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AN INVESTIGATION INTO THE FEASIBILITY OF AN
AIR CURTAIN FOR A SOLAR CENTRAL RECEIVER

Final Report

By
O. J. McMillan

June 1984

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Nielsen Engineering & Research, Inc.
Mountain View, California

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O. J. McMILLAN

NIELSEN ENGINEERING & RESEARCH, INC.
510 CLYDE AVENUE
MOUNTAIN VIEW, CA 94043

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AN INVESTIGATION INTO THE FEASIBILITY OF AN AIR CURTAIN FOR A SOLAR CENTRAL RECEIVER

SUMMARY

An experimental investigation into the feasibility of using an air curtain to decrease the convective losses from a solar central receiver cavity is described. The investigation makes use of a simplified small-scale model of a cavity, the side walls of which are electrically heated to simulate the effects of the sun. Although optimization of air-curtain performance was not within the scope of this work, a decrease in convective losses of approximately 35% was achieved.

In this report, the apparatus, instrumentation, and test procedures used are described. The results achieved are presented and discussed. Some preliminary notions regarding extrapolating the model-scale results to full size are presented, and a research program to continue the development of the concept is briefly outlined.

INTRODUCTION

For several years, the Department of Energy and the Solar Energy Research Institute have administered a research and development program centered on solar alternatives to the use of fossil fuels for the production of electrical energy. One of the more promising of these alternatives is the solar central receiver concept.

Two basic configurations of central receiver are under consideration, the external receiver and the cavity receiver. In the external receiver, concentrated sunlight illuminates the outer surface of a large cylinder. Thermal energy from the absorbed sunlight is transferred through the cylinder walls to the working medium for the electrical generation cycle. In the cavity receiver, the sunlight is directed through an aperture into an enclosure. In this concept, the heat transfer to the working fluid occurs through the interior walls of the cavity.

Convective losses play a significant role in limiting the ultimate performance of each of these configurations. In fact, the variation of these losses under different conditions may be a pivotal factor in deciding which configuration is the most efficient for a given proposed site. In any event, it is desirable to minimize the convective losses because they represent a significant cost penalty over the intended life of a central-receiver installation.

As part of Phase I of the DOE Innovative Research Program in Solar Thermal Energy, Nielsen Engineering & Research, Inc. (NEAR) has conducted an experimental investigation into the feasibility of using an air curtain to decrease the natural convective losses from a model central receiver cavity. This report documents this work.

The approach taken is to use a simplified small-scale model of a central receiver cavity to investigate whether an air curtain can significantly reduce convective losses if there is no effect of ambient winds. The cavity geometry studied is a right circular cylinder of length-to-diameter ratio near one; the aperture is circular and approximately the same diameter as the cavity. The sides of the cavity are electrically heated to simulate the effects of the sun. The air curtain to be investigated issues radially inward (with provision for variable swirl) from an annular nozzle at the front of the aperture. The width of the air curtain and its mass flow rate are variable.

An energy-balance approach is used to deduce the effect of the air curtain on the convective losses from the cavity. By running the cavity at constant average wall temperature both with and without the air curtain, the wall conduction and aperture radiation losses remain approximately constant in the two cases. Therefore, any change in the power required to heat the cavity with the air curtain operating results from a change in the convection through the aperture.

The remainder of the report is organized as follows. In the next section, a description is given of the apparatus and instrumentation developed for the tests. This is followed by an explanation of the testing procedures used. Then the results are presented together with some discussion. A section which deals with some considerations relevant to extrapolating the model-scale results to a full-scale receiver is next, and the final section consists of some concluding remarks.

APPARATUS AND INSTRUMENTATION

The first tasks in Phase I were the design, construction, and instrumentation of the apparatus necessary to prove the feasibility of the air-curtain concept. A schematic of the resulting test rig is shown in figure 1; figure 2 is a photograph showing approximately the same view. In figure 1, the major components are identified. The solar-central-receiver cavity is simulated by a portion of a 55-gallon barrel, the interior of which is heated by electrical current flowing through ten stainless-steel shim-stock strips. The barrel is supported in an outer enclosure which forms a plenum chamber for the air curtain. Ambient air to supply the curtain enters the outer enclosure at the rear after passing through a metering orifice and a 1200-CFM variable-speed axial blower.

Allowance is made to provide swirl to the air curtain by means of the swirl injection system, also shown in figure 1. This system consists of an auxiliary loop powered by a separate 1200-CFM variable-speed blower. When the swirl blower is operating, air is removed from the plenum at the bottom rear and is reinjected near the top; the injection leg of the swirl loop is turned approximately 10 degrees forward from tangential to provide the reentering air with both axial and swirl velocity components. Additionally, a vent to ambient from the center of the cavity rear wall is provided to allow investigation of whether or not this helps stabilize the curtain when it is operated with swirl. The exhaust pipe for this vent can be either open or closed.

An interior view of the barrel is shown in figure 3. In this photograph, it is seen that the heated strips run longitudinally along the sides of the barrel. They are electrically and thermally insulated from the barrel by a layer of Fiberfrax. The rear wall of the cavity is formed by a piece of stainless-steel sheet to provide a surface of approximately the same emissivity as the strips, although the rear wall is not directly heated. The rear-wall vent opening can also be seen in figure 3.

Further detail of the construction of the simulated cavity can be seen in figure 4. This figure is a cross-section through the barrel and the front end of the outer enclosure. In this figure, it is shown that each strip is clamped at the front and rear and is kept under tension by a coil spring as it heats and expands. A 'rear-wall strongback' has been added to the barrel

to support the resulting axial load. The strip supports are electrically isolated from the barrel. The outside surfaces of the barrel (sides and rear) are thermally insulated to limit the losses through these surfaces. Figure 5, taken with the rear cover of the outer enclosure removed, shows the rear of the barrel. The outer insulation, barrel strongback, tensioner springs, and rear-wall vent pipe are apparent.

Returning to figure 4, it is seen that where the strip passes through the first layer of insulation on the cavity stainless-steel rear wall, a 'shunt strip' has been added. The shunt strip is a thicker piece of shim stock (0.005 in. versus 0.0015 in. for the main strip) which is in direct contact with the main strip. It is provided to lower the heat released in this local region to prevent a hot spot from forming which could compromise the structural integrity of the strip.

A flexible radius is shown in figure 4 in the front corner of the outer enclosure. This provides a smooth seal between the main portion of the outer enclosure (which is stationary) and the front portion which can be moved axially to vary the width of the air curtain.

A final feature of the apparatus shown in figure 4 is one of the series of 23 vent holes in the outer enclosure. The total area of these holes equals the area at the air-curtain outlet. Under normal operation of the air curtain, these vent holes are sealed. However, when it is desired to run a no-air-curtain baseline case, the air-curtain gap is filled with insulation and the vent holes are opened. Maintaining the air flow through the outer enclosure in this way and holding the average strip temperature constant keeps the conduction losses through the outside of the barrel approximately the same in the air-curtain and no-air-curtain cases, as is discussed further in the PROCEDURES section.

Details of the air-curtain outlet are shown in figure 6. Two sheet-metal disks form the nozzle for the air curtain, providing an axial injection angle of approximately 15 degrees into the cavity, as shown. As previously discussed, the air-curtain gap width (b) is adjustable by varying the axial position of the front portion of the outer enclosure. Care was taken to maintain a converging (accelerating) passage for the air-curtain flow and to keep all surfaces as smooth as possible to minimize the turbulence in the air curtain.

In summary, the air-curtain parameters are the gap width, the mass flow rate $\dot{m}$ (proportional to the initial radial inflow velocity V_r), the swirl rate (corresponding to the initial tangential velocity V_θ), and whether or not the back vent is open. All of the testing was done with a 15° axial injection angle (into the barrel).

The major instrumentation used allowed measurement of $\dot{m}$, the electrical power ($\dot{e}$) supplied to heat the strips in the simulated cavity, and the temperature at selected locations in the cavity. The air-curtain mass flow rate is measured using a standard ASME sharp-edged orifice. Strip electrical power is determined by separate measurements of the strip voltage drop (using a voltage-divider network) and current (using a calibrated shunt).

Cavity temperatures are measured at thirteen locations using chromel-alumel (type K) thermocouples. The thermocouples are located beneath the heated strips using the "sandwich" construction shown in figure 7. Each thermocouple is electrically insulated from the heated strip by a piece of mica sheet; the extra layer of shim stock helps to protect the mica from abrasion caused by the relative motion of the parts due to thermal expansion and contraction. Alternate strips had thermocouples located on the strip axial centerline at positions corresponding to approximately one-quarter and three-quarters of the strip length, for a total of ten thermocouples. The remaining three thermocouples were located behind the unheated stainless-steel rear wall on a circle of approximately half the cavity diameter at $\theta = 100^\circ$, 240° , and 340° (measured clockwise from the top, looking into the cavity from the front).

The swirl condition was characterized by measurement of the voltage and current supplied to the swirl-injection blower, as well as by flow visualization of the angle (β) between the velocity vector at the air-curtain outlet and a radial line.

PROCEDURES

A schematic representation of a control volume for the analysis of the rate of energy transfer to the simulated cavity is contained in the following sketch.

Applying the principle of conservation of energy, when the apparatus is at thermal equilibrium:

$$\dot{e} = \dot{q}_{\text{conv}} + \dot{q}_{\text{cond}} + \dot{q}_{\text{rad}} + \dot{q}_{\text{cond},s} + \dot{q}_{\text{cond},b} + \dot{q}_{\text{conv},v} \quad (1)$$

where

- $\dot{e}$ = power supplied to strips
- $\dot{q}_{\text{conv}}$ = rate of convection of energy out of aperture
- $\dot{q}_{\text{cond}}$ = rate of conduction heat transfer out of aperture
- $\dot{q}_{\text{rad}}$ = rate of radiation heat transfer out of aperture
- $\dot{q}_{\text{cond},s}$ = rate of conduction heat transfer out of cavity side walls
- $\dot{q}_{\text{cond},b}$ = rate of conduction heat transfer out of cavity back wall
- $\dot{q}_{\text{conv},v}$ = rate of convection of energy out of cavity rear vent

In this relation, the rate of work done by shear forces at the open end of the cavity and the conduction and radiation out of the back vent opening have been neglected because they are small in comparison to the other terms.

Rearranging eq. (1),

$$\dot{q}_{\text{conv}} + \dot{q}_{\text{cond}} + \dot{q}_{\text{conv},v} = \dot{e} - (\dot{q}_{\text{rad}} + \dot{q}_{\text{cond},s} + \dot{q}_{\text{cond},b}) \quad (2)$$

Consider a hypothetical situation where the apparatus is operated in two discrete equilibrium conditions with no back-wall vent flow in which the temperature distribution in the cavity and the boundary conditions on the outside of the insulation around the barrel remain identically the same. The latter condition is approached by maintaining a constant air flow through the outer enclosure. In condition 1, there is no air curtain (the air flow exits through the vent holes in the outer enclosure). In condition 2, the air curtain is operating. Eq. (2) can be written for each of these equilibrium situations, and in these expressions the terms in parentheses on the right-hand side are the same. Then,

$$(\dot{q}_{\text{conv}} + \dot{q}_{\text{cond}})_1 - (\dot{q}_{\text{conv}} + \dot{q}_{\text{cond}})_2 = \dot{e}_1 - \dot{e}_2 \quad (3)$$

and a change in power to the strips represents a change in the net non-radiative energy loss out of the aperture. The change due to the air curtain in the combined convection and conduction losses through the aperture is what is sought in this study and is hereafter referred to as the convective loss.

In practice, of course, when the air curtain is turned on the temperature distribution in the cavity changes. But if the average temperature is maintained constant, eq. (3) holds approximately, and changes in strip power are indicative of changes in the convection losses from the cavity opening. The consequences of changes in radiation and conduction associated with a change in temperature distribution are minimized if these effects are small in comparison with the convective losses from the cavity opening. Design calculations for this apparatus indicate that the side- and back-wall conduction losses are only about 10% of the aperture convection losses, and the radiation is about 40%,

so reasonably small changes in these quantities have proportionally small effects on the changes in convective losses indicated by a given change in strip power. Using the design values, eq. (3) indicates that a 25% change in strip power corresponds to a 38% change in convective losses from aperture.

Use of the energy-balance method to infer changes in the convective losses requires that each data point be taken with the rig at thermal equilibrium. It was determined that a satisfactory approximation to thermal equilibrium for the apparatus was achieved when the thermocouples imbedded behind the metal rear wall of the cavity indicated a change in temperature of less than $0.5^{\circ}\text{F}/\text{min}$. Using this criterion, the time required for the apparatus to reach a new equilibrium state is on the order of one-half to one hour after a change in level of the power supplied to heat the strips. An initial warmup period of approximately two hours is required to bring the entire apparatus to thermal equilibrium at the initial operating temperature.

This large thermal capacitance means that it is impractical to operate the rig at constant temperature while screening the air-curtain parameters. Therefore, it was decided that the preliminary screening studies would be accomplished at a constant level of strip power. If for a certain combination of parameters the average strip temperature increases, a reduction in the net value of the thermal losses is indicated. A few conditions determined to be interesting in this way can then be investigated at constant average temperature.

All testing was done in the hours between 4 a.m. and 10 a.m. to minimize the effects of atmospheric winds. No testing was done if there were appreciable winds.

RESULTS AND DISCUSSION

Using the above procedures, testing was initiated with an air-curtain gap width of 0.3 inches. It was determined that the air curtain could reduce convective losses, but only at very low mass flows. These flow rates result in a pressure difference at the orifice which is a very small fraction of the range of the pressure transducer being used, a situation which results in unacceptable accuracy. It was therefore decided to increase the gap width to the largest value to be tested, 1.2 inches. At this condition, values of air-curtain radial velocities similar to the 0.3-inch case are achieved at larger mass flow rates, resulting

in operation in a satisfactory portion of the range of the orifice pressure transducer. The remainder of the test was conducted using a gap width of 1.2 inches.

Figure 8 shows the constant-power results for a non-swirling air curtain (rear-wall vent closed). In this figure, the average strip temperature is shown as a function of air-curtain mass flow rate at a strip power level of 4.60 kw. It is seen that increasing the mass flow increases the average temperature until a mass flow rate of approximately 0.4 lbm/sec, where a maximum is seemingly reached. While instrumentation limitations precluded reaching a higher flow rate and proving this is indeed a local maximum, previous experience at the smaller gap setting suggests that at higher flow rates the air curtain begins to entrain excessive amounts of hot air from within the cavity, and the average temperature drops. The filled symbol in figure 8 is the constant-power baseline (no-air-curtain) case for 0.4 lbm/sec. This baseline point was achieved by filling the air curtain gap, but allowing the same mass flow to escape through vent holes in the outer enclosure. The indicated change (about 80°F) in the average strip temperature between these two cases at 0.4 lbm/sec indicates a considerable decrease in losses with the air curtain operating.

Note that there are three points at $\dot{m} = 0.4$ lbm/sec with the air curtain operating. One of these points was taken just before and another was taken just after the no-air-curtain case in figure 8; the third point was taken on a different day. This excellent repeatability is evidence of the suitability of the procedures described earlier to eliminate transient effects from the data.

The temperature distributions of the heated strips for the three points with the air curtain operating at $\dot{m} = 0.4$ lbm/sec are shown in figures 9(a) and (b). Figure 9(a) shows the temperatures indicated by the thermocouples near the front ends of the strips; figure 9(b) shows the temperatures measured near the rear ends of the strips. These figures are polar plots, with the polar angle to the measurement location (looking rearward from the front of the cavity) as the independent variable; strip temperature is indicated by radial position. The strip orientations are indicated in the central portion of the figures. As in figure 8, generally good repeatability among the runs is indicated by the detailed temperature distributions.

In figure 10, the average of the temperatures at each location in figure 9 is plotted as an open symbol. For comparison, the temperatures measured for the no-air-curtain case at the same strip power are shown using filled symbols. With no air curtain, the front ends of the strips are cooler than the rear ends, except near the top of the barrel (strip 5, $\theta \approx 15^\circ$). This is a result of the natural-convection pattern for this case, when hot air from the cavity flows out at the top of the aperture and is replaced by cold air which flows in near the bottom. With the air curtain, both the front and rear temperatures are increased (except at $\theta \approx 15^\circ$) because the airflow out of the aperture is reduced, and the front ends of strips 1, 3, and 9 are actually hotter than the rear. The reason for the relatively low temperature indicated at the front end of strip 5 with the air curtain operating is not understood at this time. Returning to figure 9, this is the location showing the largest scatter in the readings; it is possible there are unsteady effects present which are not adequately accounted for by the averaging process used here. The average strip temperatures for the air-curtain and no-air-curtain cases in figure 10 are 865°F and 788°F , respectively. These values are shown as dashed circular lines in that figure.

To maintain the same average temperature as the air-curtain case (865°F) without an air curtain (with 0.4 lbm/sec. still exiting from the vent holes in the outer enclosure) required 5.65 kw, an increase of 23%. Using estimates of the relative magnitudes of the convection, radiation and conduction losses based on design calculations, this 23% change in power corresponds to a change in the convective losses of about 35%.

The temperature distributions for the air-curtain and no-air-curtain cases with constant average strip temperature of 865°F are shown in figure 11. The conduction and radiation losses for these two cases are nearly constant, as discussed earlier. In this figure, it seems that the air curtain has the desired effect of preventing the cooling of the front of the strips by eliminating the natural-convection pattern which prevails in the no-air-curtain case. Again, the low temperatures for strip 5 are not understood.

While funding limitations in Phase I precluded any attempt at optimizing the air-curtain performance, the single operating point of figure 11 demonstrates that the air curtain can reduce the convective losses from a simulated cavity receiver by a considerable amount.

The effects of introducing swirl and/or opening the rear-wall vent are shown in Table 1. These results were obtained at constant strip power ($\dot{e} = 4.60$ kw) at the mass flow rate ($\dot{m} = 0.4$ lbm/sec) which resulted in the improved performance indicated above with no swirl.

Table 1.- Effects of Swirl and Back-Vent Flow on the Average Strip Temperature, $\dot{e} = 4.60$ kw, $b = 1.2$ in, $\dot{m} = 0.4$ lbm/sec

<u>Swirl</u> <u>Condition</u>	<u>Average</u> <u>Strip Temperature, °F</u>	
	<u>Back Vent</u>	
	<u>Closed</u>	<u>Open</u>
0	865	822
1	856	829
2	859	824

The two swirl conditions were characterized by smoke flow visualization as: condition 1, low swirl, $\beta \approx 1-2^\circ$; and condition 2, high swirl, $\beta \approx 25^\circ$. Under the conditions of these tests, with the vent closed, it is seen that swirl has essentially no effect on the average strip temperature relative to the no-swirl case. However, the temperature distribution in the simulated cavity did change a significant amount for the high-swirl case. Figure 12 compares the strip temperatures measured in the no-swirl and high-swirl cases. It is seen that swirl generally results in cooling of the front ends of the strips and heating of the back ends. On the basis of these limited data, the effects of including swirl are clearly not well defined, and this parameter should be studied further.

Opening the rear-wall vent, however, serves mainly to provide an additional loss path (table 1 and figure 13). No significant changes in flow pattern or stability were observed for the swirling air curtain as a result of opening this vent, so further investigation of this parameter seems unwarranted.

SCALING CONSIDERATIONS

The question of ultimate interest which underlies this work is: Can an air curtain reduce the convective losses from a full-scale cavity receiver sufficiently that a net economic benefit results over the life of the installation? The inputs to the economic analysis which will provide the answer to this question are performance estimates which depend on a basic understanding of the physical processes involved in the full-scale application.

These physical processes are complex. Because of the large size and high temperatures involved, the pertinent phenomena include mixed free and forced convection in ranges of the governing parameters that are not well studied. Also, because these receivers will be located on tall towers and are thus exposed to strong and gusty winds, the effects of turbulence may be important. Studies designed to understand these phenomena would be prohibitively expensive if carried out at full scale, so carefully designed model-scale tests are required. But the question remains: How can the model-scale results be applied to full scale? Here we examine some preliminary notions regarding scaling.

Designs for full-scale or prototype-scale cavity receivers (see, for example, references 1 and 2) generally feature complicated internal cavity geometry and apertures which vary in size relative to the characteristic dimension of the cavity. The apertures have varying orientations relative to the gravity vector, and the receivers are to be immersed in a turbulent wind of varying speed and direction. For the purpose of the present discussion, let us ignore the influence of these complexities and consider the Phase I construction: a cylindrical cavity geometry of length-to-diameter ratio (L/D) near one; a circular aperture of diameter essentially equal to the cavity diameter; the aperture and cavity axes perpendicular to the gravity vector; and no influence of external winds. Dimensional analysis of this problem (following references 3 and 4) reveals that the average Nusselt number has a functional dependence which can be expressed as:

$$Nu = f(Re, Gr, Pr, T_w/T_\infty, b/D, V_\theta/V_r, Tu_c) \quad (4)$$

where

$$Nu = \frac{hD}{k}$$

$$Re = \frac{V_r b}{\nu}$$

$$Gr = \frac{g\beta D^3 (T_w - T_\infty)}{\nu^2}$$

$$Pr = \frac{\mu c_p}{k}$$

Tu_c = turbulence intensity of the air curtain

In these relations, h is defined as the average cavity convective heat transfer coefficient, $h = \dot{q}_{conv}/[A(T_w - T_\infty)]$, where A is the interior surface area of the cavity and T_w the average cavity wall temperature, k is the thermal conductivity, μ is the viscosity, ν is the kinematic viscosity and c_p is the specific heat of the fluid involved. In the definition of the Grashof number Gr , β is the coefficient of thermal expansion and g is the acceleration due to gravity.

In a perfectly scaled experiment, besides geometric similitude, all of these dimensionless groups retain their full-scale values. If this is achieved, it is not necessary to know the functional relationship implied in eq. (4) because the dimensionless model-scale results are directly applicable to full scale. However, if the restriction to model-scale tests in air is made, as is done here, this perfect similitude is impractical and some knowledge of f must be gained to allow for the extrapolation to full scale. Because the Prandtl number is essentially constant for air, its influence in this extrapolation can be neglected. The extent of extrapolation required is indicated by the fact that in a full-scale central receiver cavity, the Grashof number would be on the order of 10^{14} ; in the current experiment, it is approximately 10^{10} .

The free convective heat transfer from a cavity receiver without an air curtain has received considerable attention. Recently, in reference 5, it has been shown that for a cubic

cavity, the convective heat transfer does not depend on the cavity's size; that is

$$\text{Nu} = 0.088 \text{Gr}^{1/3} \left(\frac{T_w}{T_\infty} \right)^{0.18} \quad (5)$$

$$10^7 < \text{Gr} < 1.2 \times 10^{12}$$

If it is supposed that a similar relation holds for cylindrical cavities of $L/D = 1$, the form of the scaling law for the "baseline" case of no air curtain is known, even though the multiplicative constant in this relation and the exponent on the temperature factor are not.

For the case where there is an optimal air curtain present, leaving for the moment the question of how to specify this air curtain, no such statement can be made. At this point, it is not even possible to specify what form the correlating dimensionless parameters will take. It is known that the value of the parameter $Ri = \text{Gr}/\text{Re}^2$ (the Richardson number) is indicative of the boundaries between the free-, mixed-, and forced-convection regimes. For example, for a vertical flat plate, the mixed-convection zone lies in the approximate range $0.7 < Ri < 10$ where the length scales in Grashof and Reynolds numbers are taken as the plate height and length, respectively (reference 6). In the mixed-convection regime, the heat transfer scales on the Richardson number. For $Ri < 0.7$, the heat transfer is dominated by the effects of forced convection and scales on Reynolds number. For $Ri > 10$, free convection dominates and scales on Grashof number. In the current tests, if a Richardson number is formed using characteristic lengths D in the Grashof number and b in the Reynolds number, the largest reduction in convective heat transfer was found at $Ri \approx 160$. If the Richardson number is formed using the cavity diameter throughout, then $Ri = 0.7$. If the boundaries between heat transfer regimes established on a flat plate carry over to the present case, it is seen that at this point it is impossible to establish the heat transfer regime and thus the form of the governing dimensionless parameter. Further information is necessary to make this specification and to allow development of the scaling law for the heat transfer from a cavity with an air curtain.

With regard to the determination of the correlating parameter which will allow specification of the appropriate air curtain for a given cavity condition, it is obvious that insufficient information exists at this point. However, if one neglects swirl and considers that effective operation of the air curtain implies a balance between the buoyant forces which serve to drive the air from the cavity and the inertial forces of the air curtain, Froude-number scaling leads to a group of the form $(Gr/Re^2)(b/D)^2$, where the length scales in the Grashof and Reynolds numbers are D and b , respectively. It should be remembered that dimensional analysis alone cannot result in specification of a unique combination of dimensionless groups. Thus, in future studies, as a starting point, correlation should be sought between the establishment of an effective air curtain and a critical value of a group of the form $(Gr/Re^2)(b/D)^n$.

CONCLUDING REMARKS

The intent of this study was to demonstrate the feasibility of using an air curtain to reduce the convective losses from a model solar central receiver cavity. This goal was achieved by demonstrating an air-curtain configuration which reduced by about 20% the power required to maintain the average heated-strip temperature at the no-air-curtain level. It is estimated that this reduction in power corresponds to a decrease in convective losses of approximately 35%.

Because only limited resources were available for this work, a simplified model of a cavity was used and only the minimum essential instrumentation could be employed. Additionally, optimization of the air-curtain performance was not within the scope of the effort. Therefore, it is considered likely that additional performance gains are possible beyond the level already achieved.

Based on its potential for significantly decreasing the losses from a cavity receiver, it is recommended that the air curtain concept be investigated further. It is appropriate at this point that the additional work be at model scale, but it is also important to begin to develop the ability to scale the results to full size. The following research program is recommended.

1. Determine the optimum air curtain for the Phase I apparatus and the decrease in convective losses it achieves. In this determination, the effect of swirl in the air curtain should be examined further, but the use of the back-wall vent can probably be excluded.
2. Investigate the influence of cavity, aperture, and air-curtain geometry on air-curtain performance through a series of model-scale experiments.
3. Determine the effects of atmospheric winds on air-curtain optimization and performance at model scale.
4. Develop a computational capability to allow for extrapolation of model-scale results to full scale.

Successful completion of this entire program would result in the ability to specify an air curtain for a full-size receiver and a means of estimating the decrease in convective losses this air curtain would achieve. If the performance already demonstrated carries over to full scale, the associated cost savings to consumers of the electrical power from a cavity central receiver would constitute a powerful incentive for the continuation of this work.

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Figure 1.- System schematic.

Figure 2.- Photograph of test apparatus.

Figure 3.- Interior view of simulated cavity.

Figure 4.- Barrel cross-section.

Figure 5.- View of the rear of the barrel. The rear cover of the outer enclosure has been removed.

Figure 6 .- Air-curtain outlet detail.

Figure 7.- Thermocouple installation (typical). Not to scale.

Figure 8.- Average strip temperature vs. mass flow rate.

(a) Front thermocouples

Figure 9.- Strip temperature distribution with the air curtain; $b = 1.2$ in., $\dot{m} = 0.4$ lb_m/sec, $\dot{e} = 4.60$ kw, vent closed, no swirl.

(b) Rear thermocouples

Figure 9.- Concluded.

Figure 10.- Strip temperature distributions; $b = 1.2$ in.,
 $\dot{m} = 0.4 \text{ lb}_m/\text{sec}$, $\dot{e} = 4.60 \text{ kw}$, vent closed, no swirl.

Figure 11.- Strip temperature distributions; $b = 1.2$ in.,
 $\dot{m} = 0.4$ lb_m/sec, $\bar{T} = 865^{\circ}\text{F}$, vent closed, no swirl.

Figure 12.- Strip temperature distributions; $b = 1.2$ in.,
 $\dot{m} = 0.4 \text{ lb}_m/\text{sec}$, $\dot{e} = 4.60 \text{ kw}$, vent closed.

Figure 13.- Strip temperature distributions; $b = 1.2$ in.,
 $\dot{m} = 0.4 \text{ lb}_m/\text{sec}$, $\dot{e} = 4.60 \text{ kw}$, high swirl.