A reference heat source for solar collector thermal testing

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A REFERENCE HEAT SOURCE FOR SOLAR COLLECTOR THERMAL TESTING

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RÉSUMÉ

Des capteurs solaires à liquide ont été soumis à des essais au moyen d'une source chaude de référence (SCR) à comparaison directe. Un radiateur électrique est relié en série au capteur. Le fluide caloporteur absorbe de l'énergie à un taux mesuré par la SCR, et le taux de captage d'énergie solaire est défini par le produit de l'énergie absorbée et le rapport entre les écarts de température dans le capteur et le radiateur. La SCR et les sondes de température associées peuvent être vues comme un indicateur thermique de débit massique, le produit du débit massique et de la chaleur spécifique étant mesuré directement dans la boucle d'essai.

Des essais d'étalonnage ont été menés sur deux SCR à différents débits et aux températures normalement observées dans un essai de capteur plan (de ~10°C à ~95°C). On a établi dans chaque cas les valeurs de η, à savoir le rapport de l'énergie thermique (mesurée) transférée par la SCR au caloporteur à l'énergie électrique consommée par la SCR. Les résultats sont pour la SCR n°1: η_1=99,61% (écart-type de 0,22), et pour la SCR n°2: η_2=99,59% (écart-type de 0,28).

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A Reference Heat Source for Solar Collector Thermal Testing

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ABSTRACT

Liquid-based solar collectors have been tested using a direct-comparison reference heat source (RHS). An electric heater is installed in series with the solar collector under test. Energy is added to the heat transfer fluid at a measured rate by the RHS, and the rate of solar energy collection is determined as the product of this energy input and the ratio of the differential temperatures across the collector and the heater. The RHS and associated temperature sensors can be thought of as a thermal mass flow meter where, in effect, the product of mass flow and specific heat is measured directly in the test loop.

Calibration tests were performed on two reference heat sources at various flowrates and at the temperatures normally encountered in flat plate solar collector testing (≈10°C to ≈95°C). Values of η, the ratio of the measured RHS thermal power output to the fluid to the RHS electrical power input were determined for each case. Results indicate that for RHS#1, η_1 = 99.61% (standard deviation of 0.22) and for RHS#2, η_2 = 99.59% (standard deviation of 0.28).

NOMENCLATURE

A = aperture area of collector (m^2)
C_p = specific heat of heat transfer fluid (J/(kg·K))
G = total solar irradiance on the plane of the solar collector (W/m^2)
$\dot{m}$ = mass flowrate of heat transfer fluid (kg/s)
P_{RHS} = input power to RHS (W)
q_c = rate of energy collection by the solar collector (W)
$\dot{q}_{RHS}$ = output power of the RHS (W)
T_a = temperature of ambient air surrounding the RHS (°C)
T_i = temperature of fluid at inlet to heater in the RHS (°C)
T_o = temperature of fluid at outlet of heater in the RHS (°C)
T_m = mean fluid temperature in the RHS (°C)
η = solar collector efficiency
η_{RHS} = thermal efficiency of the RHS
Δp = pressure drop between temperature wells in the RHS (Pa)
ΔT_c = fluid temperature rise through solar collector (K)
ΔT_{RHS} = fluid temperature rise through reference heat source (K)
τ = measured time interval (s)
M = mass of fluid collected during time interval τ (kg)
V = volume flowrate of heat transfer fluid (m^3/s)
ρ = density of the heat transfer fluid (kg/m^3)

Subscripts

c = related to the collector
RHS = related to the reference heat source

1. INTRODUCTION

The thermal efficiency (η) of a solar collector, as determined by testing, is given as,

$$\eta = q_c / A \cdot G \tag{1}$$

where,

q_c = rate of energy collection by solar collector,
$= (\dot{m} \cdot C_p \cdot \Delta T)_c$

ASHRAE Standard 93-77 [1] outlines a formalized solar collector testing procedure and specifies the accuracy requirements of all the measured terms. In particular, for collectors that use liquid as the heat transfer fluid, the accuracy of the mass flowrate measurement should be equal to or better than ±1% of the measured value. It is normal procedure to measure the volumetric flowrate and derive mass flowrate from published values of fluid density, (i.e., $\dot{m} = V \cdot \rho$).

Liquid flowmeters capable of achieving the accuracy requirement of the standard are available [2,3]. Turbine flowmeters are among the most popular, but as noted by Reed [4], their accuracy is sensitive to differences in flow pattern, gas entrainment in the heat transfer liquid and wear of the bearings. The accurate determination of q_c by this method also depends on a knowledge of the thermal and physical properties of the heat transfer fluid, primarily the fluid density and specific heat, over a range of temperatures. The temperature dependencies of these properties are often not well known for the commonly used heat transfer fluids such as the water-glycol mixtures.

Recognizing these difficulties, it was decided to use the reference heat source (RHS) method to determine the rate of energy collection of the test solar collector in the Division of Building Research Solar Collector Test Facility [5]. The reference heat source method, also called the calorific ratio technique by Reed and Allen [6], utilizes an electrical heater installed in series with the test solar collector in the calorimetry loop, as shown in Figure 1. With the same mass flowrate through both test collector and RHS, electrical energy is input to the RHS at a rate P_{RHS}, and the resultant fluid temperature rise (ΔT_{RHS}) and the corresponding fluid temperature rise through the test collector (ΔT_c) are measured. The rate of energy collection by the solar collector (q_c) is given by,

$$q_c = \frac{(C_p \cdot \Delta T)_c}{(C_p \cdot \Delta T)_{RHS}} \cdot P_{RHS} \cdot \eta_{RHS} \qquad (2)$$

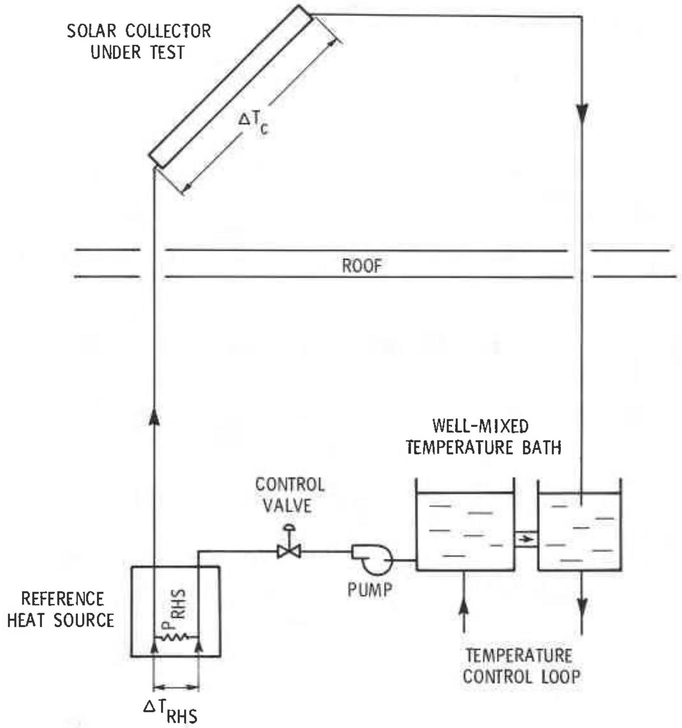

Fig. 1. DBR/NRCC solar collector test loop using the reference heat source method

The thermal transfer efficiency of the reference heat source (η_{RHS}) is defined:

$$\eta_{RHS} = \frac{\text{RHS thermal power output}}{\text{RHS electrical power input}} = \frac{q_{RHS}}{P_{RHS}} \qquad (3)$$

where,

q_{RHS} = rate of thermal energy output to heat transfer fluid in RHS
$= (\dot{m} \cdot C_p \cdot \Delta T)_{RHS}$.

In most solar collector tests, the difference in mean fluid temperature between the RHS and collector can be kept sufficiently small so that the difference in fluid specific heat can be neglected. Good design of the RHS can assure that η_{RHS} approaches 1. The rate of energy collection by the solar collector can, therefore, be approximated by

$$q_c \cong \frac{\Delta T_c}{\Delta T_{RHS}} \cdot P_{RHS} \qquad (4)$$

Thus, the RHS method permits determination of the energy collection by the test collector without directly measuring the fluid flowrate and without knowing the thermophysical properties of the heat transfer fluid. This paper describes the potential sources of error with this method, and the specific design and calibration of the RHS used in the DBR/NRCC Solar Collector Test Facility.

2. SOURCES OF ERROR

To ensure that accurate results are obtained when using a reference heat source, consideration must be given to both its design and its operation.

Heat losses from the reference heat source result in an overprediction of the RHS power input to the fluid and, as a result, an overprediction of the collector performance. These losses can be minimized by insulating the RHS and by using a thermal guard (described in Section 3) within the RHS.

Temperature measurement errors can result from local boiling of the heat transfer fluid on the RHS heater element. This problem is most likely to occur at low degrees of subcooling* and low fluid flowrates. Experiments have been conducted [7] on surface boiling in an annulus flow passage similar to that in the DBR/NRCC RHS. The results of this study can be used to estimate the minimum flowrates and degrees of subcooling required to avoid surface boiling. The maximum allowable heater power density increases with increasing degrees of subcooling and with increasing heater-to-fluid heat transfer coefficient. Thus temperature measurement errors due to local boiling can

* degrees of subcooling are referred to here as the difference (in K) between the fluid average temperature in the RHS and the saturation temperature at the operating pressure. For example if the fluid is water and the average temperature and pressure in the RHS are 90°C and 108.15 kPa respectively (which corresponds to a saturation temp. of 101.85°C) then the fluid has 11.85 K of subcooling.

be avoided by careful consideration of heater power density (power input), degrees of subcooling (operating temperature) and heat transfer coefficient (flowrate).

Effective mixing devices located ahead of the temperature wells are also essential for good temperature measurement. These mixers must remain effective over a wide range of flowrates. As well, fluid friction can increase the fluid temperature and cause an overprediction of ΔT_{RHS}; consequently, the reference heater should be designed to minimize the pressure drop between the temperature sensors.

A major source of error in solar collector testing is non-steady testing conditions. The reference heat source is similar to a collector in that it has thermal mass; thus it can store and release energy during transient operation. This can result in an over- or under-prediction of collector performance. For this reason, the flow rate, inlet temperature, heater power, and subsequent temperature rise must be held constant during each test period. Reference [1] outlines requirements for steady state testing of solar collectors: these same requirements apply to the reference heat source.

To minimize the error resulting from the assumption that the fluid specific heat (C_p) is the same in the heater and the collector, the mean fluid temperature in the RHS must be kept close to that in the solar collector. With water as the heat transfer fluid, this error can usually be kept to less than 0.25% for the temperatures normally encountered in testing. With glycols and oils, however, the value of C_p is strongly dependent on temperature. It is, therefore, suggested that with these heat transfer fluids, the inlet temperature and temperature rise for both the RHS and the test collector be kept approximately the same. This can be accomplished by cooling the fluid between the RHS and the collector, and by adjusting the heater input to make the temperature rise nearly equal across both the RHS and the collector. As shown by the example in Table 1, the error resulting from a large temperature differential between the RHS and the collector is quite significant (≃2%) with a glycol/water mixture, but insignificant (≃0.1%) with water. Although the example is for a "worse case" situation, it does indicate the need for avoiding large differences in temperature conditions between RHS and collector.

TABLE 1 EXAMPLE OF ERROR IN ASSUMING $C_{p,c} = C_{p,\ RHS}$

Collector type: evacuated tube, high fluid residence time

Fluid: (50% by volume ethylene glycol/water mixture)

$T_{m,RHS} = 21.6°C$

$\Delta T_{RHS} = 3.6$ K

$C_{p,RHS} = 3271$ J/(kg·K)

$T_{m,c} = 33.7°C$

$\Delta T_c = 20.2$ K

$C_{p,c} = 3334$ J/(kg·K)

Therefore,

$$\frac{C_{p,RHS}}{C_{p,c}} = \frac{3271}{3334} = 0.981$$

For water, the corresponding ratio would be:

$$\frac{C_{p,RHS}}{C_{p,c}} = \frac{4181}{4178} = 1.0007$$

To summarize: the RHS should be designed to achieve the following:

1) negligible or definable heat loss,
2) low power density heating element, and a high heat transfer coefficient between the heater and the fluid to avoid boiling,
3) good fluid mixing upstream of each temperature measurement point to ensure accurate fluid temperature measurements,
4) small pressure drop between temperature wells to avoid frictional heating of the fluid.

The desired operating conditions for the RHS are:

1) steady operation, i.e., a constant flowrate, inlet temperature, and heater power input,
2) sufficient flowrate, or low outlet temperature, so that the fluid in the RHS does not boil,
3) temperature rise large enough to minimize ΔT errors,
4) mean fluid temperature in the RHS close to that in the collector to minimize C_p errors.

3. DESIGN OF THE DBR/NRCC REFERENCE HEAT SOURCE

Because the DBR/NRCC Solar Collector Test Facility has the capability of testing two collectors simultaneously, it has two separate calorimeter loops, each with its own reference heat source. The two (referred to as RHS#1 and RHS#2) are identical except for minor differences in the thermopile design.

The reference heat source designed at NRCC is shown in Figure 2. It consists of a 750-watt electric immersion heater element enclosed in a section of copper pipe. This represents a maximum power density of approximately 49 kW/m^2.

The RHS was designed to be compact to facilitate the incorporation of a thermal guard ring. The thermal guard is intended to limit the heat losses (or gains) between the internal components and the surroundings. Glass fibre insulation is used to insulate the internal components from each other and from the guard ring. The thermal guard consists of a coiled copper tube soldered to a copper sheet. The fluid is circulated through the guard before entering the heater section of the RHS. This configuration significantly reduces the temperature difference and resultant heat loss between the heater's internal components and its immediate surroundings. The guard ring temperature follows the inlet fluid temperature without the need for a controller even when the heat transfer fluid is cooler than the air adjacent to the RHS. The exterior of the thermal guard ring is insulated to minimize heat losses and limit the temperature gradient along its flow length. An advantage of the fluid guard ring over an electric guard ring, in addition to its capability to operate below the ambient temperature, is that it does not require a control circuit which could introduce "noise" into the temperature measurement.

A mixing device and a temperature measurement well are located at both the inlet and the outlet of the heater. To achieve high resolution and reliability, type "T" (copper-constantan) differential thermopiles are used to measure the fluid temperature rise across the heater elements. (RHS#1 has 10 junctions and RHS#2 has 8.) The thermopiles were calibrated by the Division of Physics of the NRCC, in terms of the International Practical Temperature Scale of 1968 (IPTS-68). The electrical power input (P_{RHS}) is measured with a calibrated electronic power transducer (±0.25% of full scale).

The fluid pipes leading to and from the RHS are insulated to limit axial thermal conduction, as are the wires going to and from the thermopile and heater. The

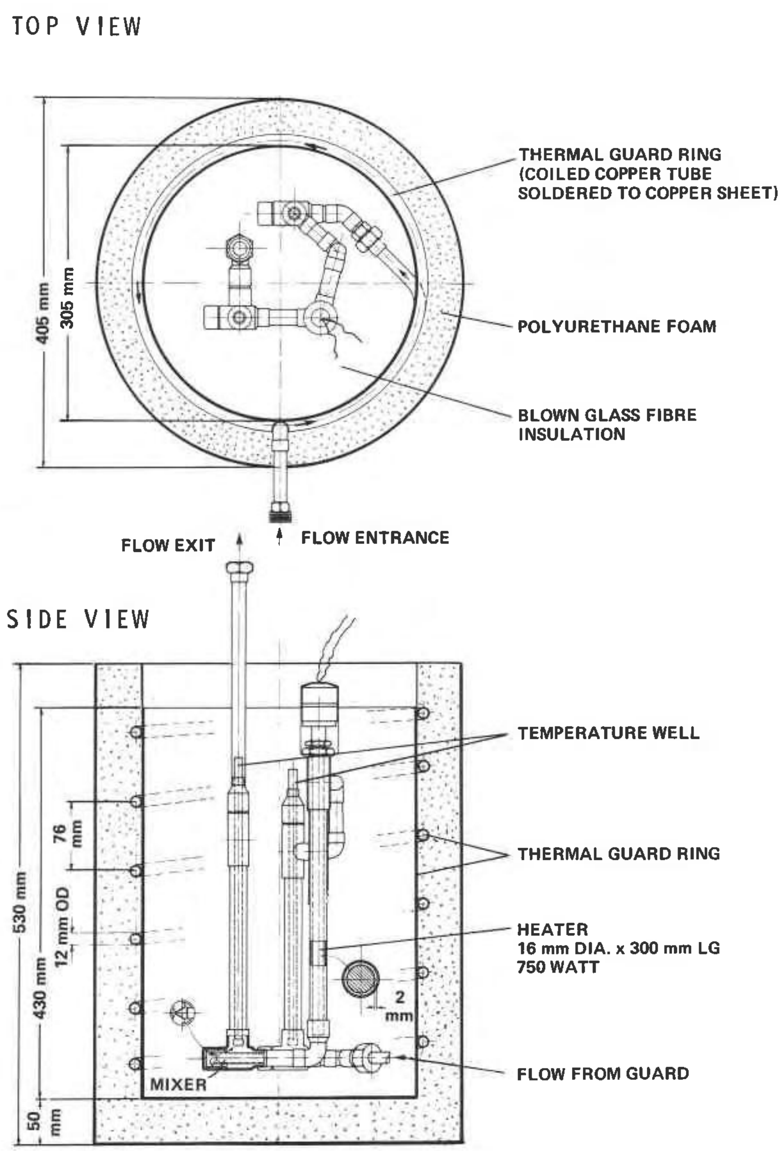

Fig. 2. DBR/NRCC reference heat source

depth of immersion of the temperature wells is approximately 30 diameters, which exceeds the recommendations of the ASHRAE Standard 41.1-74: Section on Temperature Measurements [8].

As mentioned earlier, a stable inlet temperature, good flowrate control and regulation of the power supplied to the RHS are all essential to steady-state operation of the RHS. The DBR/NRCC solar collector test loop uses a pressure-sensitive control valve to maintain the flowrate to ±0.1% of the set point. The stability of the inlet temperature, using a well-mixed temperature bath upstream of the RHS, was ±0.1 K during a test period. Heater power fluctuations are reduced to ±0.1% with the use of a voltage regulator.

4. PERFORMANCE CALIBRATION TESTING

4.1 Test Procedure

A series of calibration tests was performed to determine the effect on η_{RHS}, of;

- temperature rise through heater,
- temperature difference between heater fluid and surrounding air,
- fluid flowrate.

In order to perform the calibration tests, the upper portion of the collector test facility was modified as shown in Figure 3, with a dump tank and a measuring scale in place of the solar collector. This permitted an independent determination of average mass flow rate, by measuring the total mass of fluid flowing through the reference heater over a precisely measured time interval. For the calibration tests, the value of fluid specific heat (C_p) was determined at the mean

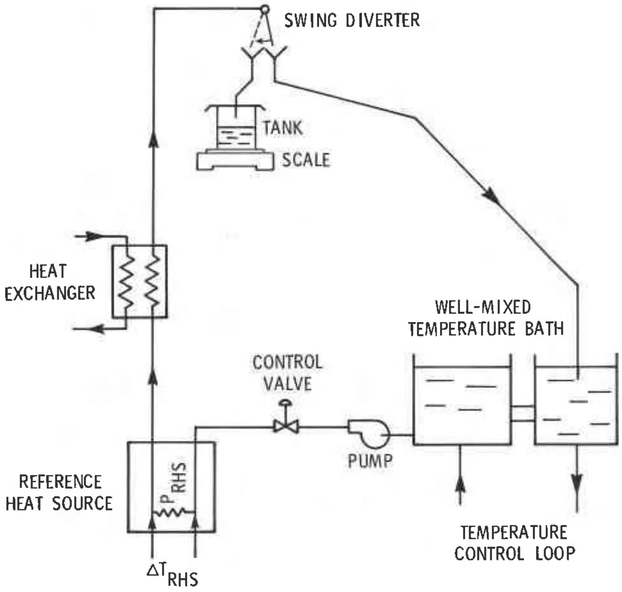

Fig. 3. Calibration set-up

fluid temperature in the heater, using values taken from reference [9].

A calibration test was performed as follows. The heat transfer fluid was drawn from a well-mixed temperature controlled bath, pumped through the heater, cooled and then returned to the bath. Flowrate was maintained constant with a pressure-sensitive control valve. The fluid temperature rise through the heater and the inlet fluid temperature were continously monitored on a strip chart recorder to detect any transient or abnormal conditions, and to indicate when steady conditions were achieved (see Figure 4 for an example of the temperature conditions during a test run). When steady conditions were achieved, i.e. when the inlet temperature did not vary by more than ±0.1 K and the temperature rise across the RHS, by more than ±0.05 K, for a period greater than the test time interval (approx. 3 to 15 min.), the flow was diverted to a weighing scale and an average mass flow measurement obtained. Table 2 gives the results of a typical test run.

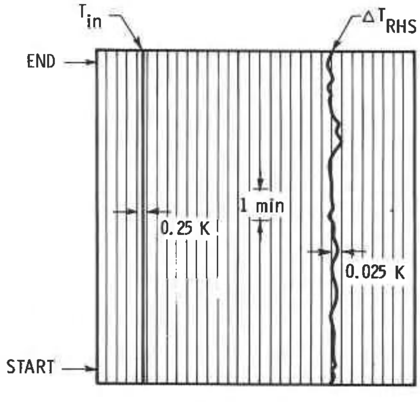

Fig. 4. Variation of the temperature conditions during a typical test run

TABLE 2 SAMPLE DATA FROM CALIBRATION TEST

RHS	#1
Run Number	57
<u>Average Temperature Measurements in RHS</u>	
Inlet fluid temperature (T_i)	44.70°C
Outlet fluid temperature (T_o)	53.08°C
$T_m = \frac{1}{2} \cdot (T_i + T_o)$	48.89°C
ΔT_{RHS}	8.39 K
Ambient temperature (T_a)	27.0°C
$T_m - T_a$	21.9 K
<u>Gravimetric Flow Rate</u>	
Net weight	9.26 kg
Time period	599.9 s
Mass flowrate	0.016 kg/s
Volume flowrate	0.98 L/min
<u>Power</u>	
P_{RHS}	568.1 watts
q_{RHS}	566.6 watts
η_{RHS}	99.7%

At high fluid temperatures, evaporation from the weighing tank introduces a measurement error, especially at low flow rates when the weighing period is long. To reduce this error due to evaporation losses, the fluid was cooled before entering the weighing tank and a plastic cover was used on the tank.

A series of tests were performed to determine the pressure drop characteristic of the reference heat source, as significant fluid friction heating could introduce errors. The pressure drop between thermopile wells was measured at several flowrates from 0.008 to 0.125 kg/s.

To determine the thermal response of the RHS to variations in power input, the following tests were performed at various fluid flowrates. For each flowrate the RHS was allowed to stabilize and the value of ΔT_{RHS} measured. The heater power input to the RHS was then switched off, while the inlet fluid temperature and mass flowrate were maintained constant. The subsequent decay of the value of ΔT_{RHS} was recorded until it was effectively zero.

4.2 Test Results

Figures 5a and 5b show calibration results for two identical RHS's with water as the heat transfer fluid. The thermal efficiency (η_{RHS}) is plotted against the difference between the mean fluid temperature in the RHS and the surrounding ambient air temperature ($T_m - T_a$) for different flowrates. The magnitude of the vertical bars represents the uncertainty associated with each data point (Appendix A). For a flowrate between 0.008 kg/s and 0.050 kg/s (0.5 to 3 L/min.), the values of measured RHS efficiency are slightly less than 100% for the normal range of test temperatures. If a "least squares" curve fit is applied to the data, the value of efficiency is seen to decrease very slightly with increasing values of ($T_m - T_a$).

If the dependence of η_{RHS} on ($T_m - T_a$) is neglected, the value of thermal efficiency (η_{RHS}) obtained for RHS#1 is 99.61% with a standard deviation of 0.22 and for RHS#2, η_{RHS} = 99.59% with a standard deviation of 0.28. Thus the requirement for 1%

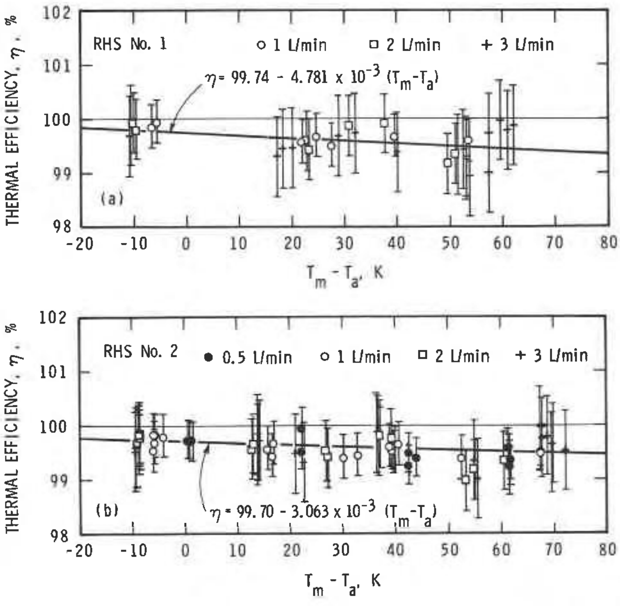

Fig. 5. Calibration results for RHS#1 (a) and RHS#2 (b)

accuracy on the measurement of mass flowrate over the test range as required by ASHRAE [1] is surpassed.

Figure 6 presents the same data (for RHS#2 only) plotted against the temperature rise through the heater (ΔT_{RHS}). A curve fit to the data shows that the efficiency is essentially independent of the temperature rise through the RHS.

Although the losses from the RHS should be zero when $T_m = T_a$ (i.e., η_{RHS} = 100%), the results in Figure 5a and 5b show that this is not the case. The residual error shown could be the result of a systematic error inherent in the design of the RHS. It could also be due to the tabulated values of C_p used in this calibration, which may not adequately account for any dissolved air or minerals present in the test fluid.

Figure 7 shows the pressure drop characteristic of the RHS with water as the heat transfer fluid. With a flow of 0.10 kg/s, a flowrate higher than that normally encountered in solar collector testing, the pressure drop between wells was approximately 350 Pa, which corresponds to approximately 0.035 watts of frictional heating. This amount is two orders of magnitude

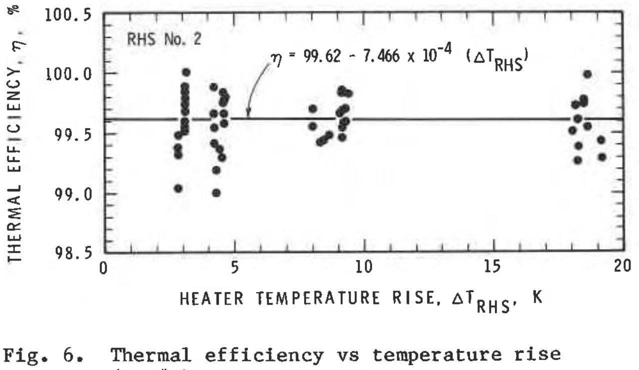

Fig. 6. Thermal efficiency vs temperature rise (RHS#2)

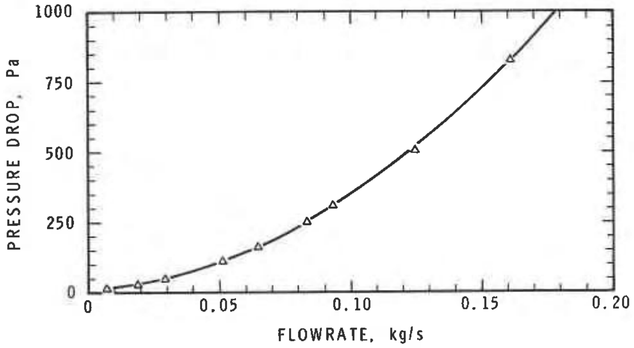

Fig. 7. Pressure drop between thermopile wells in RHS

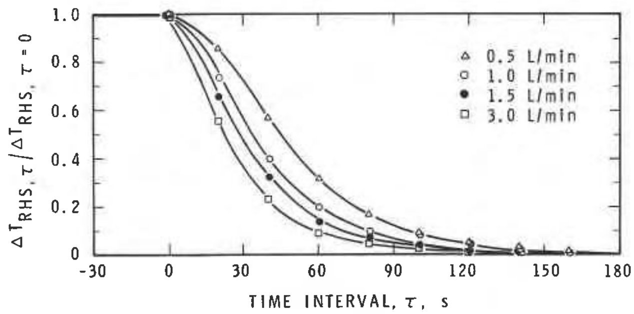

Fig. 8. Thermal response of RHS to a step change in heater power level

smaller than the apparent heat loss from the RHS and is, therefore, negligible.

The transient response of the reference heat source was determined by shutting the heater off during steady-state operation and recording the time decay of the differential temperature measurement. Results are shown in Figure 8 as values of temperature rise ΔT_{RHS} at time interval τ, (normalized to ΔT_{RHS} at $\tau = 0$) vs τ. The results indicate that the response time of the RHS is dependent on the fluid flowrate. The response of ΔT_{RHS} to a step variation in heater power is effectively exponential, preceded by a small time lag which increases at lower flowrates. To avoid errors introduced by the thermal inertia of the RHS, the device should be operated under steady state conditions.

5. CONCLUSIONS

A major advantage of the RHS is its capability to measure the product of mass flow and specific heat directly in the test loop during collector testing. This avoids errors that could be introduced through the use of mechanical flowmeters, which require an accurate knowledge of the temperature dependence of the thermal and physical properties of the heat transfer fluid being used.

To achieve satisfactory results with an RHS, however, care must be exercised in its design and steady-state operation is essential; a stable inlet temperature, good flowrate control and regulation of the power supplied to the heater, are required.

Calibration tests were performed on the NRCC reference heat source over a range of flowrates and inlet temperatures normally encountered during the testing of liquid cooled solar collectors, (i.e., $\dot{m}$ from 0.008 kg/s to 0.05 kg/s and T_i from 10°C to 95°C). When using the RHS outside these operating conditions, caution must be exercised. At low flowrates (≤0.008 kg/s), localized boiling may introduce errors if the heater power density is not reduced as well. Operation at flowrates greater than 0.05 kg/s reduces the temperature rise across the RHS, increasing temperature measurement uncertainty.

Results indicate that the present RHS exceeds the ASHRAE test method requirement [1] of 1% accuracy in the determination of mass flowrate.

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APPENDIX A Analysis of the Random Measurement Uncertainty on η_{RHS}

The uncertainty associated with η_{RHS} can be evaluated using the propagation of random uncertainty techniques [10].

Using equation (3) we have:

$$\eta_{RHS} = \frac{\dot{m} \cdot C_p \cdot \Delta T}{P_{RHS}} \qquad (5)$$

The mass flowrate ($\dot{m}$) is determined by weighing a mass of fluid (M) flowing through the RHS over a precisely measured time interval (τ) so Eq. (5) becomes,

$$\eta_{RHS} = \frac{M \cdot C_p \cdot \Delta T}{\tau \cdot P_{RHS}} \qquad (6)$$

Performing an analysis as described in Ref. 10, the uncertainty in the dependent variable $(W_{\eta_{RHS}})$ can be expressed in terms of the uncertainties in each of the independent variables (W_M, W_τ, W_{cp}, $W_{\Delta T}$, $W_{P_{RHS}}$).

$$\frac{W_{\eta_{RHS}}}{\eta_{RHS}} = \pm\left[\left(\frac{W_M}{M}\right)^2 + \left(\frac{W_{C_p}}{C_p}\right)^2 + \left(\frac{W_{\Delta T}}{\Delta T}\right)^2 + \left(\frac{W_\tau}{\tau}\right)^2 + \left(\frac{W_{P_{RHS}}}{P_{RHS}}\right)^2\right]^{\frac{1}{2}} \qquad (7)$$

where,

$W_M = \pm 0.01$ kg
$W_{C_p} = \pm 0.1\%$ of C_p
$W_{\Delta T} = \pm 0.02$ K
$W_\tau = \pm 0.1$ s
$W_{P_{RHS}} = \pm 1.9$ watts (i.e., 0.25% of 750 watts)

Inserting the values in Eq. (7), we have:

$$W_{\eta_{RHS}} = \pm\left[\left(\frac{0.01}{M}\right)^2 + (0.001)^2 + \left(\frac{0.02}{\Delta T}\right)^2 + \left(\frac{0.1}{\tau}\right)^2 + \left(\frac{1.9}{P_{RHS}}\right)^2\right]^{\frac{1}{2}} \cdot \eta_{RHS}$$

By substituting the corresponding values of M, ΔT and τ, values of $\frac{W_{\eta_{RHS}}}{\eta_{RHS}}$ are obtained. Using the results given in Table 2, for example, the corresponding uncertainty in the determination of η_{RHS}, is:

$$W_{\eta_{RHS}} = \pm\ 0.44\%$$

Values corresponding to each data point are represented by error bars in Figures 5a and 5b.