Development and Validation of Flat-Plate Collector Testing Procedures

Report for March, 2007

Focus on Energy (FOE) supports solar thermal systems that displace conventional fuels by offering cash-back rebates that provide an incentive for residents to invest in this renewable energy technology. To be eligible for rebates, FOE requires solar collectors to be certified by the Solar Rating and Certification Corporation (SRCC). The certification program involves testing of the solar collectors in accordance with ASHRAE Standard 93-2003[1]. Currently, these tests are only provided in Florida (outdoors) by the Florida Solar Energy Center (FSEC).

Wisconsin's flat plate collector testing program will be done at Madison Area Technical College (MATC). The UW-Solar Energy Laboratory is assisting MATC personnel in establishing a suitable implementation of the ASHRAE test method. The UW further intends to identify alternative test methods that can be done indoors or under conditions that are more suitable to Wisconsin weather, but still provide the information required by the ASHRAE 93-2003 test. What follows is the sixth report of this activity.

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1. Determining the Heat Loss of a Flat Plate Collector with an Indoor Test

1.1. Introduction

There are three important parameters that describe the performance of solar thermal collectors: the efficiency at normal incidence with inlet temperature equal to ambient temperature $F_R(\tau\alpha)_n$, the heat loss coefficient F_RU_L, and the incidence angle modifier coefficient b_0. The performance parameters $F_R(\tau\alpha)_n$ and F_RU_L can be determined based on the collector efficiency curve obtained during collector tests. The group $F_R(\tau\alpha)_n$ is the y-intercept of the curve while F_RU_L is the slope. According to ASHRAE 93, 16 efficiency tests must be performed to generate the efficiency curve. The efficiency tests are all performed under near normal incidence conditions (Report 2, Chapter 3.1.4). An incidence angle modifier test as described in Report 2, Chapter 4 is used to determine the incidence angle modifier coefficient b_0. The incidence angle modifier test includes four or eight additional efficiency tests, for an altazimuth or fixed test mount, respectively. For a fixed test mount at an outdoor test facility this means that 24 efficiency tests must be conducted to derive the three mentioned collector performance parameters. As climatic conditions in Wisconsin limit the period of time suitable for collector testing in accordance with ASHRAE Standard 93 and as it is desirable to reduce the overall time effort necessary for the collector tests, alternative test methods should be evaluated.

Symons (1976) has described and conducted indoor tests for determining the overall heat loss coefficient of a solar flat-plate collector[2]. This test method does not need a solar irradiance simulator. Instead, hot water is circulated through the collector and the temperature drop is measured. From this information the overall heat loss coefficient, U_0, can be calculated. The coefficient U_0 is based on the mean fluid temperature in the collector. As will be shown below, the heat loss parameter F_RU_L can also be determined from these test results. The only difference is that the calculation is based on the fluid temperature at the collector inlet instead of the mean fluid temperature.

If F_RU_L can be determined by an indoor test, the number of required outdoor tests could be reduced from 24 to 8 tests. The incidence angle modifier test still must be performed outdoors to experimentally determine the incidence angle modifier coefficient b_0. However, the parameter $F_R(\tau\alpha)_n$ can be derived from the incidence angle modifier test alone without additional tests, as one or two of the tests determine the efficiency at normal incidence.

1.2. Test setup

An indoor test aimed at determining F_RU_L has been performed running hot water through the array at different inlet temperatures and two different flow rates. A fan is used to move air across the collector and the local air speed measured at 7 positions above the collector surface as shown in Figure 1.

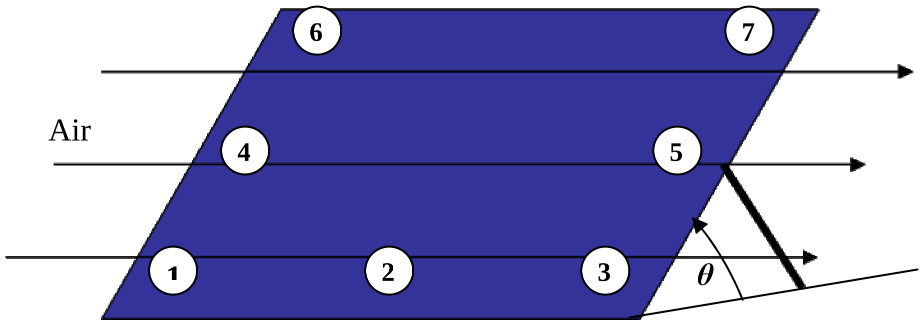

Figure 1 Air flow measurements across collector, θ=50.5°

The following air speed values have been measured parallel to the collector plane and the ground, 10 cm above the collector plane:

Table 1 Air speeds measured before testing

Position	Wind speed [FPM]	Wind speed [m/s]
1	250	1.3
2	500	2.5
3	550	2.8
4	950	4.8
5	400	2.0
6	0	0.0
7	220	1.1

The air flow provided by the fan was not parallel as indicated in Figure 1. An average value for the wind speed has been estimated by first averaging the measured values at the same horizontal line and then calculating the mean of these three values.

Table 2 Average air speed over collector

Positions	Average wind speed [m/s]
1, 2, 3	2.2
4, 5	3.4
6, 7	0.6
Average wind speed	**2.1**

The ceiling temperature of the test room has been measured during the last two tests and has been found to be constant and slightly above the ambient room temperature.

1.3. Calculations

Symons has used the following equation to calculate the overall heat loss coefficient.

$$U_0 = \frac{\dot{V}\rho c_p \left(T_i - T_o\right)}{A\left(\overline{T}_W - T_a\right)} \tag{1.1}$$

The variables presented in Table 3 must be measured or known for the calculations.

Table 3 Indoor test variables

Variable	Description
$\dot{V}$	Volume flow of water through the collector
ρ	Density of water at temperature $\overline{T}_W$, pressure of 101.3 kPa
c_p	Specific heat of water at temperature $\overline{T}_W$, pressure of 101.3 kPa
T_i	Collector inlet temperature
T_o	Collector outlet temperature
A	Collector gross area
$\overline{T}_W$	Mean water temperature (arithmetic mean of inlet and outlet)
T_a	Ambient temperature

Duffie and Beckman (2006) have defined the collector heat removal factor F_R based on the collector inlet fluid temperature:

$$F_R = \frac{\dot{m}C_p\left(T_o - T_i\right)}{A\left[S - U_L\left(T_i - T_a\right)\right]} \tag{1.2}$$

During the indoor test, the irradiance S upon the collector plane is approximately zero. This fact allows Equation (1.2) to be rewritten as:

$$F_R U_L = \frac{\dot{m}C_p\left(T_i - T_o\right)}{A\left(T_i - T_a\right)} \tag{1.3}$$

So the overall heat loss coefficient U_0 and the product $F_R U_L$ can be calculated from the same set of variables presented in Table 3. Only $F_R U_L$ will be calculated in the following test analysis as this is the efficiency parameter that must be reported according to ASHRAE Standard 93.

The thermal loss parameter $F_R U_L$ is calculated from steady state time periods of 3.5 minutes in duration. A time period was considered steady state if the 10 seconds average values of volume flow rate, inlet temperature, and outlet temperature remained constant within the limits listed in Table 4.

Table 4 Allowed variation for steady state measurements

Variable	**Allowed variation**
Inlet temperature	±0.1K
Outlet temperature	±0.1K
Ambient temperature	±0.1K
Volume flow rate	±0.5%

For time periods that met the steady state requirements defined above, the $F_R U_L$ was calculated by averaging the measured variables over 3.5 minutes and using these average values in Equation (1.3). The results are presented in Table 5.

1.4. Results

Table 5 Test results

Test No.	Average mass flow rate [kg/s]	Average temperature difference inlet - ambient [°K]	Average temperature difference inlet – outlet [°K]	Fan	$F_R U_L$ [W/m^2-K]
1	0.020	33.3	1.8	On	4.49
2	0.020	45.2	2.4	On	4.39
3	0.020	26.4	1.3	On	4.10
4	0.020	37.2	2	On	4.48
5	0.020	44.6	2.4	On	4.47
6	0.020	44.6	2.3	On	4.27
7	0.020	45.3	2.2	Off	4.02
8	0.009	37.2	3.7	On	3.67
9	0.009	44.8	4.4	On	3.57

The results are plotted in Figure 2.

1.5. Uncertainty Analysis

The following accuracies have been assumed for the measurements:

Table 6 Measurement errors

Variable	Uncertainty
Inlet temperature	± 0.2 K
Outlet temperature	± 0.2 K
Ambient temperature	± 0.2 K
Volume flow	± 1%
Collector gross area	±1.0%

An uncertainty analysis based on the uncertainties presented above has been performed with EES[3]. The results are presented as error bars in Figure 2.

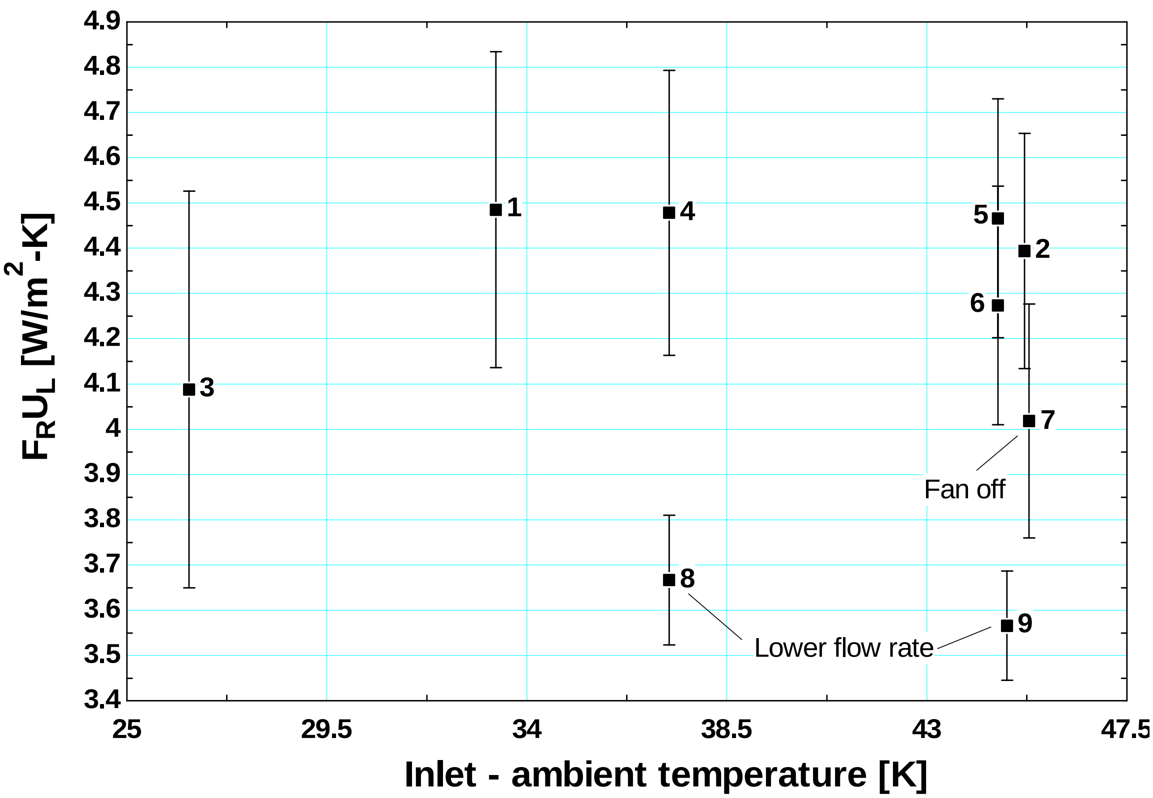

Figure 2 Overall heat loss coefficient vs. flow rate

1.6. Discussion

The tests with numbers 1 through 6 have been performed at the same flow rate as the preceding outdoor tests of the collector. The average value of F_RU_L of these tests is 4.4 W/m^2-K which is close to the value of 4.5 W/m^2-K obtained by the outdoor test.

For the range of temperatures used during testing, F_RU_L is assumed to be constant. The test results show an uncertainty in the obtained values for F_RU_L that is somewhat larger, but comparable to, the uncertainty in F_RU_L determined in outdoor tests. The highest calculated value is 4.49 W/m^2-K (test no. 1) and the lowest value is 4.10 W/m^2-K. The reason for this variation is the high sensitivity of $F_R U_L$ with respect to the measured temperature difference between inlet and outlet. A variation of only 0.1 K in temperature difference causes a change in $F_R U_L$ of about 0.2 W/ m^2-K for the test conditions of test no. 2.

Tests 8 and 9 have been conducted with a flow rate of 0.09 kg/s-m^2 instead of 0.20 kg/s-m^2. The average value obtained for $F_R U_L$ is 3.6 W/m^2-K. This value is 18% lower than the average value of the seven high flow rate tests.

The plot shows that the uncertainty of the test results is about ±0.6 W/m^2-K or ±14 % for the high flow tests (data points 1 – 7) and about ±0.25 W/m^2-K or ±7 % for the low flow tests (data points 8 and 9). Reducing the mass flow rate by 50% has decreased the uncertainty of the test result by 50%.

1.7. Conclusions

In case a fixed test mount is used to conduct the thermal efficiency test in accordance with ASHRAE Standard 93, 24 single efficiency tests are necessary. Determining the heat loss of the collector by the described indoor test can reduce the necessary outdoor tests by 16. However, the uncertainty of the test results is high. The reason for this is the high sensitivity of the heat loss parameter $F_R U_L$ with respect to the measured temperature difference in combination with the low absolute values for the temperature differences. The uncertainty of the results can be decreased by increasing the achieved temperature difference. This can be done by lowering the flow rate or increasing the surrounding air velocity. Another possibility is to measure the temperature difference between inlet and outlet directly with higher accuracy. ASHRAE Standard 93 prescribes an accuracy of 0.1 K for the measurement of temperature differences. The uncertainty of $F_R U_L$ obtained by the indoor test introduced in this report and the outdoor test in accordance with ASHRAE Standard 93 will be compared.

During the discussion of the indoor test method, another important question concerning the validity of the direct heat loss measurement has been raised. During operation the temperature of the plate is higher than the temperature of the fluid and the direction of the heat flow is therefore from the plate to the fluid. The bonding between the plate and the tubes has a significant influence on the efficiency of the collector. A bonding of low quality with low conductance decreases the heat transfer from the plate to the fluid. As a result the plate temperature is higher compared to a collector with a good bonding. The higher plate temperature increases the heat loss to the surroundings. As a result, a collector with bad bonding shows a higher heat loss. However, during the described indoor test hot water is circulated through the collector to heat the plate and measure the heat loss. The heat transfer is in opposite direction compared to the heat transfer during operation or outdoor testing. In this situation a bad bonding reduces the rate of heat transfer from the fluid to the plate. Consequently, the temperature of the plate is lower than it would be for a collector with good bonding. But lower plate temperature means less heat loss to the surroundings. Consequently, a collector with a bad bonding would show a lower heat loss than a collector with a good bonding during the indoor test. This result is just the opposite of the real situation. It will be evaluated in how far the quality of the bonding effects the temperature distribution and the test results of the indoor test.

[1] ANSI/ASHRAE Standard 93-2003, *Methods of Testing to Determine the Thermal Performance of Solar collectors.* ISSN 1041-2336, ASHRAE, Inc., 2003, 1791 Tullie Circle, Ne, Atlanta, GA30329

[2] Symons, J.G., *The Direct Measurement of Heat Loss From Flat-Plate Solar Collectors on an Indoor Testing Facility*, CSIRO, Melbourne, ISBN 0 643 00209

[3] EES: *Engineering Equation Solver* information is available at www.fchart.com.