5105-129
Solar Thermal Power Systems Project
Parabolic Dish Systems Development

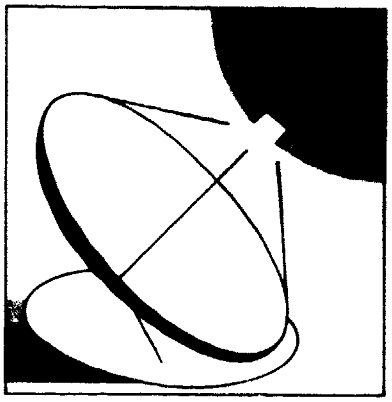
DOE/JPL-1060-65
Distribution Category UC-62

(NASA-CR-173732) SECOND-LAW EFFICIENCY OF SOLAR-THERMAL CAVITY RECEIVERS (Jet Propulsion Lab.) 61 p HC A04/MF A01 CSCL 10A

N84-28223
Unclas
G3/44 19842

Second-Law Efficiency of Solar-Thermal Cavity Receivers

P.I. Moynihan



October 1, 1983

Prepared for
U.S. Department of Energy
Through an Agreement with
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This report was published by the Jet Propulsion Laboratory, California Institute of Technology, for the U.S. Department of Energy through an agreement with the National Aeronautics and Space Administration (NASA Task RE-152, Amendment 327; DOE/NASA Interagency Agreement No. DE-AM04-80AL13137).

The JPL Solar Thermal Power Systems Project is sponsored by the U.S. Department of Energy and is part of the Solar Thermal Program to develop low-cost solar thermal and electric power plants.

ABSTRACT

Properly quantified performance of a solar-thermal cavity receiver must not only account for the energy gains and losses as dictated by the First Law of thermodynamics, but it must also account for the quality of that energy. However, energy quality can only be determined from the Second Law. In this paper an equation for the Second-Law efficiency of a cavity receiver is derived from the definition of available energy or "availability" (occasionally called exergy), which is a thermodynamic property that measures the maximum amount of work obtainable when a system is allowed to come into unrestrained equilibrium with the surrounding environment. The fundamental concepts of the entropy and availability of radiation are explored from which a convenient relationship among the reflected cone half angle, the insolation, and the concentrator geometric characteristics is developed as part of the derivation of the Second-Law efficiency. A comparison is made between First- and Second-Law efficiencies around an example of data collected from two receivers that were designed for different purposes. The author attempts to demonstrate that a Second-Law approach to quantifying the performance of a solar-thermal cavity receiver lends greater insight into the total performance than does the conventional First-Law method.

PREFACE

This report is the result of a Directed Research effort performed at the University of Southern California in partial fulfillment of the requirements for the Degree of Engineer in Mechanical Engineering. The work was conducted for the academic advisory committee comprising Professors R. Choudhury, S. Lampert, and H. T. Yang, chaired by Dr. Lampert who served as Academic Advisor. Although this work itself was not funded by the Solar Thermal Project at the Jet Propulsion Laboratory, it was focused upon a specific area applicable to the overall Solar Thermal effort and was published with Project support.

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SECTION I

INTRODUCTION

Power generated from a point-focusing, solar-thermal collector is based on the principle of direct normal sunlight being focused through the aperture of a cavity receiver from a parabolic mirrored-surface concentrator. Once in the cavity, the solar energy is then absorbed by the receiver and transferred to a working fluid. The working fluid would be a phase change medium such as water or an organic fluid for a Rankine-cycle application, or a gaseous medium such as air for a Brayton-cycle application or helium for a Stirling-cycle application. The ultimate application of the working fluid is to drive a turbine or displace a piston to do work.

Although the principles are similar for linear troughs and central receivers (i.e., "power towers"), the scope of this report is limited to the cavity receivers of parabolic dish collector systems.

The established approach for quantifying receiver performance is from First-Law analysis wherein the efficiency is defined as the energy absorbed by a working fluid flowing through the receiver divided by the solar energy passing through the aperture. The insolation at the aperture is typically corrected for the optical losses sustained during the reflection process [1,2,3].* However, a proper method of quantifying receiver performance must not only account for the energy balance, but it must also account for the quality of that energy. The accounting for energy quality can be accomplished only through a Second-Law approach.

In this report an attempt is made to establish a practical, working method whereby Second-Law analysis can be applied to determining the performance of cavity receivers. Furthermore, an argument is ventured and justified that this Second-Law method should be adopted as the preferred approach.

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*Numbers in brackets designate references at end of paper.

SECTION II

SECOND-LAW APPROACH

If the First Law could be said to be the law of energy, then the Second Law could be called the law of entropy. The most common method of determining the thermodynamic performance of power-producing systems is through a First-Law energy balance, a method that is often not more than a simple accounting procedure wherein energy gained is credited and energy loss is debited. For equilibrium, the credits and debits balance. However, the conventional definition of energy accounts only for the quantity of energy involved, and does not consider the value or quality of that energy. For example, everyone would agree that a Btu of electricity has greater value than a Btu of heat rising from a warm surface. Because there is critical information missing from a purely First-Law approach, it is not a true measure of the usefulness of the energy available.

Some measurement of the quality of the energy must be brought into the equation in order to properly assess the degree to which it is available to do work. The universally accepted parameter to provide such a function is the thermodynamic property called "available energy," "availability," or "exergy."

In this report this discriminator will generally be referred to as availability, which will be defined in this section with examples given of its various forms. Equations for availability will be developed for both direct and scattered radiation that will later be applied to solar-thermal cavity receivers. Since entropy is implicit in the definition of availability and since the availability source for solar cavity receivers is radiation, a derivation of the entropy of radiation will be presented to provide the reader with insight into its concept. The final expression for the entropy of radiation may be unfamiliar to many people because of the influence of radiation pressure.

A. AVAILABILITY

As defined, the availability of a system is a property that measures the maximum amount of work obtainable when the system is allowed to come into unrestrained equilibrium with the surrounding environment. When the system is in the same condition as its surrounding environment, it is in a "dead state," which is, by definition, a state of zero availability. Although many authors have offered various statements of the definition of availability, simply stated, it is that part of energy that can be converted for a useful function under given environmental conditions.

The earliest use of the term availability seems to be traceable to Tait in 1868, although Maxwell referred to availabile energy in his "Theory of Heat" published in 1871 [4]. Both alluded to the same concept. In 1873, Gibbs provided the analytical basis for determining available energy through the concept of "dissipated energy" that years later, in 1931, Keenan was able to present in simple, more practical terms [5, 6]. Keenan is accredited with having coined the expression "dead state". Since then, a fair, although not extensive, amount of work has been done in this area with a sizable portion of it performed outside the United States. There are many references to availability in foreign literature [7, 8, 9, 10, 11, 12] where the term exergy seems to be preferred.

Although several authors use the symbols "A", "a", or "Φ" when defining availability [13,14,15,16], the symbol "B" (or "b" for specific availability) proposed by Keenan [6] seems to be more commonly accepted and will be used in this report.

Availability is defined in equation form as

$$B = E + p_oV - T_oS - (E_o + p_oV_o - T_oS_o)$$

where

E = U + KE + PE + ..., total energy

U = internal energy

KE = kinetic energy

PE = potential energy

p = pressure

T = temperature

V = volume

S = entropy.

The subscript, o, refers to the dead state.

For this paper the kinetic, potential, and other energy sources will be assumed negligible compared with the internal energy source. Therefore, the general definition becomes

$$B = U + p_oV - T_oS - (U_o + p_oV_o - T_oS_o). \qquad (1a)$$

Equation (1a) is derived with reference to the amount of work obtainable between an initial state and the surroundings, or dead state, and it must not be confused with the similar expression for an open system that is presented next.

For an open system where the flow energy must be included, the expression often used [17] is

$$B = H - T_oS - (H_o - T_oS_o)$$

where

$$H = \text{enthalpy}$$
$$= U + pV.$$

Therefore, the above equation can be written as

$$B = U + pV - T_oS - (U_o + p_oV - T_oS_o). \tag{1b}$$

This expression for availability differs from that of Equation (1a) because of the influence of the flow energy, pV, that is a necessary contribution to open, steady-flow systems [18].

For a constant-volume closed system, the expression for availability becomes

$$B = U - T_o\, S - (U_o - T_o\, S_o). \tag{1c}$$

Many authors implicitly recognize the term in parentheses in Equation (1a) as the dead state and choose to write availability merely as

$$B = U + p_o\, V - T_o\, S \tag{1d}$$

for the general equation, and

$$B = H - T_o\, S \tag{1e}$$

and

$$B = U - T_o\, S \tag{1f}$$

for the open system and the constant-volume closed system, respectively.

The specific availability or availability density, b, which will be used in later derivations, is defined as

$$b = \frac{B}{V}\,. \tag{1g}$$

An availability balance of a system can now be written as follows:

$$\text{Availability into system} = \text{Availability out of system} + \text{Availability destroyed}$$

wherein the destruction of availability is the irreversibility, which has been quantified by Gaggioli [15] and others as

$$I = T_o \Delta S. \tag{2}$$

Some availability is destroyed in all real processes, for unlike energy, availability is not conserved. Availability lost from the system is implicit in the term "availability out of system". Many authors [e.g., 16,19] prefer to identify the lost availability term explicitly and choose to write the availability balance as

$$\text{Availability into system} = \text{Availability to products} + \text{Availability lost} + \text{Availability destroyed} .$$

Heat into and out of a closed system is commonly related to the internal energy through the First Law as

$$\begin{aligned} \delta Q &= dU + \delta W \\ &= dU + pdV \end{aligned} \tag{3a}$$

for a system involving work, δW, and as

$$\delta Q = dU \tag{3b}$$

where no work occurs.

The conventional expression for entropy for a reversible condition consistent with the above equations is

$$dS = \frac{\delta Q}{T} , \tag{4}$$

which will be used in subsequent derivations in this report.

B. ENTROPY OF RADIATION

One would surmise that an entity like radiation that has energy and can produce a temperature should also have entropy, and that is the case indeed. As a means of quantifying the entropy of radiation, Planck [20] suggested the use of an imaginary, well insulated, frictionless cylinder into which a piston is placed. (Other authors like Spanner [21], Richtmyer and Kennard [22], and Petela [7] have embellished upon this concept.) All inner surfaces of the cylinder, as well as the back face of the piston, are perfect reflectors, and the resulting cavity volume V is a vacuum. A non-volatile, black, minute material object is placed within the cavity. If the piston were displaced by an elemental volume dV, then the system will be out of equilibrium unless the energy per unit volume of the radiation is held constant. If, as suggested by Planck, the total energy of radiation is denoted as U, where

$$U = uV \ , \tag{5}$$

then u represents the energy density or specific energy, given by

$$u = \frac{U}{V} \ . \tag{6}$$

In order for the energy per unit volume to remain constant during the displacement dV, a quantity of energy dU must appear. Since the only source of this energy is the solid object in the cavity, and since this body does no work, it must give up its energy as heat. Hence, the material object has experienced a decrease of entropy by the amount of dU/T. The whole process has, by definition, taken place reversibly, so the net change of entropy must be zero. A zero net entropy change can occur only if the radiation has experienced an increase in entropy equal to that lost by the solid body. Hence, radiation also possesses entropy.

However, there is more to the equation of the entropy of radiation than merely dU/T. As derived by theory and backed up by experiment [20,21,22,23], radiation also exerts pressure. This pressure, which is referred to by Planck as "Maxwell's radiation pressure," has the magnitude of

$$p = \frac{u}{3} \ . \tag{7}$$

Hence, in order to maintain equilibrium within the cavity volume after the incremental displacement dV, not only must heat energy from the object be released to create new radiation, but also as dictated by the First Law, an additional quantity of energy must be given up to equal the work done on the piston. Since the work done on the piston is pdV, the entropy increase associated with this work, dS_w, is found from Equations (4), (5), and (7) for the adiabatic system to be

$$dS_w = \frac{pdV}{T} = \frac{u}{3T}\,dV \ . \qquad (8)$$

But

$$dU = Vdu + udV \ . \qquad (9)$$

However, for constant energy per unit volume

$$du = 0 \ . \qquad (10)$$

Therefore,

$$dS_w = \frac{dU}{3T} \qquad (11)$$

which makes the total entropy lost by the minute material object as

$$dS = \frac{dU}{T} + \frac{dU}{3T} = \frac{4}{3}\frac{dU}{T} \ . \qquad (12)$$

Hence, the radiation has acquired a net increase of entropy by the amount of

$$dS = \frac{4}{3}\,\frac{dU}{T} \ . \qquad (13)$$

Since the increment of radiation is of identically the same quality as the remaining radiation, the integration constant is zero, and Equation (13) becomes

$$S = \frac{4U}{3T} \ . \qquad (14)$$

Equation (14) is valid for isotropic, unpolarized radiation of any wavelength or combination of wavelengths.

Planck offered a direct derivation of Equation (14) for perfectly reversible processes in equilibrium for which there is no net increase in entropy. For a reversible adiabatic process, the entropy remains constant. Therefore,

$$dS - \frac{\delta Q}{T} = 0. \tag{15}$$

But, from Equation (3a) for the First Law

$$\delta Q = dU + pdV. \tag{16}$$

Hence,

$$dS = \frac{dU + pdV}{T}. \tag{17}$$

If the volume V and temperature T are taken as independent variables, then from Equations (7) and (5) the following can be derived:

$$dS = \frac{dU + pdV}{T}$$

$$= \frac{udV + Vdu + \frac{u}{3}dV}{T} \tag{18}$$

$$= \frac{V}{T}du + \frac{4}{3}\frac{u}{T}dV \tag{19}$$

$$= \frac{V}{T}\frac{du}{dT}dT + \frac{4}{3}\frac{u}{T}dV, \tag{20}$$

which is of the form

$$dS = \left.\frac{\partial S}{\partial T}\right)_V dT + \left.\frac{\partial S}{\partial V}\right)_T dV \tag{21}$$

where

$$\left(\frac{\partial S}{\partial T}\right)_V = \frac{V}{T}\frac{du}{dT} \tag{22}$$

and

$$\left(\frac{\partial S}{\partial V}\right)_T = \frac{4}{3}\frac{u}{T} . \tag{23}$$

The partial differentiation of Equations (22) and (23) results from the reciprocity relations in the following:

$$\frac{\partial^2 S}{\partial T\ \partial V} = \frac{1}{T}\ \frac{du}{dT} = \frac{4}{3}\left[\frac{T\dfrac{du}{dT} - u}{T^2}\right] = \frac{4}{3T}\ \frac{du}{dT} - \frac{4}{3}\ \frac{u}{T^2} . \tag{24}$$

Combining both sides of Equation (24) gives

$$\frac{du}{dT} = \frac{4u}{T}$$

or

$$\frac{du}{u} = 4\ \frac{dT}{T} . \tag{25}$$

Integration results in

$$u = aT^4 \tag{26}$$

where "a" is the conventional radiation constant.

Hence, from Equation (7)

$$p = \frac{u}{3} = \frac{aT^4}{3} . \tag{27}$$

Similarly, from Equation (5)

$$U = uV = aT^4V . \tag{28}$$

Substituting Equation (28) into Equation (9) for constant energy per unit volume, u,

$$dU = udV = aT^4\, dV\ . \tag{29}$$

Therefore, from Equations (17), (27), and (29)

$$dS = \frac{dU + pdV}{T} = \frac{aT^4 dV + \frac{aT^4}{3}\, dV}{T} \tag{30}$$

or

$$dS = \frac{4}{3}\, aT^3 dV\ . \tag{31}$$

The integration from zero of Equation (31) results in

$$S = \frac{4}{3}\, aT^3 V = \frac{4}{3}\, \frac{aT^4}{T}\, V \tag{32}$$

which, upon substituting Equation (28), becomes

$$S = \frac{4}{3}\, \frac{U}{T}\ , \tag{33}$$

which is identical to Equation (14).

Equation (26) can be derived from a different approach with identical results, as suggested in Chapter V of Reference 22.

C. AVAILABILITY OF SOLAR RADIATION

Although the sun is not a true blackbody radiator, there is consensus in the literature that this is a sufficiently close approximation that it will be considered so in this report. The derivation of the availability of direct solar radiation after it has been scattered into a state of random direction will first be shown. This will result in the expression

$$1 - \frac{4}{3}\,\frac{T_o}{T} + \frac{1}{3}\left(\frac{T_o}{T}\right)^4$$

for the maximum ratio of availability to total energy of this phenomenon. This ratio is occasionally referred to as maximum conversion "efficiency" [24]. However, to avoid confusion with what is conventionally understood as efficiency, this term will not be used in this report.

The availability that results from the highly directional characteristic of direct sunlight will then be derived, and from this we will see that its maximum ratio of availability to total energy is

$$1 - \frac{2}{\sin\theta}\left(\frac{4}{3}\right)^{\frac{1}{2}}\left(\frac{T_o}{T}\right)^2 + \frac{4}{3}\,\frac{1}{\sin^2\theta}\left(\frac{T_o}{T}\right)^4 ,$$

where θ is the half angle of the cone subtended by the solar disk. Although directional and highly ordered, direct solar energy nonetheless has an associated entropy. However, the latter condition has an inherently higher availability and is specifically relevant to point-focusing parabolic collectors, since they are designed to deliver an ordered beam of sunlight to a focal point. The influence of the reflecting surface on the availability of the solar energy will be covered in Section V.

The availability of solar radiation can be derived directly from Equation (1a), whether the radiation is directional or scattered. Derivations of availability for unpolarized uniform radiation from a black source where the radiation is propagated within a solid angle of 2π have been offered by Petela [7], Press [25], and Spanner [21]. Press has specialized his derivation to the sun, sky, and ambient surroundings as a blackbody source; Petela's and Spanner's derivations are generalized. If we assume that the radiation dead state is isotropic at temperature T_o, then

$$B_o = 0$$

and Equation (1a) becomes Equation (1d) or

$$B = U + p_oV - T_oS. \tag{34}$$

If we further assume that the radiation is contained within a constant volume, we can write Equation (34) in terms of the more convenient form of availability density, or

$$b = u + p_o - \frac{T_oS}{V} . \tag{35}$$

After substituting Equations (27), (28), and (33) into Equation (35), we have

$$b = aT^4 + \frac{aT_o^{\,4}}{3} - T_o \frac{4}{3} \frac{aT^4}{T}$$

or

$$b = aT^4 \left[1 - \frac{4}{3} \frac{T_o}{T} + \frac{1}{3} \left(\frac{T_o}{T} \right)^4 \right] . \tag{36}$$

Applying the relationship between energy density and energy flux derived by Planck [20] to availability, one can then develop the availability density into an availability flux, which is analogous to heat flux. The energy flux is the energy density multiplied by a constant, and this relationship is written in the following form:

$$q'' = \frac{c}{4} \frac{U}{V} = \frac{c}{4} u \tag{37}$$

where

q'' = energy flux

c = radiation propagation velocity

= 2.998×10^{10} cm/sec.

If both sides of Equation (36) are multiplied by $c/4$, we now have an expression for availability flux as

$$b^* = \frac{ca}{4} T^4 \left[1 - \frac{4}{3} \frac{T_o}{T} + \frac{1}{3} \left(\frac{T_o}{T} \right)^4 \right] \tag{38}$$

where, as seen for Equation (26), "a" is the conventional radiation constant, having the value 7.561×10^{-15} erg/cm^{3}K^4. When the factor $ca/4$ is evaluated, it is found to be 5.667×10^{-12} W/cm^{2}K^4, which is recognized as the familiar Stefan-Boltzmann constant that is conventionally represented as σ. The parameter b^* is used to discriminate from b after the multiplication.

Equation (38) can be rewritten as

$$b^* = \sigma T^4 \left[1 - \frac{4}{3} \frac{T_o}{T} + \frac{1}{3} \left(\frac{T_o}{T} \right)^4 \right] \tag{39}$$

where the coefficient σT^4 is the familiar expression for radiation heat flux from a blackbody source. The ratio $b^*/\sigma T^4$, which is the ratio of the availability to the total energy, could be interpreted as the fraction of maximum usable energy, since this fraction represents the maximum useful energy that can be derived from non-polarized radiation propagating within a solid angle of 2π from a black source. Therefore, the expression in the brackets represents the maximum theoretical ratio of availability to total energy for such an energy transfer and can be rewritten as

$$\eta = 1 - \frac{4}{3}\frac{T_o}{T} + \frac{1}{3}\left(\frac{T_o}{T}\right)^4 . \qquad (40)$$

As an example, if T is taken as 5800°K, the temperature of the surface of the sun, and T_o is assumed to be 300°K, the environmental temperature at the surface of the earth, substitution into Equation (40) results in

$$\eta = 1 - \frac{4}{3}\left(\frac{300}{5800}\right) + \frac{1}{3}\left(\frac{300}{5800}\right)^4$$

or

$$\eta = 0.931.$$

In other words, only 93.1% of the black solar radiation is available for use as it arrives at earth.

The derivation of availability for directional or directed sunlight proceeds somewhat differently and is the subject of considerable controversy [24,25,26,27]. As one practical consideration of this study, which will become apparent shortly, we would like to express availability as a function of the cone angle and "temperature" of the solar image.

Although not specifically measured, radiation has an associated temperature that can be derived from parameters such as physical constants, wavelengths, and intensities [20,21]. Whenever this connotation of temperature is referenced in this paper, it will be expressed as "temperature" in quotes.

As seen from earth, the sun is a finite body that forms a cone with a half angle of approximately 0.005 radians with the apex at the earth's surface. For our purposes where we are attempting to enhance availability by concentrating solar radiation, the degree of concentration possible is limited to the cone angle and "temperature" initially available. A reflected solar cone angle from a real mirror surface will always be greater than the initial solar cone angle, resulting in a "temperature" of the solar image that is always less than that of the sun.

The derivation of an equation for the availability of directed solar energy and the interpretation of such an equation have been attempted by several authors with varying degrees of acceptance. Parrott [24] offered a very strong argument expressing the ratio of availability to total energy in the form of

$$1 - \frac{4}{3}\,\frac{T_o}{T}\,(1 - \cos\theta)^{1/4} + \frac{1}{3}\left(\frac{T_o}{T_s}\right)^4 ,$$

which is even referenced in the text "Principles of Solar Engineering" by Kreith and Kreider [16]. However, a computational error acknowledged by Parrott in Reference 26 exists in the derivation of this relation. The expression for the corrected variation that he presents in Reference 26 is

$$1 - \frac{4}{3}\,\frac{T_o}{T} + \frac{T_o^{\,4}}{3T_s^4\,(1 - \cos\theta)} . \tag{41}$$

This is still bothersome, because if θ is small, we can make the approximation

$$1 - \cos\theta \cong \frac{\theta^2}{2} ,$$

substitute it into Equation (41), and obtain

$$1 - \frac{4}{3}\,\frac{T_o}{T_s} + \frac{2\,T_o^{\,4}}{3\,T_s^{\,4}\,\theta^2}$$

as the ratio of availability to total energy. For small θ, however, the third term dominates over the second term and results in an availability exceeding the total energy, which is not possible.

The presentation of the availability of a directed beam of solar energy developed by Byrd, Adler, and Coulter [27] results in an equation that relates the availability with the image cone half angle and "temperature" which is in agreement with test experience gained at the JPL Parabolic Dish Test Site. To expand on the efforts of Byrd, et al, the availability in a cylindrical beam of solar radiation is derived as the maximum work that can be done when expanding to the dead state, less the work done against the surrounding radiation. The entropy within the beam is assumed constant, which is consistent with the derivation of Equation (33), and the expansion is isentropic. The authors also note and take advantage of Pomraning's observation [23] that directed radiation pressure is equal to the energy density, u, which is greater than the radiation pressure in an enclosed volume as indicated by Equation (7). In other words, for a directed beam of radiation

$$p = u. \tag{42}$$

The availability of the initial beam of solar radiation can be represented by

$$B = \int_{V_i}^{V_f} p dV - \int_{V_i}^{V_f} p_o dV , \tag{43}$$

where "i" and "f" refer to the initial and final states, respectively. From Equation (3a) and (4) for the First Law for a closed system

$$\begin{aligned} dU &= \delta W + \delta Q \\ &= -pdV + TdS. \end{aligned}$$

Since the expansion process is isentropic,

$$dS = 0 .$$

Therefore,

$$dU = -pdV . \tag{44}$$

From Equation (5)

$$U = uV$$

from which

$$dU = udV + Vdu . \tag{9}$$

Equating Equations (9) and (44), we have

$$udV + Vdu = -pdV.$$

Substitution of Equation (42) results in

$$udV + Vdu = -udV$$

$$Vdu = -2udV$$

$$\frac{du}{u} = -2\frac{dV}{V} .$$

Integrating,

$$\ln u = -2 \ln V + \text{constant}$$

or

$$\ln uV^2 = \text{constant}$$

which results in

$$uV^2 = \text{constant} = C_1 \tag{45}$$

or with Equation (42)

$$pV^2 = C_1 . \tag{46}$$

Substituting Equations (46) and (7) into Equation (43) results in

$$B = \int_{V_i}^{V_f} \frac{C_1 dV}{V^2} - \int_{V_i}^{V_f} \frac{u_o}{3} dV$$

$$= -\frac{C_1}{V}\Bigg|_{V_i}^{V_F} - \frac{u_o}{3} V \Bigg|_{V_i}^{V_F}$$

$$= -C_1 \left[\frac{1}{V_f} - \frac{1}{V_i}\right] - \frac{u_o}{3} (V_f - V_i) . \tag{47}$$

C_1 can be set equal to unity without loss of generality. Therefore,

$$B = -\frac{1}{V_f} + \frac{1}{V_i} - \frac{u_o}{3} (V_f - V_i) . \tag{48}$$

From Equation (45)

$$u_i V_i^2 = u_f V_f^2 \tag{49}$$

or

$$V_f = V_i \left(\frac{u_i}{u_f} \right)^{\frac{1}{2}} \quad . \tag{50}$$

From Equation (42) the final or dead state for the energy density of the directed beam, u_f, is equal to p_f, which in turn is equal to p_o. From Equation (7) p_o is equal to $u_o/3$. Therefore, u_f is equal to $u_o/3$. Hence,

$$B = - \frac{1}{V_i} \left(\frac{u_f}{u_i} \right)^{\frac{1}{2}} + \frac{1}{V_i} - u_f \left[V_i \left(\frac{u_i}{u_f} \right)^{\frac{1}{2}} - V_i \right]$$

$$= - \frac{1}{u_i V_i} (u_f u_i)^{\frac{1}{2}} + \frac{1}{V_i} - V_i (u_f u_i)^{\frac{1}{2}} + u_f V_i$$

$$= - \frac{V_i (u_f u_i)^{\frac{1}{2}} + u_i V_i}{u_i V_i^2} - V_i (u_f u_i)^{\frac{1}{2}} + u_f V_i \quad . \tag{51}$$

From Equation (45) with C_1 equal to unity, we have

$$u_i V_i^2 = 1.$$

Therefore,

$$B = - V_i (u_f u_i)^{\frac{1}{2}} + u_i V_i - V_i (u_f u_i)^{\frac{1}{2}} + u_f V_i$$

$$= u_i V_i - 2 u_i V_i \left(\frac{u_f}{u_i} \right)^{\frac{1}{2}} + u_f V_i \quad . \tag{52}$$

Dividing through by V_i, we have

$$b = \frac{B}{V_i} = u_i - 2\, u_i \left(\frac{u_f}{u_i}\right)^{\frac{1}{2}} + u_f$$

or

$$b = u_i \left[1 - 2\left(\frac{u_f}{u_i}\right)^{\frac{1}{2}} + \frac{u_f}{u_i} \right] . \tag{53}$$

Equation (53) is the general equation for the availability per unit volume for a directed beam of solar radiation in terms of the ratio of the energy densities.

Byrd, et al [27] modeled a spherical black-body radiation source of radius ρ and temperature T. Its center was located a distance R from a reference point to which energy was beamed through a cone of half angle θ. With $\sin\theta$ set as ρ/R, the energy density and radiation pressure at the reference point were found to be, respectively,

$$u = \frac{3}{8} \frac{\sin\theta \, \ln\,(1 + 2 \sin\theta)}{1 + 2\cos\theta} aT^4 \tag{54}$$

and

$$p = \frac{1}{4}\left[\frac{1}{4} \sin\theta \, \ln\,(1 + 2\sin\theta) + \frac{1}{2}\sin^2\theta - \sin^4\theta\right] aT^4 . \tag{55}$$

For a sun-earth system, θ is very small, which in turn implies that $\sin\theta$ is very small. If we assume the approximations for small x that

$$\cos x \cong 1$$

$$\ln\,(1 + x) \cong x$$

$$\sin^4 x \ll \sin^2 x ,$$

then Equations (54) and (55) can be simplifed for small angles as follows:

$$u = \frac{3}{8} \frac{\sin\theta \, \ln\,(1 + 2\sin\theta)}{1 + 2\cos\theta} aT^4$$

$$= \frac{3}{8} \frac{\sin\theta\,(2\sin\theta)}{3} aT^4$$

or

$$u = \frac{1}{4} \sin^2\theta \; aT^4 \tag{56}$$

and

$$p = \frac{1}{4} \left[\frac{1}{4} \sin\theta \ln (1 + 2 \sin\theta) + \frac{1}{2} \sin^2\theta - \sin^4\theta \right] aT^4$$

$$= \frac{1}{4} \left[\frac{1}{4} \sin\theta \, (2 \sin\theta) + \frac{1}{2} \sin^2\theta \right] aT^4$$

or

$$p = \frac{1}{4} \sin^2\theta \; aT^4 . \tag{57}$$

Observation of Equations (56) and (57) confirms Equation (42). Recalling the earlier statement that

$$p_f = u_f = p_o = u_o/3,$$

we can now derive an expression for the ratio of the energy densities as a function of the cone half angle and source temperature.

From Equation (26), for the dead state we have

$$u_f = \frac{1}{3} u_o = \frac{1}{3} aT_o^4 . \tag{58}$$

Therefore, from Equations (56), (57), and (58) we obtain

$$\frac{u_f}{u_i} = \frac{\frac{1}{3} u_o}{u_i}$$

$$= \frac{\frac{1}{3} aT_o^4}{\frac{1}{4} \sin^2\theta \; aT^4}$$

or

$$\frac{u_f}{u_i} = \frac{4}{3} \frac{1}{\sin^2\theta} \left(\frac{T_o}{T}\right)^4 . \tag{59}$$

By substituting Equation (59) into Equation (53), we have

$$b = u_i \left[1 - \frac{2}{\sin\theta} \left(\frac{4}{3}\right)^{\frac{1}{2}} \left(\frac{T_o}{T}\right)^2 + \frac{4}{3} \frac{1}{\sin^2\theta} \left(\frac{T_o}{T}\right)^4 \right] . \quad (60)$$

Expressed as the ratio of the availability to total energy, a form equivalent to Equation (40) results, or

$$\eta' = 1 - \frac{2}{\sin\theta} \left(\frac{4}{3}\right)^{\frac{1}{2}} \left(\frac{T_o}{T}\right)^2 + \frac{4}{3} \frac{1}{\sin^2\theta} \left(\frac{T_o}{T}\right)^4 . \quad (61)$$

Equation (40) is the ratio of the availability to total energy for a uniformly radiating black body of temperature T, while Equation (61) is that for a directionally radiating black body of temperature T and cone half angle θ through which the energy is beamed.

SECTION III

RECEIVER TECHNOLOGY

A. STATE OF THE ART OF RECEIVERS

Although there has been considerable experimental activity around the world on receivers of both laboratory and field-size scale [12,28,29], only recently has there been a directed effort in the United States to develop solar receivers of a design that could ultimately lead to mass production and commercialization [30,31,32,33,36]. At the time this paper was written, most of the cavity receivers in the references cited were under development in support of organic-Rankine and air-Brayton thermodynamic cycles. In addition, an extensive program was underway at United Stirling of Sweden to develop a cavity receiver in support of a Stirling thermodynamic cycle. Typical design characteristics of these receivers are summarized in Table 3-1.

B. ORGANIC RANKINE RECEIVER

A receiver designed for the organic Rankine thermodynamic cycle was developed by Ford Aerospace and Communications Corporation, Newport Beach, California [32]. This receiver was to supply vaporized toluene at approximately 750°F to a nominal 20 kW_e power conversion unit. Laboratory tests began on this receiver in February 1981 wherein both sub- and super-critical pressures were investigated over a thermal output range of 25 to 100 kW_t. After completion of the receiver qualification tests and its integration with the power conversion unit, the entire assembly was shipped to the JPL Parabolic Dish Test Site at Edwards, California, where in January 1982 it was assembled onto an 11-m diameter concentrator for solar tests. The test program was completed in March 1982, and highlights of some of the test data [34] are presented in Table 3-2.

C. AIR BRAYTON RECEIVER

Two receivers designed to support an air-Brayton thermodynamic cycle have been developed and tested at the JPL Parabolic Dish Test Site. One was built by the Garrett AiResearch Corporation, Torrance, California [30], and the other was fabricated by Sanders Associates, Nashua, New Hampshire [33]. The Garrett unit was a metallic plate-fin, open-cycle configuration designed to heat air to approximately 1500°F from a 85 kW_t solar thermal source. The Sanders assembly, on the other hand, employed a sealed quartz window to allow the receiver cavity to be pressurized to approximately 2 atm wherein the solar flux heated a beta silicon-carbide honeycomb matrix that acted as the heat-exchange surface. Air exit temperatures as high as 2600°F were obtained during testing. Typical test results from both receivers [33, 35] are also shown in Table 3-2.

D. STEAM RANKINE RECEIVER

In addition to the units described above, Garrett AiResearch has also developed a receiver to generate steam from a nominal 85 kW_t solar thermal source [36]. This receiver, which was successfully tested up to 1000 psia and 1300°F at the JPL Parabolic Dish Test Site, could find commercial application as a source for industrial process heat, such as in the application of solar energy for the development of fuels and chemicals, in addition to providing the condensible working fluid for a Rankine thermodynamic cycle.

E. STIRLING RECEIVER

There was considerable test activity in early 1982 by United Stirling of Sweden at the JPL Parabolic Dish Test Site on a receiver designed for adaptation to a Stirling engine. The basic configuration was a tube bundle designed for maximum heat transfer area, maximum internal gas film coefficient, minimum internal volume, and minimum tube thermal stresses, similar to the Stirling heater-head design adopted for their P-40 engine automotive application. Other than to say that the test program met the objectives, specific details of the performance results are proprietary to United Stirling and are not available. The receiver has since been mated to a Stirling engine modified for solar applications and has undergone extensive tests at the JPL Parabolic Dish Test Site.

Table 3-1. Summary of Solar Receiver Design Characteristics of Select Manufacturers

	Ford	AiResearch	AiResearch	Sanders Associates	Fairchild[b]
Working fluid	Toluene	Steam	Air	Air	Helium
Nominal rating, kW_t	95	85	85	-	-
Fluid outlet temperature, °F	750	1300	1500	2500	1500
Fluid inlet temperature, °F	400	300	1050	-	-
Flowrate, lb_m/h	1204	157	0.61	-	-
Aperture diameter, in.	15	9	10	7.75	11
Maximum fluid pressure, psia	790	2000	38	100	2000
Efficiency, %[a]	70 to 90	80 to 92	70 to 80	90	85
Application	Rankine Cycle	Rankine Cycle	Brayton Cycle	Brayton Cycle	Stirling Cycle

[a]First-Law efficiency, temperature-dependent

[b]Hybrid design with a combination solar and fossil-fuel energy source [29]

Note: psia x 6894.76 = Pa

lb_m/h x 0.4536 = kg/h

in. x 0.0254 = m

°C = (°F-32)/1.8

Table 3-2. Typical Data from Receivers Tested at PDTS[a]

	Ford	AiResearch	Sanders
Test Date	3 March 82	7 May 81	16 December 80
Working fluid	Toluene	Air	Air
Insolation, W/m^2	984.0	953.6	960.1
Temperatures along cavity walls, °F	789.0 776.2 677.4 607.4 588.2 393.8	1584 1423 1683 1738 1735 1598 1584 1603 1578	(not reported)
Average cavity wall temperature, °F (standard deviation)	638.7 (145.9)	1614 (96.65)	(not reported)
Working fluid inlet temperature, °F	378.4	1209.2	1123.0
Working fluid outlet temperature, °F	750.4	1513.4	1876.0
Average working fluid temperature, °F	564.4	1361.3	1499.5
Working fluid flow-rate, lb_m/h	780.0	2174.4	730.8
Working fluid exit pressure, psia	494.4	35.8	27.6

Table 3-2. Typical Data from Receivers Tested at PDTS[a] (Cont'd)

	Ford	AiResearch	Sanders
Working fluid inlet enthalpy, Btu/lb_m	-22.28	414.3	391.2
Working fluid exit enthalpy, Btu/lb_m	275.99	497.4	599.0
Working fluid inlet specific entropy, Btu/lb_m°F	-0.089	0.2819	0.2677
Working fluid exit specific entropy, Btu/lb_m°F	0.202	0.3275	0.3748
First-Law Efficiency, %	95.27[b]	75.4[c]	72.0[d]

a – Parabolic Dish Test Site, Edwards, California

b – Calculated by manufacturer from the data

c – Strong gusty winds on day of test

d – Corrected for additional shading of insulated aperture plate

Note: psia x 6894.76 = Pa

lb_m/h x 0.4536 = kg/h

in. x 0.0254 = m

°C = (°F-32)/1.8

Btu/lb_m x 2326.72 = J/kg

Btu/lb_m°F x 4186.8 = J/kg°K

SECTION IV

DEFINITION OF SECOND-LAW EFFICIENCY

Before examining a specific expression for the Second-Law efficiency for solar-thermal cavity receivers, it would be beneficial to discuss the meaning of the term and how the meaning has varied and evolved with different authors. Several examples offered in the literature are presented below.

A. EVOLUTION OF THE CONCEPT

Keenan [6], one of the pioneers in the definition and application of availability, referred to Second-Law efficiency "for want of a better name" as "effectiveness", which he symbolized as ε and defined as work per decrease in availability. He presented an excellent example of how both availability and effectiveness could be applied to the various components of a steam power plant. He later [37] went on to devise "performance coefficients," that were specialized for several classes of cases, while having the same general definition.

Meyer, et al [38], used an approach to Second-Law efficiency similar to the one used by Keenan but referred to it as "thermal efficiency" in reference to a complete power plant. Their thermal efficiency is defined as the maximum possible thermal efficiency (i.e., the thermal efficiency of the heat added) less the summation of the availability losses in various parts of the plant and less any availability rejected, all divided by the heat added. The potential confusion with the conventional definition of thermal efficiency is obvious.

Obert [39], another original contributor in this field of thermodynamics, sought a definition that would relate how closely the true reversibility of a system could actually be approached as being the true test of how efficiently a system is operating. This thought grasps the fundamental essence of Second-Law efficiency, the desire to devise a system that is completely reversible and the ability to quantify its limitations. Following the nomenclature of Keenan, Obert identified this criterion of performance, symbolized it as ε, and defined it as follows:

$$\varepsilon = \left| \frac{\text{Increase in available energy}}{\text{Decrease in available energy}} \right| . \qquad (62)$$

This same definition following the same premise was later used by Gaggioli [15].

Following Obert and Gaggioli, Reistad [17] also used the term effectiveness, symbolized by ε, for the Second-Law efficiency, and expanded Obert's definition into the form

$$\varepsilon = 1 - \frac{\text{Irreversibility}}{\text{Availability decrease}} , \qquad (63)$$

which is similar to an expression derived by Kotas.

Kotas [9] referred to Second-Law efficiency as "rational efficiency" and denoted it as ψ. He observed that, for steady conditions, availability transfers for a given process can generally be grouped into those units representing the desired output and those representing the necessary input. When defining rational efficiency, for which he used the term exergy (symbolized as $\dot{E}$) in place of availability, Kotas accounted for the availability input and output in relation to a control surface enclosing all irreversibilities, $\dot{I}$, related to the process, and arrived at

$$\sum \Delta\dot{E}_{in} = \sum \Delta\dot{E}_{out} + \dot{I} \quad . \tag{64}$$

The rational efficiency was then expressed as

$$\psi = \frac{\sum \Delta\dot{E}_{out}}{\sum \Delta\dot{E}_{in}} = 1 - \frac{\dot{I}}{\sum \Delta\dot{E}_{in}} \quad . \tag{65}$$

Kreider [40] actually used the expression "Second-Law efficiency" and identified it as η_2. With an intent similar to that expressed by Obert, Kreider defined Second-Law efficiency as the ratio of the maximum amount of available energy required to perform a task to the available energy actually consumed by use of a given system. This definition was modified slightly by Kreith and Kreider [16] to be taken as "the ratio of the minimum available energy consumed in performing the task."

Ford, et al [41], used the term "Second-Law efficiency", but following Keenan, symbolized it as ϵ. The intention of Ford was to develop a more generalized meaning that would be associated with the expression "Second-Law efficiency" than is given "effectiveness" as used by Keenan. The purpose of the new expression was to be a measure of the actual performance of a process relative to the optimal performance as limited by the laws of thermodynamics. For a system whose output is the transfer of either useful work or heat, Ford defined Second-Law efficiency as the ratio of the heat or work usefully transferred by a given device or system, B_{Min}, to the maximum possible heat or work usefully transferrable for the same function by any device or system using the same energy input as the given device or system, B_{Actual}. This definition is represented in equation form as

$$\epsilon = \frac{B_{Min}}{B_{Actual}} \quad . \tag{66}$$

Simply stated, the Second-Law efficiency as defined by this equation is the ratio of the least availability that could have done the job to the actual availability used to do the job.

Adler, et al [42], modified Ford's definition slightly, while maintaining the same equation. They defined B_{Min} as the minimum availability needed to perform a task and B_{Actual} as the availability in the energy source actually used. They went on to clarify the definition by presenting an example of the expression for Second-Law efficiency for a solar water heater, relevant to the direction we are ultimately heading. B_{Actual} was taken as the product of the insolation, I, and the first two terms of Equation (40). The third term in this equation was dropped as being small. B_{Min} was taken as

$$B_{Min} = Q\left(1 - \frac{T_o}{T_{water\ tank}}\right) \tag{67}$$

where Q was assumed to be the product of the collector efficiency and the insolation, or

$$Q = \eta I. \tag{68}$$

The resulting equation for Second-Law efficiency then becomes

$$\varepsilon = \eta\,\frac{\left(1 - \dfrac{T_o}{T_{tank}}\right)}{\left(1 - \dfrac{4}{3}\dfrac{T_o}{T_{solar}}\right)}\ , \tag{69}$$

the general form of which has appeared in many of the above cited references [e.g., 16, 21, 40].

Petit and Gaggioli [19] suggested that Second-Law efficiency is the true efficiency as it indicates the degree to which availability contained in any commodity can be completely transferred to any other commodity with the theoretical limit being 100%. They identified Second-Law efficiency as η_{II} and defined it as

$$\eta_{II} = \frac{\text{Available energy in useful products}}{\text{Available energy supplied in "fuels"}}\ . \tag{70}$$

This general form of the Second-Law efficiency equation, enhanced by the definition proposed by the authors cited, provides the foundation for the definition of Second-Law efficiency for solar receivers used in this report.

The definition of Second-Law efficiency for cavity receivers as referred to in this report is the ratio of the availability gained by a working fluid to the availability supplied through the receiver aperture. The availability supplied is the available radiation energy entering the cavity and is equivalent to the sum of the availability gained by the working fluid, the availability destroyed in the process, and the availability lost. We, too, will adopt the symbol η_{II} for Second-Law efficiency. Hence, in equation form, the Second-Law efficiency for a cavity receiver can be expressed as

$$\eta_{II} = \frac{\text{Availability gained by working fluid}}{\text{Available radiation energy entering the cavity}} \tag{71}$$

which stands as our definition.

B. RAMIFICATIONS OF THE SECOND-LAW EFFICIENCY

As indicated earlier, researchers who have seriously applied the Second Law to the determination of system performances generally agree that the Second-Law efficiency is the only true efficiency [e.g., 19, 41]. In all cases, its theoretical upper limit is 100%. Depending upon the system and its operating temperatures, the First-Law efficiency may be less than, equal to, or even exceed 100%, as in the case of a heat pump where the First-Law efficiency is typically referred to as Coefficient of Performance.

The Second-Law efficiency is especially useful for identifying how well the components within a system are matched. If there is room for improvement, it identifies where the improvement should be directed. Condensers represent an example of where the Second-Law efficiency would probably show very little room for improvement. Although, typically, large quantities of energy are exchanged from condensers, the quality of this energy is often so poor that its availability is very low.

Poor use of high-quality energy results in low Second-Law efficiency. The classic example of this is the gas-fired furnace used for space heating. Its First-Law efficiency may be 60 to 70%, but its Second-Law efficiency is typically less than 10%! The results between First- and Second-Law efficiencies can differ quite dramatically.

Maximizing the Second-Law efficiency for non-solar applications is equivalent to minimizing fuel consumption, and hence the recurring cost. For cases involving no consumption of fuel, as would be for solar energy, maximizing Second-Law efficiency should reduce capital cost because the system hardware can be optimized to use more of the energy it takes in.

SECTION V

SECOND-LAW EFFICIENCY FOR RECEIVERS

The operation of any system that collects solar energy is thermodynamically irreversible from three difference aspects [22]: The sun-to-receiver energy exchange, the receiver-to-ambient heat loss, and the internal receiver irreversibilities. Hence, entropy is generated and availability is lost or destroyed upstream, downstream, and inside the receiver. The objective of a good design is to minimize this destruction process.

A. RECEIVER SYSTEM

One of the more common sources of confusion in definitions of First-Law efficiencies for receivers stems from the lack of pre-established system boundaries within which the information is referenced. The system definition that will be followed in this report is depicted in Figure 5-1. The First-Law efficiency, as referenced herein, will be taken as the total energy absorbed by the working fluid per the net radiation energy entering the receiver aperture. Corrections for concentrator effects and intercept factor take place outside the system. The focal region itself is outside the system and transmits its energy across the system boundary and into the cavity.

B. MODEL FOR FOCAL REGION

In this study the model that we will use for the focal region is based on the assumption that the focal region is a "virtual" solar source that behaves like a "fireball" that radiates uniformly as a black body. The construction of the virtual solar source, along with the determination of its "temperature" and size, is accomplished by focusing directed solar radiation. The receiver aperture will be located at the focal region so that the receiver itself sees this virtual solar source as an omnidirectional blackbody radiator. This model is not rigorously correct, but within the field of view of the receiver aperture it should introduce only small error, especially for concentrators with large rim angles.

The virtual solar source from an ideal concentrator would be identical to that of the sun. However, all real surfaces are imperfect and the reconstructed solar image will be of lower quality than the original source because of the higher entropy of the reflected energy that resulted from the disordering that occurred during the reflection process.

C. AVAILABILITY AT THE FOCAL REGION

Prior to determining the available radiation energy that enters a receiver cavity, it is necessary to quantify the availability in the focal region. The focal region of a parabolic concentrator is defined and treated in this report as a "virtual" solar source wherein the sun's image is reconstructed,

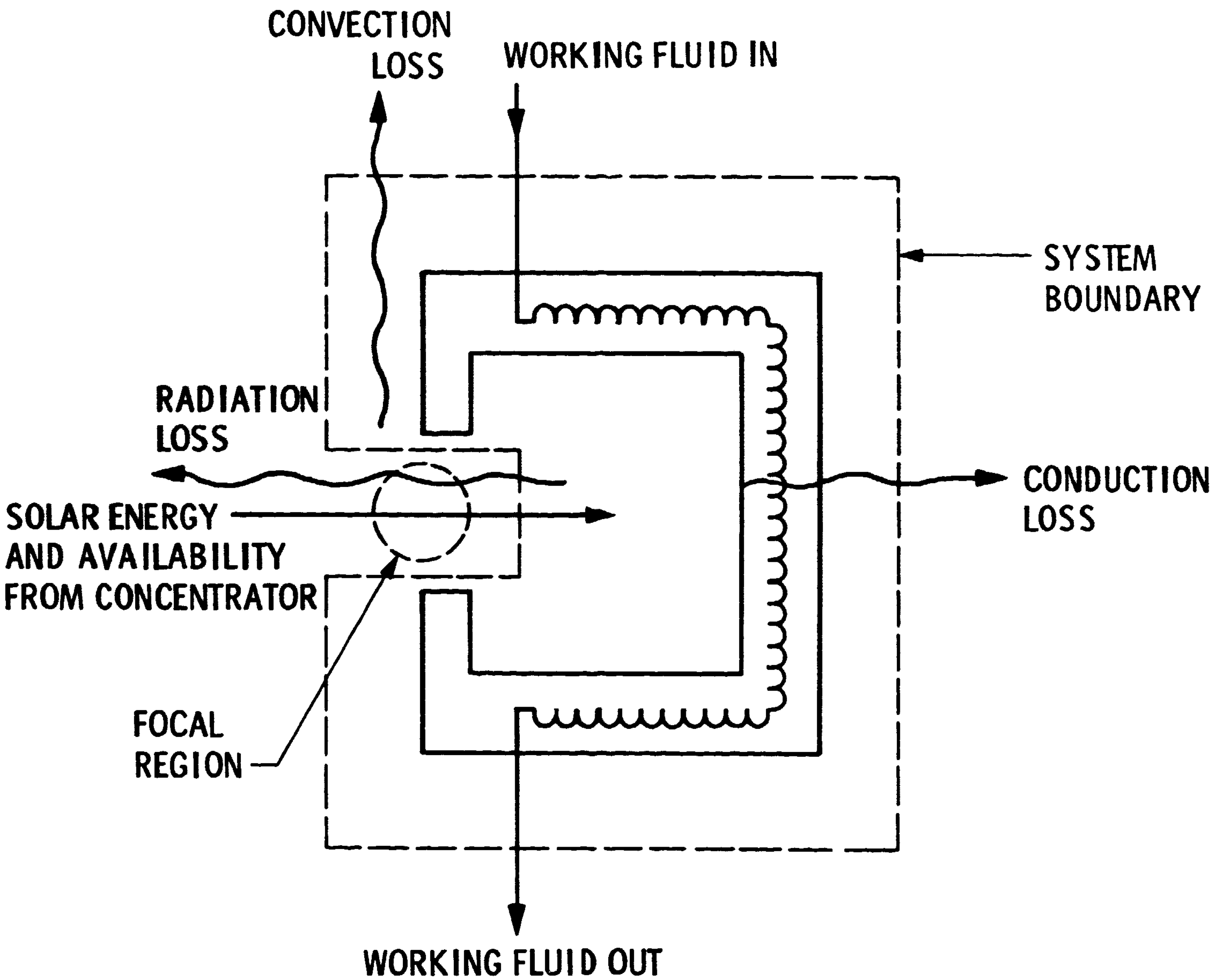

Figure 5-1. Schematic Representation of Receiver System

accounting for disordering caused by the reflective surface. We will follow this model to develop a virtual solar source at the focal region and estimate its "temperature", T_F, which will become the blackbody source temperature seen by the receiver.

As a point of departure for quantifying the availability at the focal region, we will assume that the incident solar radiation (insolation) arriving at the surface of the concentrator is identical to that measured by a pyrheliometer. This is a valid assumption as they both see undisturbed insolation. However, errors could possibly be introduced if they were not in close proximity. Referring now to Equation (61) for directed radiation, we can approximate $\sin\theta$ as θ, since the half angle subtended by the sun is small (approximately 0.005 radian), and rewrite Equation (61) as

$$\eta' = 1 - \frac{2}{\theta}\left(\frac{4}{3}\right)^{\frac{1}{2}} \left(\frac{T_o}{T}\right)^2 + \frac{4}{3\theta^2}\left(\frac{T_o}{T}\right)^4. \tag{72}$$

Equation (72), which is defined as the ratio of the availability to the total energy for directed radiation, is also equivalent to the ratio of the availability flux to the total energy flux, of which the total energy flux is the insolation measured by the pyrheliometer. We will assume for this analysis that this ratio remains constant during the reflection process such that it is equivalent before and after reflection. This assumption is based on the premise that the primary loss of availability resulting from the reflection process is a direct, first-order effect of the energy losses experienced from influences such as imperfect reflectivity, intercept factor, and shadowing and blocking. We will see shortly that the results of this assumption agree well with experience. In order for Equation (72) to remain constant, the two variables, θ and T, must be related such that

$$\theta T^2 = \text{constant}, \tag{73}$$

which can be written in non-dimensional form as

$$\theta\left(\frac{T}{T_o}\right)^2 = \text{constant}. \tag{74}$$

If θ, which is directly affected by the reflection process, represents the reflected cone half angle, then T becomes the "temperature" of the virtual solar source, which we have defined as the focal region.

The half angle of the reflected radiation can be evaluated from the characteristics of the reflective surface. It is beyond the scope of this report to cover the details of solar cone optics, as numerous excellent references exist (see, for example, Wen, et al [1]). However, some mention of the four major influences on the reflected cone half angle is warranted, as they represent the primary sources of imperfection. These factors are the following:

(1) The slope error of the reflective surface, a measure of the optical surface accuracy or the deviation of the surface normal from that of perfect geometry. Causes stem from such sources as surface waviness, fabrication tolerances, structural deflections, and thermal gradients.

(2) The beam non-specularity, a condition of all real surfaces. Surface conditions, incidence angle, and wavelength all influence the reflection characteristics.

(3) The pointing error where the geometric centerline does not coincide with the centerline of the solar image. This can be caused by inaccurate sun tracking with the concentrator or misalignment of the receiver itself.

(4) The sun source itself, caused by the non-uniform radiance emitted over the solar dish and the influence of the atmosphere on the solar beam as it passes through.

An excellent discussion of these influences with supporting equations is given by Wen, et al [1]. All of these factors contribute to increasing the half angle of the reflected solar image, which we will call θ_R. The amount of increase of the solar half angle will be called δ, defined as

$$\delta^2 = 2\sigma_{sl}^2 + \sigma_{sp}^2 + \sigma_{pe}^2 + \sigma_{sun}^2 \qquad (75)$$

where

σ_{sl} = slope error standard deviation

σ_{sp} = non-specularity standard deviation

σ_{pe} = pointing error standard deviation

σ_{sun} = sun source standard deviation.

The reflected half angle now becomes

$$\theta_R = \theta + \delta \,. \qquad (76)$$

If, consistent with test experience, the non-specularity standard deviation of the JPL Test-Bed Concentrator is taken as 3 milliradians and the remaining three standard deviations are assumed to be approximately 1/8 degree, or 2.2 milliradians, then δ is found to be 6.168 milliradians. The sun subtends an angle of approximately 32' at the surface of the earth. This results in a cone half angle of 0.0047 radians. When these values for δ and θ are substituted into Equation (76), θ_R becomes 0.0109 radians.

If the surface of the sun is assumed to be 5800°K (10440°R) and the environmental dead state is taken as 300°K (540°R), then for a θ of 0.0047 radians the constant in Equation (74) is evaluated as 1.75676, or

$$\theta_R \left(\frac{T_F}{T_o}\right)^2 = 1.75676. \tag{77}$$

With θ_R developed in Equation (76) substituted for θ in Equation (74), Equation (77) results. After substituting the value calculated for θ_R into Equation (77), we find the "temperature" of the focal region, or virtual solar source, to be 3809°K (6856°R), symbolized as T_F. Based on actual test data derived from flux mapping experiments conducted on the Test-Bed Concentrator at the JPL Parabolic Dish Test Site [43], the "temperature" of the focal region was calculated by the experimenters to be approximately 3600°K (6480°R). Comparing this with our result, we find that our method predicts a value within 6% of actual experience, thus validating our earlier assumption regarding Equation (72). Our predicting slightly high is probably due to other influences of a nonideal system.

Since the focal region is assumed to be a virtual solar source radiating as a black body, its availability can be developed and evaluated from Equation (39), where the equation is modified by multiplying by the appropriate area to convert from flux units to rate units. b* now becomes the more conventional $\dot{b}$. If the symbol q_F is taken as the energy rate (or power) at the focal region, then following the format established by Equation (39), we can express the availability rate for the virtual solar source as

$$\dot{b}_F = q_F \left[1 - \frac{4}{3}\left(\frac{T_o}{T_F}\right) + \frac{1}{3}\left(\frac{T_o}{T_F}\right)^4\right]. \tag{78}$$

The quantity q_F is evaluated in terms of measured parameters as follows:

$$q_F = IA\rho G\phi \tag{79}$$

where

I = insolation

A = projected area of the concentrator

ρ = reflectivity of the mirrored surface

G = shadowing and blocking factor

ϕ = intercept factor of the solar energy at the receiver aperture.

Equation (78) can now be rewritten as

$$\dot{b}_F = IA\rho G\phi\left[1 - \frac{4}{3}\left(\frac{T_o}{T_F}\right) + \frac{1}{3}\left(\frac{T_o}{T_F}\right)^4\right], \tag{80}$$

and $\dot{b}_F$ is observed to be the available radiation energy entering the receiver cavity -- the denominator of Equation (71).

D. AVAILABILITY GAINED BY THE WORKING FLUID

The development of an expression for the numerator of Equation (71) is much more straightforward. First, it is necessary to know the rate of the net energy absorbed by the receiver cavity, which we will call q_R. Next, we observe the relationship that the availability received by the working fluid is equal to the maximum available energy in the receiver cavity prior to the irreversible heat transfer to the working fluid, less the availability destroyed during the transfer process. This is expressed in equation form as

$$b_W = b_R - b_D \tag{81}$$

where

b_W = availability gained by working fluid

b_R = availability in receiver cavity

b_D = availability destroyed.

To quantify Equation (81) in terms of known parameters, it is necessary to evaluate b_R and b_D. Both of these terms can be expanded from Equations (1f) and (4) for a closed system. With the internal energy expressed as heat, Equation (1f) can be rewritten as

$$B = Q - T_oS\ . \tag{82}$$

If S is taken as Q/T, then Equation (82) becomes

$$B = Q - T_o \frac{Q}{T}$$

or

$$B = Q\left[1 - \frac{T_o}{T}\right]. \tag{83}$$

Equation (83) represents a general form, so we can apply it to the evaluation of b_R and b_D. In terms of energy rate (or power), as q_R has units of energy per unit time, we find for the availability rate in the receiver cavity prior to transfer that

$$\dot{b}_R = q_R\left[1 - \frac{T_o}{T_R}\right] \tag{84}$$

where

T_R = receiver cavity wall temperature.

The rate of availability destroyed in the transfer process is equal to the availability rate prior to transfer less the availability rate after transfer, or

$$\dot{b}_D = q_R\left[1 - \frac{T_o}{T_R}\right] - q_R\left[1 - \frac{T_o}{T_W}\right]$$

or

$$\dot{b}_D = q_R T_o\left[\frac{1}{T_W} - \frac{1}{T_R}\right] \tag{85}$$

where

T_W = working fluid temperature.

The availability rate gain by the working fluid is simply the difference between Equations (84) and (85), or the availability rate after transfer, expressed as

$$\dot{b} = q_R \left[1 - \frac{T_o}{T_W} \right] . \tag{86}$$

Equation (86) is observed to be the numerator of Equation (71).

E. EQUATION FOR SECOND-LAW EFFICIENCY FOR SOLAR CAVITY RECEIVERS

The general definition used in this paper for the Second-Law efficiency of solar cavity receivers is given by Equation (71). The numerator and denominator of this equation are given by Equations (86) and (80), respectively; thus, Equation (71) can be written as

$$\eta_{II} = \frac{q_R \left[1 - \frac{T_o}{T_W} \right]}{IA\rho G\phi \left[1 - \frac{4}{3}\left(\frac{T_o}{T_F}\right) + \frac{1}{3}\left(\frac{T_o}{T_F}\right)^4 \right]} . \tag{87}$$

So far we have not quantified q_R in terms of known parameters. However, it is derived quite simply from the heat balance of the energy received and the energy lost. The net rate of energy absorption in a receiver cavity is the rate of energy entering through the receiver aperture and initially absorbed, less the rate of energy being radiated, convected, and conducted from the receiver. The equations for radiation, convection, and conduction are of the conventional forms found in any text on heat transfer. However, the term for energy absorption takes into account the insolation received, the concentrator area and its reflectivity and shading factor, and the intercept factor and effective absorptivity of the receiver. Experience has shown that radiation and convection from the receiver aperture predominate over all other radiation and convection. Therefore, the aperture area can be taken as the reference area for both, without introducing significant error. The conduction area, however, must be the area of the walls and ends through which the heat is conducted.

When written in equation form, an expression very similar to that offered by Jaffe [44] is obtained.

$$q_R = IA\rho G\phi\alpha_e - A_R \left[\varepsilon_e \sigma (T_R^4 - T_o^4) + h(T_R - T_o) \right] - A_e \frac{k}{L} (T_R - T_o) \tag{88}$$

where

A_R = receiver aperture area

h = convection film coefficient

A_e = effective conduction area

k = thermal conductivity of insulation

L = insulation thickness

α_e = effective absorptivity of cavity

ε_e = effective emissivity of cavity

and all other parameters are as defined earlier. This relatively simplified version of the energy balance generally gives results accurate to within 5 to 10% when compared with experimental data. The effective absorptivities and emissivities are assumed equal for this analysis and are derived as follows [45]:

$$\alpha_e = \varepsilon_e = \frac{\alpha}{1 - \left[1 - \frac{1}{R}\right]\left[1 - \alpha\right]} \tag{89}$$

where

α = absorptivity of receiver surface

R = ratio of the cavity inner surface area to the receiver aperture area.

One can see that if the cavity surface is large compared with the aperture area, then the effective absorptivity and emissivity approaches unity.

In terms of known measured or derived parameters, the Second-Law efficiency for a cavity receiver can be expanded from Equation (87) to become

$$\eta_{II} = \frac{\left\{IA_\rho G\phi\alpha_e - A_R\left[\varepsilon_e\sigma(T_R^4 - T_o^4) + h(T_R - T_o)\right] - A_e\frac{k}{L}(T_R - T_o)\right\}\left[1 - \frac{T_o}{T_W}\right]}{IA_\rho G\phi\left[1 - \frac{4}{3}\left(\frac{T_o}{T_F}\right) + \frac{1}{3}\left(\frac{T_o}{T_F}\right)^4\right]} . \tag{90}$$

Equation (90) represents the complete equation for the Second-Law efficiency for cavity receivers. A simplification to Equation (90) is possible if we note that the ratio of q_R to $IA_aG\Phi$ [i.e., the ratio of Equation (88) to Equation (79)] is the ratio of the net energy absorbed by the receiver to the radiation energy passing through the aperture, which is the First-Law efficiency η_I. Equation (90) can now be written as a function of the First-Law efficiency as

$$\eta_{II} = \eta_I \frac{\left(1 - \frac{T_o}{T_W}\right)}{1 - \frac{4}{3}\left(\frac{T_o}{T_F}\right) + \frac{1}{3}\left(\frac{T_o}{T_F}\right)^4} . \qquad (91)$$

Although Equation (91) represents a valid short form of the equation for the Second-Law efficiency for cavity receivers when the First-Law efficiency and the working-fluid and focal-region temperatures are known, Equation (90) should still be used for system optimizations, because the cavity temperature, T_R, which is implicit in η_I, will vary as T_W and T_F are changed.

As we shall see later, the working fluid temperature, T_W, and the cavity temperature, T_R, can both be estimated within acceptable engineering accuracy as the arithmetic means of their end-point temperatures.

SECTION VI

COMPARISON WITH FIRST-LAW EFFICIENCY

We are now in a position to evaluate the merits of a Second-Law approach to determine the performance of solar-thermal cavity receivers and to compare the First- and Second-Law efficiencies. The Ford and AiResearch receivers have been chosen as the candidate examples that we will present. This choice is for three primary reasons: First, the working fluids for each are quite different -- toluene for the Ford receiver and air for the AiResearch receiver; and second, both receivers are well documented by analysis and test. The third reason, which relates to the efficiencies themselves, will become apparent during the discussion of the results.

In this section we will present the results of the calculations of the availability gain of the working fluid and of the Second-Law efficiency for each receiver, based on the equations derived in the earlier sections, and compare these estimations with actual test data listed in Table 3-2. We will demonstrate that the derived analytical method predicts quite well, independent of working-fluid phase change.

A. RECEIVER AND CONCENTRATOR CHARACTERISTICS

The typical physical characteristics of each of these receivers have been identified in Table 6-1. The geometry data were obtained from the references. The value 2.82 Btu/h-ft^2-°F (16 W/m^2-°K) for the convective film coefficient listed under Ford, the conductivity per unit length of 0.3 Btu/h-ft^2-°F (1.7 W/m^2-°K), and the intercept factor of 0.987 have all been shown by experience to be reasonable values to assume for estimating purposes [44], and can be used with confidence when other information is not available. This convective film coefficient, however, is for a calm day only, and this number would be used if more specific data did not exist.

Convection losses on days with strong gusty winds can be quite significant, and the data for the AiResearch receiver was taken on just such a day. Its convective film coefficient for the day of the test was estimated by the experimenters and was used in the development of Table 6-3. The difference between this result and one that would have been obtained for a calm day is addressed in the discussion of the results.

Since the energy and availability into a receiver cavity are directly affected by the concentrator, as seen in the derivation of the relevant equations, the characteristics of the JPL Test-Bed Concentrator upon which both receivers were tested are given in Table 6-2.

B. RESULTS OF CALCULATIONS

Tables 6-1 and 6-2 were taken as inputs to the relevant equations derived earlier, and Table 6-3 was developed from the results of these calculations. Because receiver cavity temperature and working fluid temperature are so important in determining availability, Table 6-3 was divided into two major columns with these two parameters as the differentiators. The results depicted in the first column are based solely on design temperatures, and a simple numerical mean of the end points was taken as the average temperatures for both the cavity and the working fluid. Data for the working fluid temperatures of both receivers are found in Table 3-1. The design average cavity temperature was extracted from Reference 46 for the Ford receiver and from Reference 30 for the one from AiResearch. In the second column these two temperatures reflect the overall averaging of the actual test data. The intent of this approach was to compare the results of a relatively simple method of estimating these temperatures with what the results might have been if the temperatures were more accurately known. Since detailed information is generally not available during design phases, the desire was that the simpler method would predict the final performance results with sufficient accuracy that it could be used with confidence. A review of the results presented in Table 6-3 verifies that this is indeed the case.

In addition, to make a more meaningful comparison with actual test data, the quantities developed for the parameters in Table 6-3 were all derived from the measured insolation values obtained from Table 3-2. As a result, the indicated powers (energy rates) and availabilities are different for each receiver, even though the same concentrator was used. However, if the actual insolation is not known, the values that are typically assumed for design purposes are 800 or 1000 W/m^2.

C. DISCUSSION OF RESULTS

Upon review of Table 6-3, the first point we wish to note is that the predicted availability gain of the working fluid agrees quite well with that derived from the actual test data in Table 3-2. One would expect that quantities based on the design temperatures would not be as close as those based on actual cavity and working fluid temperatures, but a projected result that falls within 10% lends considerable credibility to the approach. The low prediction errors resulting from applying temperatures derived from the data imply that if more accurate cavity and working fluid temperature information were available during the design phase, then estimates of the availability of the working fluid could be derived well within acceptable engineering tolerances. As can be seen, this additional information would probably have a stronger influence on a cavity design like the Ford receiver, because its temperature profile is more non-linear than that of the AiResearch receiver.

The availability destroyed in the process of transferring heat to the working fluid is not a large factor, but the availability lost, which is implicit in the derivation of the net availability in the receiver cavity, is quite large because of the entropy associated with the power absorbed by the receiver. If we look, for example, at the first column of Ford parameters, we see that the availability at the focal point is 67.6 kW_t, while the net availability in the receiver cavity is only 35.84 kW_t -- a loss of 31.76 kW_t! By comparison, we also see that the power (energy rate) entering the receiver cavity is 75.37 kW_t of which 73.06 kW_t are absorbed, representing a loss of only 2.31 kW_t. The very strong influence of the quality of the energy is demonstrated by this apparent paradox. (A more refined analysis will most likely show that some of the lost availability can also be attributed to availability destruction occurring during other processes.)

As mentioned earlier, the estimates for the AiResearch receiver were developed from prior knowledge of convection losses for a very windy day. If this information had not been available and if the lower convection coefficient suggested in Table 6-1 were used, then based on the design temperatures the availability gain of the working fluid would have been predicted high by 22.3% by not accounting for the wind. If the actual average temperature were used, it would have been predicted high by 20.6%.

Directing our attention now to the comparison of the First- and Second-Law efficiencies, we observe a very interesting development. First, for the record, note that the estimates of the First-Law efficiencies agree very well with those derived from the data, since they fall within approximately two percentage points of each other. The First-Law efficiency for the Ford receiver is very high, while that for the AiResearch receiver may be perceived as disappointingly low, as it falls below the efficiency of the Ford receiver by about 20 percentage points. However, when we look at the Second-Law efficiency we find that not only is the variation between the two receivers less, but the positions have also been interchanged. A very important consequence that we will elaborate upon stems from this finding.

These two examples were specifically chosen because the exchanged positions of the two efficiencies strengthen the argument for using Second-Law efficiency as the preferred performance indicator. The higher Second-Law efficiency of the AiResearch receiver indicates a greater quantity of energy available in the working fluid to perform a useful function. The AiResearch receiver is able to use ten percentage points more of the available solar radiation energy that arrives at the focal region than can the Ford receiver, even though the AiResearch receiver has a lower First-Law efficiency. However, if a selection of these two receivers were to be made for some application based solely on First-Law efficiency, as is typically the case, then without question the selection would be in favor of the one that would deliver the less availability that ultimately would result in the performance of less useful work. It is not sufficient that a receiver merely capture the arriving energy: The essential element is how much of the captured energy is available for use. Knowledge of the quality of this energy is a necessary condition and must also be considered.

In the development of solar thermal receivers, there is some reluctance to design units to operate at very high temperatures because of inherent increase of First-Law radiation losses. However, the focal region of a concentrator is a very high quality energy source, and effort should be made to minimize loss of this quality in its application. The trade-offs should be made in terms of where the maximum availability occurs. Although radiation losses do increase with increasing cavity temperature, the decreasing entropy in the receiver allows the availability also to increase, as illustrated in Table 6-3. First-Law losses alone are not sufficient criteria for system performance optimization. However, because there is also some availability lost with the reradiation, we would expect to observe a temperature where the availability would be maximized.

Table 6-1. Typical Receiver Characteristics for Ford and AiResearch Receivers

	Ford	AiResearch
Working fluid	Toluene	Air
Aperture diameter, inches	15	10
Receiver internal diameter, inches	24	20
Receiver length, inches	20	32
Receiver outer diameter, inches	32	30
Effective conduction area, ft^2	15.36	19.58
Convective film coefficient, Btu/h-ft^2-°F	2.82[a]	55.48[b]
Absorptivity (emissivity) of inner wall	0.95	0.95
Conductivity per unit length, Btu/h-ft-°F	0.3[a]	0.3
Intercept factor (ϕ)	0.978[a]	0.978
Effective absorptivity (emissivity)	0.9953	0.9982

[a] Value is typical for calm days; good for estimating.

[b] Very windy day; quantity determined by experimenters.

Table 6-2. Characteristics of the JPL Test-Bed Concentrator

Reflectivity (ρ)	0.92
Net concentrator area (shading factor included), m^2	84.35
Focal ratio (f/D)	0.6
Concentrator diameter, m	11
Slope error (σ_{sl}),rad	2.2×10^{-3}
Non-specularity (σ_{sp}),rad	3×10^{-3}
Pointing error (σ_{pe}),rad	2.2×10^{-3}
Sun source error (σ_{sun}),rad	2.2×10^{-3}
Rim angle, deg	45.24

Table 6-3. Predicted Second-Law Efficiencies for Two Solar-Thermal Cavity Receivers

	With Design[a] Temperatures		With Temperatures[b] From Data	
	Ford	AiResearch	Ford	AiResearch
Insolation, W/m^2	984.0	953.6	984.0	953.6
Reflected solar half angle, rad	0.0109	0.0109	0.0109	0.0109
Power entering receiver, kW_t	75.37	72.37	75.37	72.37
(Btu/h)	(257230)	(247010)	(257230)	(247010)
Availability[c] at focal point, kW_t	67.60	64.91	67.60	64.91
(Btu/h)	(230720)	(221553)	(230720)	(221553)
Average cavity temperature, °R	1060	1795	1098.67	2074.0
Net power absorbed by receiver, kW_t	73.06	56.14	72.85	53.6
(Btu/h)	(249355)	(191619)	(248636)	(182937)
Net availability in receiver cavity, kW_t	35.84	39.25	35.81	39.64
(Btu/h)	(122325)	(133973)	(122206)	(135306)
Average working fluid temperature, °R	1035	1735	1024.4	1821.3
Availability destroyed, kW_t	0.9	0.58	2.6	1.94
(Btu/h)	(3072)	(1993)	(8860)	(6609)

Table 6-3. Predicted Second-Law Efficiencies for Two Solar-Thermal Cavity Receivers (Cont'd)

	With Design[a] Temperatures		With Temperatures[b] From Data	
	Ford	AiResearch	Ford	AiResearch
Availability gained by working fluid, kW_t	34.94	38.67	33.21	37.7
(Btu/h)	(119254)	(131981)	(113346)	(128670)
Actual availability[b] gain by working fluid, kW_t	32.25	37.24	32.25	37.24
(Btu/h)	(110078)	(127085)	(110078)	(127085)
Prediction error of availability gain, %	8.3	3.84	3.0	1.24
Second-Law efficiency, %	51.7 (47.7 actual)[b]	59.6 (57.4 actual)[b]	49.1 (47.7 actual)[b]	58.4 (57.4 actual)[b]
First-Law efficiency calculated, %	96.9	77.6	96.7	74.1
(From Table 3-2)	(95.27)	(75.4)	(95.27)	(75.4)

[a] The receiver cavity temperatures and the working fluid temperatures are design predictions (see Table 3-1).

[b] The receiver cavity temperatures, the working fluid temperatures, and the actual availability gain by the working fluid were derived from test data (see Table 3-2).

[c] Based on T_F calculated from Equation (77).

Note: A solar cone half angle of 0.0047 radian and a dead state temperature of 540°R were assumed.

As indicated earlier in Section IV, another significant aspect of Second-Law efficiency is that it demonstrates how well the components within a system are matched. The AiResearch receiver with the higher Second-Law efficiency, therefore, is better suited for the JPL Test-Bed Concentrator than is the Ford receiver. In fact, if the windy-day convection losses had not been factored into the calculations for the AiResearch receiver, then the estimate of its Second-Law efficiency based on the design temperatures would have been 70.2%.

These results are completely consistent with the intended application of each receiver. The AiResearch receiver was designed specifically for the Test-Bed Concentrator, while the Ford receiver was designed for integration with a lower quality concentrator to meet requirements for reduced cost and mass production. Its application to the Test-Bed Concentrator was for qualifying tests only, and it was known to be a non-optimal match even before testing began. Had the availability of the arriving radiation been lower for the same receiver conditions, the Second-Law efficiency of the Ford receiver would have been higher. This will undoubtedly be experienced when tests are conducted with the properly matched concentrator.

SECTION VII

SUMMARY AND CONCLUSIONS

In this report we explored the fundamental concepts of availability and entropy, derived an equation for the Second-Law efficiency of solar-thermal cavity receivers, presented a summary of the state of technology for solar receivers, and discussed the comparison between First- and Second-Law efficiencies using an example of two receivers that were designed for different purposes. It was necessary to elaborate on the definitions of availability and the entropy of radiation because the concepts are quite abstract and often not immediately understood, yet insight into their meaning is essential to the development of Second-Law efficiency.

Because radiation is the form of energy transfer to the receiver cavity, added emphasis was given to the derivation of the availability of radiation at the focal region in order to develop a convenient relationship among the reflected cone half angle, the insolation, and the concentrator geometric characteristics. A simple method of independently determining the "temperature" of the focal region as a function of the reflected cone half angle with its implicit reflection errors was first derived (Equation 77), giving the necessary input to Equation (80), which is the resulting expression for the availability of a virtual solar source. The relation for the availability gained by the working fluid, Equation (86), followed more conventionally.

Although, as indicated in Section II, there is still controversy in the literature about the correct expression for directed solar radiation, Equation (77) evolved from the premises used to derive Equation (53) and was found to agree well with experience. Improvements can be made on these relationships as work continues in the future.

An equation for the Second-Law efficiency of solar-thermal cavity receivers (Equation 90) was derived in a form intended for easy use in that all of the required variables are either known or readily determined. A summary of the evolution of the concept and definition of Second-Law efficiency was presented to lend perspective to how the subtlety of the meaning has developed. The two critical variables, the working fluid temperature and the cavity temperature, can be estimated directly as the arithmetic mean of the end-point temperatures for each. Calculations of the working fluid availability gain and of the Second-Law efficiency based on this method should fall within a 10% accuracy. This is an important finding in this analysis because it permits a confident projection of the Second-Law efficiency within acceptable engineering accuracy during the design phases when knowledge of the complex temperature and flux profiles of the receiver have not been established. Design changes can be based on these projections early in the development.

We attempted to demonstrate that a Second-Law approach to quantifying the performance of a solar-thermal receiver lends greater insight into the total picture than is possible from the conventional First-Law approach. We know from conventional First-Law energy balances that not all energy entering the receiver cavity is absorbed and used, but only by exercising the Second Law are we able to determine how much of that arriving energy can be applied to perform a useful function. Energy is often defined as the ability to do work, but because of entropy, not all of the energy is available for the desired function. It is only through the Second Law that the true meaning of the potential to cause change can be realized.

Application of First-Law efficiency alone is subject to several limitations. For instance, decisions based solely on First-Law considerations not only may not be optimal, but may even suggest erroneous directions. This was shown in Section VI through the comparison of two receivers which revealed that the higher radiation losses from the higher cavity temperature of one resulted in its having a lower First-Law efficiency. However, when the lower entropy within the cavity of this receiver was accounted for, greater availability and higher Second-Law efficiency resulted -- different from that suggested by the First Law.

First-Law efficiencies may come out much higher than the state of technology really is, and these higher values may suggest much better component matching than really exists. The state of technology of a thermodynamic component such as a solar-thermal cavity receiver is maximized only when the destruction and loss of its availability is minimized, which occurs, ideally, only when complete reversibility is approached. If we assume that proper component matching will result in the maximum transfer of availability, then, as we saw in Section VI, there is no correlation of component matching with First-Law efficiency.

Knowledge of energy quality is a necessary condition in the total equation of directing energy to cause change. Only from the Second-Law efficiency can one know how well a thermodynamic system approaches its ideal performance.

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