NRC Publications Archive Archives des publications du CNRC

Thermal performance representation and testing of air solar collectors Bernier, M. A.; Plett, E. G.

This publication could be one of several versions: author's original, accepted manuscript or the publisher's version. /
La version de cette publication peut être l'une des suivantes : la version prépublication de l'auteur, la version
acceptée du manuscrit ou la version de l'éditeur.

For the publisher's version, please access the DOI link below. / Pour consulter la version de l'éditeur, utilisez le lien
DOI ci-dessous.

Publisher's version / Version de l'éditeur:

<https://doi.org/10.4224/40000670>

*Paper (National Research Council of Canada. Institute for Research in
Construction); no. IRC-P-1576, 1988-05*

NRC Publications Archive Record / Notice des Archives des publications du CNRC :

<https://nrc-publications.canada.ca/eng/view/object/?id=32b6c30b-595d-484c-8d54-9ae30f25213a>

<https://publications-cnrc.canada.ca/fra/voir/objet/?id=32b6c30b-595d-484c-8d54-9ae30f25213a>

Access and use of this website and the material on it are subject to the Terms and Conditions set forth at

<https://nrc-publications.canada.ca/eng/copyright>

READ THESE TERMS AND CONDITIONS CAREFULLY BEFORE USING THIS WEBSITE.

L'accès à ce site Web et l'utilisation de son contenu sont assujettis aux conditions présentées dans le site

<https://publications-cnrc.canada.ca/fra/droits>

LISEZ CES CONDITIONS ATTENTIVEMENT AVANT D'UTILISER CE SITE WEB.

Questions? Contact the NRC Publications Archive team at

PublicationsArchive-ArchivesPublications@nrc-cnrc.gc.ca. If you wish to email the authors directly, please see the
first page of the publication for their contact information.

Vous avez des questions? Nous pouvons vous aider. Pour communiquer directement avec un auteur, consultez la
première page de la revue dans laquelle son article a été publié afin de trouver ses coordonnées. Si vous n'arrivez
pas à les repérer, communiquez avec nous à PublicationsArchive-ArchivesPublications@nrc-cnrc.gc.ca.

Ser
TH1
N21d
no. 1576
c. 2
BLDG

**National Research
Council Canada**

**Conseil national
de recherches Canada**

**Institute for
Research in
Construction**

**Institut de
recherche en
construction**

Thermal Performance Representation and Testing of Air Solar Collectors

by M.A. Bernier and E.G. Plett

ANALYZED

Reprinted from
Transactions of the ASME
Journal of Solar Energy Engineering
Vol. 110, May 1988
p. 74-81
(IRC Paper No. 1576)

NRCC 29852

RÉSUMÉ

La performance thermique d'un capteur solaire à air du type à plaque a été évaluée expérimentalement. Les essais ont été effectués aux deux débits recommandés dans la norme 93-1986 de l'ASHRAE et à diverses pressions d'entrée pour simuler trois cas de fuites du capteur (vers l'intérieur, vers l'extérieur, vers l'intérieur et l'extérieur). Ces essais montrent que la performance thermique des capteurs à air dépend du débit et de la pression manométrique d'entrée (ou du débit de fuite correspondant). Si le capteur est utilisé avec une pression manométrique d'entrée proche de la pression atmosphérique, comme c'est souvent le cas dans les systèmes de chauffage sans stockage d'énergie, il est recommandé de mettre le capteur à l'épreuve à une pression manométrique d'entrée nulle. Dans le cas des capteurs fonctionnant à plusieurs pressions d'entrée, il est recommandé de les éprouver à trois pressions d'entrée différentes. Pour le débit d'essai, les présents travaux confirment les résultats des autres études et insistent sur la nécessité de mettre les capteurs à l'épreuve au débit prescrit par le concepteur. Trois méthodes de représentation de la performance thermique ont été utilisées et comparées. La première base le rendement sur le débit d'entrée, la deuxième sur le débit de sortie et la troisième sur l'enthalpie globale. Lorsqu'elles sont représentées sur des graphes normaux, c'est-à-dire de n en fonction de $(T_i - T_a)/G$, les trois méthodes présentent des écarts significatifs, surtout dans le cas de fuite vers l'extérieur. Une analyse d'incertitude des résultats obtenus indique que dans le cas de fuite vers l'intérieur, des incertitudes inacceptables se produisent pour les valeurs élevées de $(T_i - T_a)/G$.

CISTI / ICIST

3 1809 00210 7958

Thermal Performance Representation and Testing of Air Solar Collectors

M. A. Bernier¹

Assistant Research Officer,
Building Services Section,
Institute for Research
in Construction,²
National Research Council Canada,
Ottawa, Canada, K1A 0R6
Mem. ASME

E. G. Plett

Professor.
Mechanical and Aeronautical Eng. Dept.,
Carleton University,
Ottawa, Canada, K1S 5B6

The thermal performance of a plate-type air collector was evaluated experimentally. Tests were performed at the two ASHRAE Standard 93-1986 recommended flowrates and at a variety of inlet gage pressures to cover the three possible cases of collector leakage (inward, outward, inward-outward). These tests show that the thermal performance of air collectors depends on flowrate and inlet gage pressure (or the associated leakage rate). If the collector is used at an inlet gage pressure near atmospheric pressure as is often the case in the no-storage type of systems, then it is recommended to test the collector at an inlet gage pressure of zero. For collectors operating at various inlet gage pressures it is suggested to test them at three inlet gage pressures. As for the test flowrate, the present work confirmed the results of other studies and emphasizes the need to test the collectors at the design flowrate. Three methods of thermal performance representation were used and compared: One method bases the efficiency on the inlet flowrate, another on the outlet flowrate, and the third involves an overall enthalpy balance. When plotted in the classic way, that is, η vs. $(T_i - T_a)/G$ the three methods exhibit significant differences, especially for the inward leakage case. An uncertainty analysis on the data obtained for this study indicates that for the inward leakage case, unacceptable uncertainties occur at high values of $(T_i - T_a)/G$.

1 Introduction

The Institute for Research in Construction of the National Research Council of Canada has been involved between 1981 and 1985 in the development of improved thermal performance test methods and equipment and better representation of air solar collector thermal performance. The equipment used in this research program is described in another paper [1]. The present paper deals with thermal performance representation and testing of air solar collectors with emphasis on flowrate and leakage effects.

The heart of every solar system is the solar collector; and it is important to know, for system design and ranking purposes, the thermal performance of such collectors. Most performance evaluations of air solar collectors are obtained experimentally as analytical evaluations are, at best, difficult. It is generally recognized [2, 3] that the thermal performance representation, testing procedures, and mathematical models associated with air solar collectors have not yet been developed to the same extent as for liquid collectors. The pres-

ent investigation attempts to alleviate some of these deficiencies.

There are two fundamental differences between air and liquid solar collectors. First, because of the relatively poor thermal conductivity of the air, the absorber-to-fluid heat transfer coefficient is much less (approximately one order-of-magnitude [4]) than for liquid collectors. This implies, as will be shown later, that air collectors are more flowrate dependent than liquid collectors. Secondly, most air collectors leak since it is uneconomical to build a perfectly sealed air collector.

Two of the most common types of air solar systems are presented in Figs. 1 and 2. Generally, in the no-storage systems (Fig. 1) several air collectors are connected in parallel. The inlet temperature to the collector is either the ambient temperature or the room temperature and the inlet gage pressure is zero (or slightly negative). These systems are relatively inexpensive and are used for such applications as barn heating, warehouse fresh-air preheaters, and "day-time" house heating. A more complete system [5] (Fig. 2) consists of a series-parallel arrangement of air solar collectors and a storage unit. The inlet temperature to the collector array is usually close to the room temperature (20°C) and the collector array inlet pressure varies from system to system depending on whether the blower is placed downstream or upstream of the collector. It is important to note that when collectors are in series, the last collector will have an inlet gage pressure, and thus a leakage rate, which is different from the first because of

¹ Present address: Graduate Student, Mech. Eng. Dept., McGill University, 817 Sherbrooke West, Montréal, Québec, Canada, H3A 2K6.

² Formerly the Division of Building Research, National Research Council Canada.

Contributed by the Solar Energy Division for publication in the JOURNAL OF SOLAR ENERGY ENGINEERING. Manuscript received by the Solar Energy Division, July 1987.

Fig. 1 Two common types of no-storage systems

Fig. 2 Example of a complete air solar system [5]

the pressure drop in the preceding collectors. For the same reason, the leakage rate along one collector will vary from inlet to outlet (particularly at high flowrates).

During the design of a typical solar energy system it is important to know the performance data of the individual system components, particularly the collectors. Usually the design flowrate, inlet temperature, and operating pressure are known as well as meteorological conditions. The designer, in order to make a good system selection, needs a thermal performance assessment of the collector under these operating conditions. This assessment is generally based on an efficiency curve obtained through testing. Tests are generally performed for the manufacturer by a recognized testing agency at conditions which, often, differ from the expected operational conditions.

Thermal performance tests, which are generally based on the ASHRAE Standard 93-1986 testing procedure [6] (update

Fig. 3 Sectional drawing of the collector used in the present study

of ASHRAE Standard 93-77 [7]), are intended to rate different collectors as tested under similar conditions, and to provide enough information for system design.

The Standard recommends testing air collectors at four different inlet temperatures and two flowrates; $0.01 \text{ m}^3/\text{s}\cdot\text{m}^2$ and $0.03 \text{ m}^3/\text{s}\cdot\text{m}^2$. In cases where the collector is designed for a flowrate much different from these two, the collector should be (according to Standard 93-1986) tested at the design flowrate. In addition, the Standard suggests an inlet gage pressure of -124 Pa or $+124 \text{ Pa}$ (or the maximum allowable operating pressure specified by the manufacturer, whichever is smaller) depending on whether the collector is tested under negative or positive gage pressure.

The present experimental investigation examines the effect of such variations of flowrate and inlet gage pressure on the thermal performance of an air collector. The collector chosen for this study is a plate-type double-glazed air collector (Fig. 3) with a nonselective absorber plate.

Furthermore, this report evaluates three methods of representing the thermal efficiency of an air collector. One method bases the efficiency on the inlet mass flowrate, another on the outlet flowrate, and the third is based on an enthalpy balance method.

2 Thermal Performance Representation

2.1 Thermal Performance Representation Based on the Inlet or Outlet Mass Flowrate. In the case where the thermal efficiency is based on the inlet or outlet flowrate the governing equation is:

$$\eta = \frac{Q_c}{G \cdot A} = \frac{\dot{m} \cdot C_p (T_o - T_i)}{G \cdot A} \quad (1)$$

where $\dot{m} = \dot{m}_i$ if based on inlet flowrate, and $\dot{m}_o$ if based on outlet flowrate.

Nomenclature

A = collector gross area (m^2)
 C_p = air specific heat ($\text{J}/\text{kg}\cdot\text{K}$)
 F_R = collector heat removal factor (dimensionless)
 G = global irradiance incident on the plane of the collector (W/m^2)
 $\dot{m}_i$ = inlet mass flowrate (kg/s)
 $\dot{m}_o$ = outlet mass flowrate (kg/s)
 P_i = collector inlet gage pressure (Pa)

P_o = collector outlet gage pressure (Pa)
 Q_c = collected energy (W)
 T_a = ambient temperature ($^{\circ}\text{C}$)
 T_i = collector inlet temperature ($^{\circ}\text{C}$)
 T_o = collector outlet temperature ($^{\circ}\text{C}$)
 U_L = overall collector heat loss coefficient (W/m^2)
 ΔT = differential temperature across the collector (K)

η = collector thermal efficiency (%)
 $(\tau \cdot \alpha)_e$ = collector transmittance-absorptance product

Subscripts

e = enthalpy
 i = in
 L = leakage
 L_i = inward leakage
 L_o = outward leakage
 o = out

Fig. 4 Possible types of leakage in an air collector

2.2 Thermal Performance Based on an Enthalpy Balance. The enthalpy balance method takes into account all of the energy flows into or out of the collector. It is based on the first law of thermodynamics, which for steady-state, steady flow conditions with negligible contributions from kinetic and potential energy terms may be expressed by:

$$Q_c + \Sigma \dot{m}_i h_i = \Sigma \dot{m}_o h_o \quad (2)$$

As three different leakage cases (Fig. 4) can occur in an air collector, three different enthalpy balances are possible.

2.2.1 Inward Leakage ($P_i < 0$, $P_o < 0$, $\dot{m}_i < \dot{m}_o$). Application of equation (2) to the control volume of Fig. 4(a) gives;

$$Q_c + \dot{m}_i h_i + \dot{m}_L h_L = \dot{m}_o h_o \quad (3)$$

Assuming constant specific heat and referencing the enthalpy to an ambient temperature base, T_a , equation (3) becomes:

$$Q_c = \dot{m}_o C_p (T_o - T_a) - (\dot{m}_o - \dot{m}_L) C_p (T_i - T_a) \quad (4)$$

or by rearranging to obtain the thermal efficiency;

$$\eta = \frac{Q_c}{G \cdot A} = \frac{\dot{m}_o C_p (T_o - T_i) + (\dot{m}_o - \dot{m}_i) C_p (T_i - T_a)}{G \cdot A} \quad (5)$$

2.2.2 Outward Leakage ($P_i > 0$, $P_o > 0$, $\dot{m}_i > \dot{m}_o$). An enthalpy balance on the control volume of Fig. 4(b) yields;

$$Q_c + \dot{m}_i h_i = \dot{m}_o h_o + \dot{m}_L h_L \quad (6)$$

Referencing the enthalpy to an ambient temperature base and making the same assumptions as for the inward leakage case, we obtain:

$$\eta = \frac{Q_c}{G \cdot A} = \frac{\dot{m}_o C_p (T_o - T_i) + (\dot{m}_i - \dot{m}_o) C_p (T_L - T_i)}{G \cdot A} \quad (7)$$

It is worth noting that when the outlet mass flowrate is zero, which is not likely to occur, the efficiency values can be greater than zero even though no heated air is delivered. The difficulty in evaluating this equation is that T_L is unknown.

But it can be approximated by assuming that the air is leaking out at the average fluid temperature; thus in the present study:

$$T_L = (T_i + T_o)/2, \quad \text{or} \quad T_L = T_i + \Delta T/2 \quad (8)$$

This method of representing thermal performance for an outward leakage case differs from the one in ASHRAE 93-1986 [6] where the enthalpy balance is performed on the load and not the collector. The present authors prefer to represent the thermal performance of the *collector* as it is the part of the system being tested. Then the collector test results could be used as inputs to the enthalpy balance on the load as was shown by Bernier [10].

2.2.3 Inward-Outward Leakage ($P_i > 0$, $P_o < 0$). This situation is most likely to occur at high flowrates where the pressure drop through the collector overcomes the inlet positive pressure thus establishing the point of zero pressure (neutral point) inside the collector. An enthalpy balance on the control volume of Fig. 4(c) results in:

$$Q_c + \dot{m}_i h_i + \dot{m}_{L_i} h_{L_i} = \dot{m}_o h_o + \dot{m}_{L_o} h_{L_o} \quad (9)$$

Assuming, as above, constant specific heats and referencing the enthalpy to an ambient temperature base, equation (9) becomes (when rearranged in terms of η):

$$\eta = \frac{Q_c}{G \cdot A} = \frac{\dot{m}_o C_p (T_o - T_i) + (\dot{m}_o - \dot{m}_i) C_p (T_i - T_a) + \dot{m}_{L_o} C_p (T_{L_o} - T_a)}{G \cdot A} \quad (10)$$

This equation is essentially the same as for the inward leakage case except for the last term on the right-hand side of the numerator. In the present study, T_{L_o} was assumed equal to the average fluid temperature:

$$T_{L_o} = (T_i + T_o)/2 \quad (11)$$

In this study, only one performance evaluation was obtained under an inward-outward leakage condition ($P_i = +150$ Pa). For this condition $\dot{m}_{L_o}$ was estimated at 0.00364 kg/s, which was the leakage rate obtained for this collector under a static leakage test where the whole collector was pressurized to +150 Pa. It is to be noted that the inward-outward leakage case is not covered in ASHRAE 93-1986 [6].

3 Experimental Results

The thermal performance of the air collector was evaluated outdoors using the NRCC air collector test facility which is depicted in Fig. 5. A complete description of the facility can be found in [1]. Briefly, the altazimuthal facility is an open-loop two-blower system with capabilities to vary and measure inlet and outlet gage pressures. In addition, the inlet and outlet mass flowrates are measured using orifice plates. Measurements of ambient temperature, inlet temperature to the collector, differential temperature, global solar irradiance, wind speed, and relative humidity are also performed. When testing air solar collectors, header heat losses (i.e., heat losses in the pipe sections from the temperature measuring locations to the collector) can introduce significant systematic errors in thermal efficiency even if the headers are well insulated [8]. In the present study the correction procedure outlined in [8] for correcting header heat losses has been followed. Individual calibrations of most measurements as well as an overall calibration [1, 9] of the facility were undertaken, the results of which are summarized in Table 1.

Experimental testing was performed during one season (Winter 1984-85) under similar meteorological conditions so that differences in efficiencies among experimental data sets are mainly due to differences in flowrate and inlet gage

Fig. 5 Air flow circuit of the NRCC air solar collector test facility

Table 1 Uncertainty analysis for representative cases

CALIBRATION RESULTS [1]		Inward Leakage (low value of $(T_i - T_a)/G$)		Inward Leakage (high value of $(T_i - T_a)/G$)		Outward Leakage (high value of $(T_i - T_a)/G$)		Inward-Outward Leakage	
Measurement	Random Uncertainty	Measured Value	Contribution to ω_η $((\delta\eta/\delta x_i)\omega_{x_i}^2)$	Measured Value	Contribution to ω_η $((\delta\eta/\delta x_i)\omega_{x_i}^2)$	Measured Value	Contribution to ω_η $((\delta\eta/\delta x_i)\omega_{x_i}^2)$	Measured Value	Contribution to ω_η $((\delta\eta/\delta x_i)\omega_{x_i}^2)$
$\dot{m}_i$	$\pm 1.5\%$	0.0247 kg/s	6.35×10^{-5}	0.0239 kg/s	2.47×10^{-4}	0.0209 kg/s	1.24×10^{-6}	0.0681 kg/s	9.52×10^{-4}
C_p	$\pm 0.1\%$	1003.8 J/kg·K	1.76×10^{-7}	1008.8 J/kg·K	3.79×10^{-8}	1010.5 J/kg·K	2.24×10^{-8}	1008.2 J/kg·K	1.09×10^{-7}
ΔT	± 0.1 K	30.80 K	1.95×10^{-6}	13.15 K	1.86×10^{-6}	12.22 K	1.47×10^{-6}	8.75 K	1.18×10^{-5}
$\dot{m}_o$	$\pm 1.5\%$	0.0234 kg/s	2.02×10^{-6}	0.0234 kg/s	1.63×10^{-4}	0.0233 kg/s	1.54×10^{-6}	0.0707 kg/s	7.47×10^{-4}
T_i	$\pm 0.5^\circ\text{C}$	-6.08 °C	1.36×10^{-7}	49.34 °C	2.03×10^{-8}	43.46 °C	4.85×10^{-7}	43.93 °C	4.31×10^{-7}
T_a	$\pm 0.5^\circ\text{C}$	-13.22 °C	1.36×10^{-7}	-14.40 °C	2.03×10^{-8}	-	-	-7.10 °C	6.38×10^{-8}
T_L	$\pm 25\%$	-	-	-	-	49.57 °C	1.81×10^{-5}	-	-
G	$\pm 3\%$	1121.4 W/m ²	1.72×10^{-4}	1114.8 W/m ²	3.51×10^{-5}	1096.2 W/m ²	2.21×10^{-5}	1259.0 W/m ²	1.01×10^{-4}
A	± 0.005 m ²	1.58 m ²	1.91×10^{-6}	1.58 m ²	3.90×10^{-7}	1.58 m ²	2.45×10^{-7}	1.58 m ²	1.12×10^{-6}
$\dot{m}_{LO}$	$\pm 25\%$	-	-	-	-	-	-	0.00364 kg/s	6.34×10^{-4}
T_{LO}	$\pm 25\%$	-	-	-	-	-	-	48.30 °C	1.58×10^{-5}
Efficiency (η)		0.44		0.20		0.16		0.34	
Absolute Uncertainty (ω_η)		± 0.016		± 0.021		± 0.007		± 0.050	
Relative Uncertainty (ω_η/η)		$\pm 3.6\%$		$\pm 10.7\%$		$\pm 4.3\%$		$\pm 14.8\%$	

pressures and not to meteorological factors. The collector was tested at the two recommended flowrates ($0.01 \text{ m}^3/\text{s}\cdot\text{m}^2$ and $0.03 \text{ m}^3/\text{s}\cdot\text{m}^2$) and at a variety of inlet gage pressures ($P_i = -150 \text{ Pa}$ to $+150 \text{ Pa}$). All the experimental data obtained for this study (tabulated in [10]) met the ASHRAE Standard 93-1986 stability and steady-state requirements.

The efficiency results were grouped into two categories (low flowrate, Fig. 6, and high flowrate, Fig. 7) and are plotted vs. $(T_i - T_a)/G$. In both figures the same data were plotted three different ways: In Figs. 6(a) and 7(a), the efficiency is based on the inlet mass flowrate; in Figs. 6(b) and 7(b), the efficiency is based on the outlet mass flowrate; and in Figs. 6(c) and 7(c), the efficiency is based on an enthalpy balance.

In each figure the inlet flowrate is approximately the same for every curve but the inlet gage pressure (and consequently the outlet mass flowrate) is different. Each curve is obtained

from a first-order least-squares fit. Error bars in Figs. 6(c) and 7(c) represent the experimental uncertainty on η (which is treated in the appendix).

3.1 Leakage Effects. It should first be noted that the collector has a leakage rate of 26 percent of $\dot{m}_i$ (at $P_i = -150 \text{ Pa}$ and $\dot{m}_i = 0.0235 \text{ kg/s}$) and it can be considered to be a "leaky" collector. In Figs. 6 and 7, a comparison among curves indicates that the collector efficiency is dependent on the inlet pressure (or leakage rate).

But what should be the inlet gage pressure during testing? If all systems were of the no-storage type (Fig. 1), the obvious choice for inlet gage pressure would be $P_i = 0 \text{ Pa}$ as these systems operate near (or at) atmospheric pressure. But for the series-parallel arrangement (Fig. 2), the inlet gage pressure will vary from system to system as well as within the system. Ideal-

Fig. 6 Plot of the same low flowrate data for the three methods considered here: (a) The efficiency values are based on the inlet flowrate. (b) The efficiency values are based on the outlet flowrate. (c) The efficiency values are based on an enthalpy balance.

Fig. 7 Plot of the same high flowrate data for the three methods considered here: (a) The efficiency values are based on the inlet flowrate. (b) The efficiency values are based on the outlet flowrate. (c) The efficiency values are based on an enthalpy balance.

Fig. 8 Plot of the same data for three different methods (outward leakage case)

Fig. 9 Plot of the same data for three different methods (inward leakage case)

ly, for such systems, the designer should have performance curves for the operational gage pressures. In reality, the designer must rely on performance curves obtained at inlet gage pressures other than the operational gage pressure (see Introduction). It is probably prohibitive to ask the designer (or the manufacturer) to have collectors tested each time there is a new system design. A compromise solution would be to have every collector tested at more than one inlet gage pressure, say at three inlet gage pressures (for example at $P_i = -124$ Pa, $P_i = 0$ Pa, to cover the no-storage systems, and $P_i = +124$ Pa) in order to give the designer a range of efficiency values obtained over a range of inlet gage pressures.

3.2 Flowrate Effects. As far as the flowrate effects are concerned, it is hard to separate them from the inlet pressure effects in the curves of Fig. 6 and Fig. 7. Nonetheless, if curves of zero inlet gage pressure ($P_i = 0$ Pa, where the leakage effects are minimal) for low (Fig. 6c) and high flowrates (Fig. 7c) are compared, the trend indicates that thermal efficiency values are higher at high flowrate. This confirms the results of other studies (e.g., [11]) on the flowrate dependency of air collectors. Because of this flowrate dependency (at least for the collector considered here), it is advisable to have the collector tested at the design flowrate. If the design flowrate is unknown when testing, then it is suggested to test the collector at the two ASHRAE recommended flowrates. These two flowrates are good as, in most cases, they represent the minimum and maximum flowrates under which air collectors operate.

3.3 Comparison Between the Inlet, Outlet and Enthalpy Balance Methods. As for the thermal performance represen-

tation, Figs. 8 and 9 have been drawn to help visualize the differences between the inlet, the outlet and the enthalpy balance method. In these figures the same data points are plotted three different ways: efficiency based on the inlet flowrate, on the outlet flowrate, and on the enthalpy balance approach. Figure 8 is for the outward leakage case and Fig. 9 for the inward leakage case.

3.3.1 Outward Leakage. As seen on Fig. 8, for the outward leakage case, all types of thermal performance representation give similar efficiency curves and the three curves merge at $\eta = 0$ (where $\Delta T = 0$) when they are extrapolated beyond the data sets. This is to be expected as a $\Delta T = 0$ yields $\eta = 0$ in equations (1) and (7).

When the thermal performance is based on the inlet mass flowrate, the user can only calculate the available input energy, i.e., $\dot{m}_i \cdot C_p \cdot (T_o - T_i)$, without knowing what is the available output energy ($\dot{m}_o \cdot C_p \cdot (T_o - T_i)$). Similarly, if the thermal performance is based on the outlet mass flowrate, the user will get the available output energy without knowing if it is greater than the available input energy. The enthalpy balance method is the only method where all the energy flows into or out of the collector are taken into account. An example will illustrate this. Assuming the following conditions:

$$G = 500 \text{ W/m}^2$$

$$T_a = 0^\circ \text{C}$$

$$T_i = 40.0^\circ \text{C}$$

$$\text{then } (T_i - T_a)/G = 0.08 \text{ m}^2 \cdot \text{K/W}$$

we can find from the curve fits to the data in Fig. 8:

$$\eta_i = 6.6\%, \eta_o = 5.8\%, \eta_e = 6.2\%$$

where, η_i , η_o , and η_e are the efficiencies based on the inlet flowrate, the outlet flowrate, and on an enthalpy balance respectively.

Using equations (1) and (7) (for $A = 1.58 \text{ m}^2$) we have:

$$Q_c = 52.1 \text{ W} \quad (\text{based on the inlet mass flowrate, available input})$$

$$Q_c = 45.8 \text{ W} \quad (\text{based on the outlet mass flowrate, available input})$$

and for the enthalpy balance method, we obtain:

$$\begin{aligned} Q_c &= \dot{m}_o C_p (T_o - T_a) + \dot{m}_L C_p (T_L - T_a) - \dot{m}_i C_p (T_i - T_a) \\ Q_c &= \text{available output} + \text{leakage output} - \text{available input} \\ &\quad \text{referenced to } T_a \quad \text{referenced to } T_a \quad \text{referenced to } T_a \\ &\quad 895 \quad + \quad 95 \quad - \quad 941 \quad (\text{W}) \\ &= 49 \text{ W} \end{aligned}$$

As can be seen the enthalpy balance method gives a clearer picture of the situation and in fact, in this particular example, it is the only method showing that the available output is less than the available input.

3.3.2 Inward Leakage. The efficiency results for the inward leakage case are presented in Fig. 9. As expected, from equations (1) and (5), the enthalpy and outlet curve merge at the intercept where $(T_i = T_a)$ and the inlet and outlet curves merge at $\eta = 0$ where $\Delta T = 0$. The efficiency values are quite dependent on the method of representation and it appears that the enthalpy balance method supplies significantly higher values of efficiency. It gives a positive efficiency even though the temperature rise across the collector is zero. This could be extremely misleading for someone who does not know that the curve is based on an enthalpy balance. Again, an example is perhaps the best way of illustrating the situation. Let's assume that the operational and meteorological conditions are such that:

$$G = 500 \text{ W/m}^2$$

$$T_i = 32.5^\circ \text{C}$$

$$T_a = 0^\circ \text{C}$$

$$\text{then } (T_i - T_a)/G = 0.065 \text{ m}^2 \cdot \text{K/W}$$

If the user takes the efficiency curves based on the inlet and outlet flowrates he finds that $\eta = 0$ for both curves, indicating that there is no available output. Based on the enthalpy balance method, the user finds that the efficiency is:

$$\eta_e = 25.6\%$$

Recalling the enthalpy balance equation (equation 4),

$$\begin{aligned} Q_c &= \dot{m}_o C_p (T_o - T_a) - \dot{m}_i C_p (T_i - T_a) \\ &= \text{available output} - \text{available input} \\ &\quad \text{referenced to } T_a \quad \text{referenced to } T_a \\ &= 967 - 768 \quad (\text{W}) \\ Q_c &= 199 \text{ W} \end{aligned}$$

and the available output is greater than the available input, something that the inlet and outlet mass flowrate methods could not detect.

3.4 Improvements to the Inlet and Outlet Mass Flowrate Methods. As shown in the previous sections, the inlet and outlet mass flowrate methods do not give a proper account of all the energy flows into or out of the collector. This tends to corroborate the results of Close and Yusoff [13] who showed analytically the errors involved with thermal efficiency by measuring only the inlet or outlet mass flowrates.

If along with the efficiency curves obtained with the inlet and outlet mass flowrate methods, the user is given the outlet and inlet mass flowrate, respectively, he could perform the same kind of analysis as the one performed with the enthalpy balance method.

For example, on Fig. 7, if along with the outlet mass flowrate curve, the user is also given the inlet mass flowrate then he could go back to equation (5) and perform the same kind of analysis as discussed in section 3.3.2. to show that the available output (referenced to T_a) is greater than the available input (referenced to T_a).

Thus, the inlet and outlet mass flowrate methods, if documented properly, could account (as they should) for all the energy flows into and out of the collector. In doing so both the inlet and outlet mass flowrate methods lead to the same results as the enthalpy balance method. It is thus recommended to use the enthalpy balance method to represent the thermal performance of air collectors.

4 Conclusions and Recommendations

In the present investigation values of thermal efficiency for a plate-type air collector were obtained experimentally using a calibrated test facility. The collector was tested at the two ASHRAE recommended flowrates ($0.01 \text{ m}^3/\text{s} \cdot \text{m}^2$ and $0.03 \text{ m}^3/\text{s} \cdot \text{m}^2$) and at a variety of inlet gage pressures ($P_i \approx -150 \text{ Pa}$ to $\approx +150 \text{ Pa}$). Based on these results the following conclusions can be drawn.

4.1 Thermal Performance Representation. The thermal

Fig. 10 Suggested test results presentation

efficiency was evaluated using three methods. In the first two methods, referred to as the inlet and outlet mass flowrate methods, the efficiency is based on equation (1). The other method, called the enthalpy balance method (equations 5, 7, 10) takes into account all of the energy flows into or out of the collector.

When used directly, the inlet and outlet mass flowrate methods cannot provide all the information needed in system design. They have to be supplemented by the outlet and inlet mass flowrates, respectively, in order to account for all the energy flows into or out of the collector. In doing so, both methods end up being identical to the enthalpy balance method. Thus, for directness and simplicity, the enthalpy balance method is recommended.

It is important to give the user the basis of the efficiency curve. To remove any ambiguity, it is recommended to put the equation alongside the thermal performance curve on the graph. For example, Fig. 10 suggests a format of presentation. It is to be noted that the parameters $F_R U_L$ and $F_R \cdot (\tau \cdot \alpha)_e$, commonly used to represent the slope and the intercept of the curve, are misleading in the context of a leaky collector. To avoid confusion and possible misinterpretation of the curve, the slope and the intercept should not be reported as being equal to $F_R U_L$ and $F_R \cdot (\tau \cdot \alpha)_e$, but simply as the slope and the intercept of the curve.

4.2 Test Procedure. A comparison of Figs. 6(c) and 7(c) at $P_i = 0$ Pa show the flowrate dependency of this collector. Based on this, it is suggested to test the collector at the design flowrate. If the design flowrate is unknown at the time of test, it is probably wise to have the collectors tested at two flowrates as currently suggested by ASHRAE [6].

The work presented here also shows that there is a need to standardize the inlet gage pressure during testing as different testing laboratories may come up with different performance curves depending on the inlet gage pressure used (see, for example, Figs. 6 and 7). In that regard, the new Standard 93-1986 [6] represents an improvement over the old Standard 93-77 [7] in that it requires testing at a prescribed inlet gage pressure.

More than the need to standardize the inlet gage pressure, there is a need to provide enough information for system design. A single performance curve obtained under a prescribed inlet gage pressure would probably not be sufficient for the designer because air collectors operate under a variety of inlet gage pressures.

If the collector is used at an inlet gage pressure near atmospheric pressure, as is often the case in the no-storage type of system, then it is recommended to test the collector at an inlet pressure of zero. For collectors operating at various inlet gage pressures, it is recommended to test them at three inlet

gage pressures (say $P_i = -124$ Pa, $P_i = 0$ Pa, $P_i = +124$ Pa) to give efficiency values covering a broader range of operating conditions.

4.3 Recommendations for Future Work. This study concentrated on one particular type of collector (plate-type) that was particularly "leaky." It is suggested to do more work on other generic types of collector (e.g., matrix) and on leakier/tighter collectors to ascertain the validity of the conclusions of the present study as applied to these other collector types.

References

- 1 Bernier, M. A., NRCC Air Collector Test Facility: Description, Calibration and Data Manipulation, Institute for Research in Construction, Internal Report no. 524, Ottawa, March 1987.
- 2 Report on the IEA Workshop on Solar Air Heating Systems, International Energy Agency, Norway, 7-8 May 1984.
- 3 Gillet, W. B., Aranovitch, E., and Moon, J. E., "Solar Collector Testing in the European Community," *International Journal of Solar Energy*, Vol. 1, 1983, pp. 317-341.
- 4 Shewen, E. C., Hollands, K. G. T., Remick, C. D., et al., "Development of a Fixed Position, Fixed Configuration Air Heating Solar Collector, Solar Energy Project," Report No. Coll-9, National Research Council Canada, Ottawa, 1980.
- 5 Solaron Corporation, Solar Energy Systems, 1977.
- 6 "Methods of Testing to Determine the Thermal Performance of Solar Collectors," American Society for Heating, Refrigerating and Air-Conditioning Engineers, ASHRAE Standard 93-1986, 1986.
- 7 "Methods of Testing to Determine the Thermal Performance of Solar Collectors," American Society for Heating, Refrigerating and Air-Conditioning Engineers, ASHRAE Standard 93-77, 1977.
- 8 Bernier, M. A., "Correcting for Header Heat Losses when Testing Solar Collectors," ISES World Congress, Montréal, June 1985.
- 9 Bernier, M. A., "NRCC Air Collector Test Facility," First E.C. Conference On Solar Heating, Amsterdam, April 30-May 4, 1984, pp. 499-503.
- 10 Bernier, M. A., "Thermal Performance Representation and Testing of Air Solar Collectors," M. Eng. Thesis, Carleton University, Ottawa, 1985.
- 11 Mumma, S. A., Yellott, J. I., and Wood, B., "The Application of ASHRAE Standard 93-77 to the Thermal Performance Testing of Air Solar Collectors," ASHRAE Transactions 1978, Paper AL-78-3, No. 2.
- 12 Close, D. J., and Yusoff, M. B., "The Effects of Air Leaks on Solar Air Collector Behavior," *Solar Energy*, Vol. 20, 1978, pp. 459-463.
- 13 Kline, S. J., and McClintock, F. A., "Describing Uncertainties in Single-Sample Experiments," *Mechanical Engineering*, Jan. 1953, pp. 3-8.
- 14 Moffat, R. J., "Contributions to the Theory of Single-Sample Uncertainty Analysis," *ASME J. of Fluids Engineering*, Vol. 104, No. 2, June 1982, pp. 250-260.
- 15 Hill, J. E., Streed, E. R., "A Method of Testing for Rating Solar Collector Based on Thermal Performance," *Solar Energy*, Vol. 18, 1976, pp. 421-429.

APPENDIX

Uncertainty Analysis

In this appendix individual measurement uncertainties are combined to look at their effect on the overall uncertainty on η .

In their classic paper, Kline & McClintock [13] presented a method of combining individual random uncertainties to give the overall uncertainty in a derived result, η in our case. Taking the inward leakage case as an example (equation 5) and assuming that each measurement is independent, normally distributed, and that individual measurement uncertainties have the same odds (20 to 1), then the overall uncertainty on η , ω_η , can be expressed as:

$$\omega_\eta = \{ [(\delta\eta/\delta\dot{m}_o)\omega_{\dot{m}_o}]^2 + [(\delta\eta/\delta C_p)\omega_{C_p}]^2 + [(\delta\eta/\delta\dot{m}_i)\omega_{\dot{m}_i}]^2 + [(\delta\eta/\delta\Delta T)\omega_{\Delta T}]^2 + [(\delta\eta/\delta T_i)\omega_{T_i}]^2 + [(\delta\eta/\delta T_a)\omega_{T_a}]^2 + [(\delta\eta/\delta G)\omega_G]^2 + [(\delta\eta/\delta A)\omega_A]^2 \}^{1/2} \quad (A-1)$$

where $\omega_{\dot{m}_o}$, ω_{C_p} , $\omega_{\dot{m}_i}$, $\omega_{\Delta T}$, ω_{T_i} , ω_{T_a} , ω_G , ω_A are the random uncertainties on the measured variables. These uncertainties are presented in Table 1 [1]. Equation A-1 was solved using the approach described by Moffat [14], to obtain the in-

dividual contribution of each measured variable, $((\delta\eta/\delta x_i)\omega_{x_i}^2)$, as well as the overall uncertainty on η , ω_η .

Error bars in Figs. 6(c) and 7(c) represent the overall uncertainty on η , ω_η . Individual contributions of each measurement to the overall uncertainty on η are presented in Table 1 for representative cases.

The first thing to note in Table 1 is that the contribution of the uncertainties in T_o , T_i , C_p , ΔT , and A are all negligible (i.e., their contribution to ω_η is at least one order-of-magnitude less than other contributors to ω_η) in the determination of the overall uncertainty on η .

Inward Leakage. Table 1 shows two representative inward leakage cases (one at a low value of $(T_i - T_o)/G$ and one at a high value of $(T_i - T_o)/G$). For the low $(T_i - T_o)/G$ case, it can be seen that the major contributor to the uncertainty on η is the uncertainty on the global irradiance (G). The relative uncertainty on η , (ω_η/η) , is ± 3.6 percent and is comparable to the uncertainty obtained in liquid collector testing [15]. For high values of $(T_i - T_o)/G$, however, ω_η/η is ± 10.7 percent and $\dot{m}_i$ and $\dot{m}_o$ are now the major contributors to the uncertainty on η , surpassing the contribution of G . This strange behavior can be best explained by looking at the numerator of equation (5):

$$Q_c = \dot{m}C_p(T_o - T_i) + (\dot{m}_o - \dot{m}_i)C_p(T_i - T_a) \quad (\text{A-2})$$

Inserting the values found in Table 1 for the low $(T_i - T_o)/G$ case in equation (1) we have:

$$Q_c = 763.6 + 9.32 \quad (\text{W})$$

So, at low values of $(T_i - T_o)/G$ the second term is small compared to the first term and even if the uncertainty of the second term is high, its impact on the overall uncertainty on η will be very small. But this is not the case for high values of $(T_i - T_o)/G$. Taking, for example, the values found in Table 1 for the high $(T_i - T_o)/G$ case and inserting them in equation (A-2):

$$\begin{aligned} Q_c &= 0.0239 \times 1008.8 \times 13.5 \\ &\quad + (0.0239 - 0.0234) \times 1008.8 \times (49.34 - (-14.40)) \\ Q_c &= 317.5 + 32.15 \quad (\text{W}) \end{aligned}$$

In this case, the second term is not negligibly small compared to the first one and measurement uncertainties in this second term become important (particularly the mass flowrate uncertainties). For example, the uncertainty on the flowrate difference $(\dot{m}_o - \dot{m}_i = 0.0005 \text{ kg/s})$ is approximately $\pm 0.0005 \text{ kg/s}$ $((0.015 \times 0.0239)^2 + (0.015 \times 0.0234)^2)^{0.5}$. Thus the uncertainty on the second term is approximately ± 100 percent yielding an uncertainty on Q_c of approximately 10 percent.

The uncertainty on G ($\pm 3\%$) is small compared to this value and this explains why the contributions of $\dot{m}_i$ and $\dot{m}_o$ to the overall uncertainty on η are high (in comparison to the contribution of G). Therefore at high values of $(T_i - T_o)/G$, the uncertainty on η becomes unacceptably high. The problem is even more severe at high flowrates because the uncertainty on the flowrate difference $(\dot{m}_o - \dot{m}_i)$ is higher at higher flowrates. This can be seen by comparing the size of the error bars in Fig. 6(c) (low flowrate) and Fig. 7(c) (high flowrate). This seems to indicate the need for more accurate mass flowrate measurements (better than ± 1.5 percent) if the overall uncertainty on η is to be reduced.

Outward Leakage. Also presented in Table 1 are the results of an uncertainty analysis performed on outward leakage data (for a high value of $(T_i - T_o)/G$). The major contributors to the overall uncertainty on η are the uncertainty on G and on T_L . Looking back at the numerator of equation (7) we have:

$$Q_c = \dot{m}_o C_p (T_o - T_i) + (\dot{m}_i - \dot{m}_o) C_p (T_L - T_i) \quad (\text{A-3})$$

where $T_L = (T_i + T_o)/2$

The problem encountered with the inward leakage case, that is, the increase in the uncertainty of the second term at high values of $(T_i - T_o)/G$, does not occur for the outward leakage case. At high values of $(T_i - T_o)/G$, the ΔT across the collector decreases because of increased heat losses from the collector so that T_L is close to T_i and thus the second term (and its associated uncertainty) is negligible versus the first one. Conversely, at low values of $(T_i - T_o)/G$, the difference between T_L and T_i increases, thus increasing the value of the second term but at the same time the first term increases due to the increase of $(T_o - T_i)$ and these effects counterbalance each other. In all, the outward leakage cases studied here, the relative uncertainty on η , ω_η/η , was below ± 6.5 percent for all values of $(T_i - T_o)/G$.

Inward-Outward Leakage Case. An uncertainty analysis was also performed on the inward-outward leakage data and the results are presented in Table 1. Looking back at equation (10), and as mentioned earlier the right-hand side of equation (10) is the same as the right-hand side of equation (5) (inward leakage case) except for the third term. Hence, the arguments used in the inward leakage case to describe the uncertainty on η , associated with the uncertainty of the first two terms, are equally applicable here. To be added to this already unacceptably high uncertainty is the uncertainty on the third term $(\dot{m}_{L_o} \cdot C_p \cdot (T_{L_o} - T_a))$ which increases the uncertainty on η even more at high values of $(T_i - T_o)/G$.

This paper is being distributed in reprint form by the Institute for Research in Construction. A list of building practice and research publications available from the Institute may be obtained by writing to the Publications Section, Institute for Research in Construction, National Research Council of Canada, Ottawa, Ontario, K1A 0R6.

Ce document est distribué sous forme de tiré-à-part par l'Institut de recherche en construction. On peut obtenir une liste des publications de l'Institut portant sur les techniques ou les recherches en matière de bâtiment en écrivant à la Section des publications, Institut de recherche en construction, Conseil national de recherches du Canada, Ottawa (Ontario), K1A 0R6.