F(i,j)
REM Shape factors from circular section to other circular sections
REM ******************************************************
REM ********* between the end plate and the aperture *****
ri=re
rj=ra
h=la+lm+lc
GOSUB shape
i=4
j=1
F(i,j) = FF
F(j,i) = re^2/ra^2*F(i,j)
REM ********* between the end plate and the annulus *****
ri=re
rj=rc
h=la+lm+lc
GOSUB shape
i=4
j=2
F(i,j) = FF-F(i,1)
F(j,i) = re^2/(rc^2-ra^2)*F(i,j)
REM ******************************************************
REM Shape factors matrix output
REM ******************************************************
FOR i=1 TO NT
FOR j=1 TO NT-1
PRINT USING "#.###";F(i,j); PRINT USING SPC(1); PRINT
NEXT j
PRINT USING "#.###";F(i,NT)
PRINT
The sum of the shape factors for one element to all the elements of the enclosure is equal to one. This property is used to verify the shape factors previously calculated. For an enclosure of $N$ elements the sum of all the shape factors should equal $N$.

FOR $i=1$ TO $NT$
FOR $j=1$ TO $NT$
$\text{sum}(i)=0$
NEXT $j$
NEXT $i$
SUMT=0
FOR $i=1$ TO $NT$
FOR $j=1$ TO $NT$
$\text{sum}(i)=\text{sum}(i)+F(i,j)$
SUMT=SUMT+$\text{sum}(i)$
NEXT $j$
NEXT $i$
PRINT
FOR $i=1$ TO $NT-1$
PRINT " SUM \";i;\" = ";
PRINT USING "##.####";sum(i)
NEXT $i$
i=NT
PRINT "SUM \";i;\" = ";
PRINT USING "##.####";sum(i)
PRINT 222 :
REM END OF SHAPE FACTOR SECTION
REM COEFFICIENTS MATRIX CALCULATIONS SECTION
REM
FOR $i=1$ TO $NT$
FOR $j=1$ TO $NT-1$
KD=1
IF $i=j$ THEN 400
KD=0
400 :
$A(i,j)=KD-(1-E(i))*F(i,j)$
NEXT $j$
NEXT $i$
PRINT coefficient matrix output
REM
FOR $i=1$ TO $NT$
FOR $j=1$ TO $NT-1$
PRINT USING "#.###";A(i,j);
PRINT SPC(1);
NEXT j
PRINT USING "#.###";A(i,NT)
REM BEGINNING OF TEMPERATURE
REM TEMPERATURE ARRAY INPUT SECTION
REM
Tmean=Top(samp)
Tin=Top(samp)+Tdiff/2:Tout=Top(samp)-Tdiff/2
PRINT Tin,Tout,Tmean
REM
REM Each element is assumed to be isothermal. Angular temperature
REM measurements for the hot cylindrical section and the frustum
REM section are averaged to give one temperature for each band. The
REM axial temperature values are determined from linear extrapolation
REM from band with temperature measurements. Temperature values
REM are in Kelvin.
T(1.)=0 :REM temperature of the aperture opening
FOR n=2 TO 4 :REM temperature of the refractory surfaces
T(n)=((Tmean-40)-32)*5/9+273.15
NEXT n
FOR n=5 TO NT
T(n)=((Tout+(Tin-Tout)*(n-5)/(NT-5)-32)*5/9+273.15
NEXT n
REM ********* OUTPUT THE TEMPERATURE DISTRIBUTION *********
FOR n=1 TO NT
PRINT "T(",n,')= ";T(n)
NEXT n
REM CONSTANT ARRAY CALCULATION SECTION
REM
FOR n=1 TO NT
C(n)=E(n)*S*T(n)^4
NEXT n
REM HEAT FLUX SOLUTIONS
REM
REM Gaussian Elimination method is used to solve for the heat output
REM of each surface, including the total heat lost from the receiver
REM through the aperture.
REM
REM ********* augmented matrix *********
FOR i=1 TO NT
FOR j=1 TO NT
G(i,j)=A(i,j)
NEXT j
G(i,NT+1)=C(i)
NEXT i
GOSUB Gauss
REM
REM OUTPUT OF HEAT LOSS THROUGH APERTURE AND HEAT
REM RADIATED FROM ALL OTHER ELEMENTS
REM

 FOR i=1 TO 4
 READ Nm$
 PRINT Nm$;SPC(5);q(i)
 NEXT i
 FOR i=5 TO Nm+4
 PRINT "hot cylindrical element";SPC(5);q(i)
 NEXT i
 FOR i=Nm+5 TO NT
 PRINT "frustum element";SPC(5);q(i)
 NEXT i
 DATA "aperture","annulus","spacer ring","end plate"
 RESTORE
 REM
 REM
 PRINT
 PRINT
 PRINT SPC(2);’apertureN;TAB( 110);"operating";TAB(20);"radiative"
 PRINT SPC(2);’diameterN;TAB( 110);"temperature";TAB(20);"heat loss"
 PRINT SPC(4);';[in]';TAB(13);"[Watts]"
 PRINT SPC(4);’ra*200/2.54;TAB(13);’Tmean;PRINT TAB(23);q(1)*pi*ra^2
 WRITE#1,ra*200/2.54,Tmean,q(1)*pi*ra^2
 BEEP
 NEXT k
 FOR k=1 TO 5
 BEEP
 NEXT k
 CLOSE#1
END

data:

shape:
FF=.5*(1+(h*h+rj*rj)/(ri*ri)-SQR((1+(h*h+rj*rj)/(ri*ri))^2-4 *(rj/ri)^2))
RETURN
Gauss:
REM

 ** GAUSSIAN ELIMINATION Method **

 For the Gaussian elimination method to work the A(1,1) element
 REM of the augmented matrix can not have a value of one. If A(1,1) is
 REM equal to one then rows of the matrix will be shifted until a non-
 REM zero value is in element A(1,1).
 Flag=0
 IF G(1,1)<=0 THEN Elimination
 Flag=Flag+1
 FOR j=1 TO NT+1
M(j) = G(1,j)
NEXT j
FOR i=1 TO NT-1
FOR j=1 TO NT+1
G(i,j) = G(i+1,j)
NEXT j
NEXT i
FOR j=1 TO NT+1
G(NT,j) = M(j)
NEXT j
GOTO 444
REM ************ PRINT CYCLE ************
CLS
FOR M=1 TO NT
FOR n=1 TO NT
PRINT G(M,n); SPC(5);
NEXT n
PRINT G(M,NT+1)
NEXT M
REM
444 :
GOTO Gauss
Elimination:
REM ************ Gaussian Elimination ************
FOR k=1 TO NT
ss=G(k,k)
FOR j=1 TO NT+1
G(k,j)=G(k,j)/ss
NEXT j
FOR i=k+1 TO NT
ss=G(i,k)
FOR j=1 TO NT+1
G(i,j)=G(i,j)-ss*G(k,j)
NEXT j
NEXT i
NEXT k
REM
FOR k=1 TO NT-1
FOR i=1 TO NT-k
ss=G(i,NT-k+1)
FOR j=1 TO NT+1
G(i,j)=G(i,j)-ss*G(NT-k+1,j)
NEXT j
NEXT i
NEXT k
GOTO 555
REM ************ PRINT CYCLE ************
CLS
FOR M=1 TO NT
FOR n=1 TO NT
PRINT G(M,n); SPC(5);
NEXT n
PRINT G(M,NT+1)
NEXT M
REM

190
555 :
FOR i=1 TO NT
q(i)=G(i,NT+1)
NEXT i
IF Flag=0 GOTO 600
FOR k=1 TO Flag
qq=q(NT)
FOR i=2 TO NT
q(i)=q(i-1)
NEXT i
q(1)=qq
NEXT k
600 :
RETURN
Appendix 16: Flow Meter Factory Calibration Specifications

**FLOW TECHNOLOGY, INC.**  
**MECHANICAL DATASHEET 7412**

Customer: CAL POLY KELLOG UNIT  
Meter: Model #: FT4-BAEXB-LAD-G  
Meter Serial #: 8407412  
End Fitting: MB-32656-G  
Bearing Type: CARBIDE  
Pickoff Type: HI-TEMP MAG  
Pickoff P/N: 80666-104

<table>
<thead>
<tr>
<th>Meter Freq (Hz)</th>
<th>Meter Flow Rate (GAL/Min)</th>
<th>Meter K Factor (P/BAL)</th>
<th>Freq / Viscosity (Hz/CTS)</th>
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<tr>
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<td>408.40</td>
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<td>286.39</td>
<td>0.4313</td>
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<td>205.55</td>
<td>0.3158</td>
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<td>151.03</td>
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<td>113.86</td>
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Calibrated by: R. GARGISAN  
Certified by: [Signature]  

Signal Output: 7 MV @ 11.3 Hz  

Job #: 23194  
Tap #:  
Size: 1/2"  

Cal. Media: FREON TE  
Viscosity: 0.03 cTS  
Temperature: 75.00 °F  
Density: 12.18 #/7  

Calib Inv #: 5109E  
Calib Recal Date: 10/1/87  
Date: 4/10/87

192
## Appendix 17: Flow Meters Voltage Output

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<th>Volts</th>
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<td>0.2686</td>
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<td>0.2112</td>
<td>0.2103</td>
<td>0.2192</td>
<td>0.2200</td>
<td>0.2236</td>
<td>0.2262</td>
<td>0.2290</td>
<td>0.2318</td>
<td>0.2348</td>
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</table>
## CALIBRATION REPORT

**CUSTOMER:** CALIFORNIA POLYTECH STATE UNIVERSITY FOUNDATION  
MECH ENGR DEPT  
3801 W TEMPLE AVENUE  
POMONA CA 91768

**REPORT NO:** OME-70609592  
**TEST ITEM:** KSS-18G  
**TEST DATE:** JUNE 9, 1987

**PURCHASE ORDER NO:** CPF66026AD02

OMEGA ENGINEERING certifies that the above item has been calibrated and its calibration is certified as traceable to the U.S. National Bureau of Standards. Traceability of these measurements is derived from the included NBS test numbers.

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**Reference:** NATIONAL BUREAU OF STANDARDS TEST NO(s): 236495-37778

Tony D'Inverno  
Supervisor, Instrumentation

CAL-4
Appendix 20: Pyromark® Paint Specifications

PYRO/INCOLOY; 540°C CURE; AS RECVD. EMITT

[Graph showing the relationship between temperature (in K) and normal emittance.]
Appendix 20: Pyromark® Paint Specifications

PYROMARK/INCOLOY:540–1; 540C CURE; AS RECD

![Graph showing reflectance vs. wavelength (microns)]
<table>
<thead>
<tr>
<th>Wavelength (nm)</th>
<th>Radiometer [mV]</th>
<th>HP emission</th>
<th>Hot Fun [Watt]</th>
<th>HPS output</th>
<th>HP temperature</th>
<th>HPS output</th>
<th>Black body</th>
<th>HP temperature</th>
<th>HPS output</th>
<th>Black body</th>
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### Table 2 — Properties of Various SARAN WRAP Films

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<th>Remarks</th>
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<td>Kinetic — tens to tens</td>
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<td>Water Absorption</td>
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<td>Water Vapor Transmission Rate (at 100°C)</td>
<td>AS1M D 3955-81 and AS1M D 1453-75</td>
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<td>Optical</td>
<td>See Figure 21 page 14</td>
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<td>White Light Transmission</td>
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<td>Ultraviolet Cut Off</td>
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#### Units

- All types: mm, m, cm, in
- All types: g, kg, lb
- All types: °C, °F
- All types: psi, kpsi
- All types: ft, m
- All types: ft/lb, m/kg
- All types: ft², m²
- All types: N/cm², kgf/cm²

#### SARAN WRAP Films

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<th>16</th>
<th>18</th>
<th>19</th>
<th>S200</th>
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<td>1.71</td>
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<td>8 mm/100 ft²</td>
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See figures 3 and 4 page 17

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**Note:** The values presented are nominal and are not derived from any specific standard or test method.
Light Transmission Characteristics

SARAN WRAP films offer good resistance to sunlight under glass. Such properties as tensile strength, elongation, flexibility, and impermeability to water vapor and gases decrease only slightly. Outdoor exposure to direct sunlight, however, is not recommended. Figure 2 below shows typical light transmission values for SARAN WRAP films.

FDA and USDA Status

SARAN WRAP films, when used unmodified and according to good manufacturing practices — when used for food contact applications — can comply with the U.S. Food, Drug and Cosmetic Act as amended.

Many of these films also have been accepted by the U.S. Department of Agriculture for packaging of meat and meat food products, and poultry products, prepared in Federally inspected plants.

Government regulations are subject to change. While it is the responsibility of users of SARAN WRAP to check the suitability of their intended use with regulatory agencies, resources of The Dow Chemical Company are available to assist customers with pertinent data and other information.

Shrink Characteristics

SARAN WRAP plastic films become highly oriented during manufacture. This orientation makes the film susceptible to shrinkage on exposure to elevated temperatures — a property very desirable in applications such as overwraps. Further, by control of the shrink-inducing temperature, the film user can control the degree of shrinkage obtained.

For use in laminates where shrink is undesirable, preshrunk SARAN WRAP 18L film is available. Differences in the shrinkage rates of 18L and other films are shown in Figures 3 and 4.

\[
\begin{align*}
\kappa &= 0.1712 \times 10^{-8} \left( \frac{\text{wave length}}{\text{micron}} \right)^{2/3} \\
\gamma &= 5.66 \times 10^{-9} \left( \frac{\text{micron}}{\text{micron}} \right)^{2/3} \\
\bar{\gamma} &= 1.0 \times 10^{-3} \text{ m}^2 \text{ K}^{-1} \\
\end{align*}
\]
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