Measurement of Fenestration Net Energy Performance: Considerations Leading to Development of the Mobile Window Thermal Test (MoWitt) Facility

We present a detailed consideration of the energy flows entering a building space and the effect of random measurement errors on determining fenestration performance. Estimates of error magnitudes are made for a passive test cell; we show that a more accurate test facility is needed for reliable measurements on fenestration systems with thermal resistance 2-10 times that of single glazing or with shading coefficients less than 0.7. A test facility of this type, built at Lawrence Berkeley Laboratory, is described. The effect of random errors in this facility is discussed and computer calculations of its performance are presented. The discussion shows that, for any measurement facility, random errors are most serious in nighttime measurements, and systematic errors are most important in daytime measurements. It is concluded that, for this facility, errors from both sources should be small.

Introduction

Many issues that affect the development and use of energy-efficient window and/or skylight systems require a quantitative knowledge of fenestration thermal performance under realistic conditions. Currently, average net energy costs/benefits are calculated based on the U-value and shading coefficient of fenestration. These calculations, often embedded in building simulation models such as DOE-2 [1] or BLAST [2], require numerous assumptions and approximations to specify conditions to which fenestration is subjected and their interaction with the adjacent building space. The method by which fenestration U-values should be measured is somewhat controversial [3], [4], [5]; some systems, such as fenestration with exterior venetian blinds, have poorly defined U-values. The validity of superposition of U-value and shading coefficient has been experimentally verified for only simple fenestration systems [6], [7]. In short, to go from measured U-values and shading coefficients to average net energy cost/benefit requires a theory with substantial physical content. Testing this theory requires measuring average net energy performance under conditions representative of use.

One way to make such measurements is to use a room-sized passive test cell with measured energy inputs. This technique, which has been used to study passive solar heating [8] and to test the predictions of BLAST [9], has two limitations: It is not sufficiently accurate for study high-performance (i.e., highly insulating or low-shading-coefficient) fenestrations, and the control volume emphasizes space loads rather than net heat flows, so it is difficult to isolate fenestration performance.

The technique can, however, be extended by improving its accuracy and changing the control volume to treat fenestration net heat flows correctly. These extensions lead to a facility quite different from an ordinary passive cell and uniquely suited to studying fenestration performance. Such a facility, the Mobile Window Thermal Test (MoWitt) Facility, has been built at Lawrence Berkeley Laboratory.

The need for measurement accuracy follows from the way fenestration systems are optimized. In general, optimal fenestration systems will have (if possible) an average net heat flow that satisfies the average heat demand of the building (e.g., energy-gaining fenestrations for a building with a heating demand); however, this must be achieved within the constraints of local thermal and visual comfort and (possibly) daylighting. The result of these often conflicting requirements is frequently that average net heat flows are small, either because all peak heat flows are small, or because daytime thermal gains cancel nighttime thermal losses through thermal storage. From a measurement standpoint, this requires either measuring a small signal or averaging the difference between two large signals. In either case good measurement accuracy is necessary.

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In this paper, an error analysis is developed for measuring the net heat flow through a fenestration system adjacent to a building space. Analysis results are applied to a hypothetical passive test cell and to the MoWiTT.

Fenestration Energy Flows in Sunlit Spaces

We consider a fenestration system, $F$, forming part of the envelope of a closed building space, and define a control volume with an imaginary surface, $E$, located, as shown in Fig. 1, an infinitesimal distance inside the envelope. We assume that $E$ has small holes through which air may pass (leaks) or through which climate control systems may move energy, and that these are sufficiently small or geometrically shielded so that we may neglect radiant or conducted energy transfer through them. It follows that the fenestration energy flow, $W$, is given by,

$$ W(t) = CV \frac{dT}{dt} - H(t) + I(t) + L_C(t). $$

$L_C$ is the rate at which the climate control system removes heat from the building space, and includes internal loads such as lights. All other energy flows are defined as flowing into the building space.

The fenestration energy flow, $W$, consists of three parts, $W = S_W + Q_R + Q_C$, where $S_W$ is the net transmitted solar energy, i.e., the transmitted visible and shortwave infrared radiation (direct and diffuse) less the transmitted outgoing radiation (from back-reflection inside the building space); $Q_R$ is the net thermal infrared radiant transfer between the fenestration and the inner surfaces of the space; and $Q_C$ is the heat transferred to or from the air by conduction/convection. The envelope heat flow, $H$, is purely conductive since surface $E$ was taken to lie inside the solid comprising the envelope. Considering the heat balance on the (infinitesimal) envelope layer inside, $E$, we find that

$$ H(t) = H_C(t) - Q_R(t) - S_W(t), $$

where $H_C$ is the heat flow to the air by conduction and convection. Integration over surface $E$ removed interreflections or radiative exchanges between different parts of the envelope.

The heat-balance equation for the air and other mass inside the building space, while similar in form to equation (1), differs in content:

$$ CV \frac{dT}{dt} = H_C(t) + Q_C(t) + I(t) - L_C(t). $$

It contains only $Q_R$, the conductive/convective part of the fenestration energy flow; the radiative and solar gain parts, $Q_R$ and $S_W$, enter only partially through $H_C$ as determined by equation (2). This shows the distinction between use of the control volume of Fig. 1, which emphasizes the net heat balance of the space, and the control volume corresponding to equation (3), which emphasizes the space load. In the latter case, the radiant part of fenestration heat flow is not directly contained; it appears in the analysis only to the extent that it drives heat to or from the air through $H$. Any parts of the radiant heat flow that go through $H$ rather than $H_C$ are not counted. Cavity back-reflection of solar-optical radiation is also undetected.

**Error Analysis**

Let us consider the effect of finite accuracy in measuring the terms on the right-hand side of equation (1). Assuming that the errors are random and uncorrelated, the fractional error in the fenestration energy flow is given by

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### Nomenclature

- $a =$ Infiltration rate (air changes per unit time)
- $A_E =$ Area of surface $E$, defined below (m$^2$)
- $A_G =$ Gross floor area of building in question (m$^2$)
- $A_S =$ Fenestration area illuminated by sunlight (m$^2$)
- $A_W =$ Area of fenestration (m$^2$)
- $C =$ Volume-weighted average of $\rho C_p$ for all thermal mass contained in $E$ (J/K)
- $C_p =$ Specific heat at constant pressure (J/kgK)
- $E =$ Denotes an imaginary surface lying just below the physical inner surface of the exterior envelope of $V$
- $f =$ Fluid flow rate (m$^3$/s)
- $H =$ Envelope heat flow across surface $E$ (W)
- $H_C =$ Heat flow by conduction/convection between the interior envelope and the air inside $E$
- $I =$ Heat transfer by infiltration into $V$ (W)
- $J_a =$ A reference solar intensity (W/m$^2$) incident on the structure and transmitted through single glazing
- $L_C =$ Rate of removal of energy from building space by climate-control systems (space load) (W). Negative $L_C$ is heating load
- $Q_C =$ Conductive/convective heat transfer from fenestration to interior air (W)
- $Q_R =$ Thermal infrared radiant heat transfer from fenestration to interior surfaces (W)
- $R =$ Relative thermal resistance of fenestration compared to that of single glazing, defined as $U_p/U$ (dimensionless)
- $R_E =$ Relative thermal resistance of envelope compared to that of single glazing (dimensionless)
- $[SC] =$ Shading coefficient of the fenestration
- $S_W =$ Net energy (inward-flowing less outward-flowing) cross-

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(8)

\[ \frac{\gamma}{\lambda} + \frac{\gamma}{\zeta} \left( \frac{3}{\kappa} \right) \frac{d}{dx} \left( \rho \frac{\partial \nu}{\partial \nu} \right) = 0 \]

The above equation is referred to as the Burgers' equation, which is a simplification of the Navier-Stokes equations. It describes the behavior of a fluid in terms of its velocity and pressure fields, and it is widely used in fluid dynamics.

(9)

\[ \rho \frac{d}{dt} \frac{\partial \nu}{\partial \nu} + \frac{\partial}{\partial x} \left( \mu \frac{\partial \nu}{\partial x} + \rho \nu \frac{\partial \nu}{\partial y} \right) = \rho f(x,y,t) \]

This equation represents the conservation of momentum in fluid dynamics. It shows how the change in momentum of a fluid element is equal to the net force acting on it, which includes the pressure gradient, shear stress, and body forces.

(10)

\[ \frac{\partial}{\partial x} \left( \rho \frac{\partial \nu}{\partial x} \right) + \frac{\partial}{\partial y} \left( \rho \frac{\partial \nu}{\partial y} \right) = \rho \frac{d^2 \nu}{dx^2} \]

This is a simplified form of the Navier-Stokes equations for incompressible fluids. It describes the horizontal component of the velocity field.

(11)

\[ \frac{\partial}{\partial t} \left( \rho \frac{\partial \nu}{\partial t} \right) + \frac{\partial}{\partial x} \left( \rho \frac{\partial \nu}{\partial x} \right) + \frac{\partial}{\partial y} \left( \rho \frac{\partial \nu}{\partial y} \right) = \nabla \cdot \tau \]

This equation represents the conservation of momentum in a more general form, including time dependence.

(12)

\[ \nabla \cdot \tau = 0 \]

This is the incompressibility condition, which states that the divergence of the stress tensor is zero for an incompressible fluid.

(13)

\[ \tau = \mu \left( \nabla \nu + \nabla \nu^T \right) \]

This equation represents the stress tensor in terms of the strain rate tensor, showing how stress is related to deformation.

(14)

\[ \rho \frac{d^2 \nu}{dt^2} = -\nabla \cdot \tau + \rho \nu \frac{d \nu}{dt} \]

This equation describes the motion of a fluid element under the influence of forces and deformation.
Table 1  Error sources in fenestration heat flow measurements

<table>
<thead>
<tr>
<th>Source</th>
<th>Contribution to ( \delta W/W )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Space Heat Content</td>
<td>( \frac{\sqrt{2} C \delta T_A}{U_0} ) t ( \delta T )</td>
</tr>
<tr>
<td>Envelope Conduction</td>
<td>( \frac{R}{U_0} ) (( \frac{\delta T}{H} ))</td>
</tr>
<tr>
<td>Infiltration</td>
<td>( \frac{R}{U_0} ) (( \frac{\delta T}{H} ))</td>
</tr>
<tr>
<td>Space Load</td>
<td>( \left( \frac{1}{W} \right) + \frac{1}{W} - \frac{A_S}{U_0} \delta T ) ( \frac{\delta T}{T_C} )</td>
</tr>
</tbody>
</table>

(b) Daytime

<table>
<thead>
<tr>
<th>Source</th>
<th>Contribution to ( \delta W/W )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Space Heat Content</td>
<td>( \frac{1}{[SC]} \left( \frac{V}{A_S} \right) \frac{\delta T}{L_C} )</td>
</tr>
<tr>
<td>Envelope Conduction</td>
<td>( \frac{\delta T}{H} )</td>
</tr>
<tr>
<td>Infiltration</td>
<td>( \frac{1}{[SC]} \left( \frac{V}{A_S} \right) \frac{\delta T}{L_C} )</td>
</tr>
<tr>
<td>Space Load</td>
<td>( \left( \frac{1 - \alpha}{W} \right) + \frac{1}{[SC]} \frac{A_S}{A_S} \frac{T}{L_C} )</td>
</tr>
</tbody>
</table>

Table 2  Estimated error source contributions to \( \delta W/W \) for an R-40 test cell

(a) Small Window

<table>
<thead>
<tr>
<th>Source</th>
<th>Nighttime</th>
<th>Daytime</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air Heat Content</td>
<td>0.08 ( R ) ( \frac{\delta T}{A} )</td>
<td>0.01 (( \frac{1}{[SC]} )) ( \frac{\delta T}{A} )</td>
</tr>
<tr>
<td>Envelope Conduction</td>
<td>1.0 ( \frac{\delta T}{H} )</td>
<td>0.4 (( \frac{\delta T}{H} ))</td>
</tr>
<tr>
<td>Climate Control System</td>
<td>(1 + R) ( \frac{\delta T}{L_C} )</td>
<td>0.6 + 0.003 (( \frac{1}{[SC]} )) ( \frac{\delta T}{L_C} )</td>
</tr>
</tbody>
</table>

(b) Large Window

<table>
<thead>
<tr>
<th>Source</th>
<th>Nighttime</th>
<th>Daytime</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air Heat Content</td>
<td>0.01 ( R ) ( \frac{\delta T}{A} )</td>
<td>0.002 (( \frac{1}{[SC]} )) ( \frac{\delta T}{A} )</td>
</tr>
<tr>
<td>Envelope Conduction</td>
<td>0.15 ( \frac{\delta T}{H} )</td>
<td>0.4 (( \frac{\delta T}{H} ))</td>
</tr>
<tr>
<td>Climate Control System</td>
<td>(1 + 0.15 R) ( \frac{\delta T}{L_C} )</td>
<td>0.6 + 0.003 (( \frac{1}{[SC]} )) ( \frac{\delta T}{L_C} )</td>
</tr>
</tbody>
</table>

With these equations, one can calculate the individual terms in equation (4), which are shown in Table 1. These are then added in quadrature (i.e., as in equation (4)) to obtain \( \delta W/W \).

Error Estimates for a Passive Test Cell

We first consider the accuracy attainable using a passive test cell 2.4 m x 3 m x 2.4 m (8ft x 10ft x 8ft) high, with a fenestration system mounted in a short side and facing south. A residential-sized fenestration of 1 m² area and a large fenestration filling the entire 2.4 m² area are considered. The (dimensionless) \( R \) value of the envelope is taken to be 40. It is assumed that the cell is so tightly constructed that the infiltration rate is negligible. The magnitudes of the potential error sources are shown in Table 2. We note that for the small window, the fenestration area is 17 percent of the floor area, which, while high, is reasonable for residential buildings. The large window is 80 percent of the floor area, atypically large for most construction.

The table shows the roughly equal importance of accuracy in measuring the climate-control system performance and the envelope heat conduction. (It is assumed that the air temperature can be maintained constant to an accuracy sufficient to make the term arising from air heat content negligible.) If we assume that both \( H \) and \( L_C \) have equal percentage error, then in the nighttime heating mode measuring a residential-sized single-glazed window to 10 percent accuracy requires a 6 percent measurement of \( H \); for an R-10 system, one would need 0.6 percent, which is probably impossible. For the large window, the situation is somewhat better; a 10 percent measurement of \( H \) would permit nighttime measurements on a system with \( R = 4 \). A measurement of \( H \) is equally important for daytime measurements on both size fenestrations. If \( L_C \) could be made very much more accurate than \( H \), then the required measurement accuracies for \( H \) become 10 percent and 1 percent, respectively.

This makes window performance measurements awkward. It would appear that one can accurately study only low-thermal-resistance, residential-sized windows with a simple passive cell. To study high-\( R \) systems that are of interest for improving building energy efficiency, one must study large windows. This compromises the study of realistic performance, since the glazing-to-floor-area ratio will be atypically high, (exaggerating the importance of radiative heat transfers).

These conclusions are for nighttime heat flows. A model that neglects thermal storage cannot predict daytime heat flows and we have, therefore, used the value \( \alpha = 0.4 \), which is estimated from the computer calculations described below, in the daytime estimates in Table 2. In addition, the simplified model is one-dimensional, whereas daytime heat flows arise from highly inhomogeneous distributions of solar flux on the interior surfaces. Spatial inhomogeneities are present to a lesser degree in the nighttime heat flows, due to radiative coupling to the fenestration.

These limitations of the model mean that Table 2 should be interpreted as presenting approximate lower bounds on errors. Model refinements should not greatly affect the magnitudes of the peak-condition terms dominating the coefficients in equation (4); however, effects left out of the model which constitute additional sources of error will add terms to the equation. Because of its root-sum-of-square form, these, if significant, can only increase the error.

A Specialized Facility For Measuring Fenestration Energy Flow

The foregoing makes clear the capabilities necessary in a facility designed to measure fenestration performance. First, it should measure fenestration performance under conditions representative of actual use: The fenestration should be exposed to outdoor weather conditions, since the combined effects of wind and radiation from the sun, sky, and ground...
The Mobile Window Thermal Test (MWT) facility, located at Lawrence Berkeley National Laboratory, is shown in FIG. 2. It consists of one or more mobile measurement modules, each with a central instrumentation module shown in FIG. 2. If desired, one or more modules can be accommodated in a single, fixed instrumentation module.

The instrumentation module contains a part of the thermal circuitry necessary for measurement, along with a data collection system. Each module is connected to the central instrumentation module through a data collection system. The data collection system allows the modules to be combined to form a single, integrated system.

Second, the module housing. The MWT can be accommodated in a building or in a mobile unit. The module should be designed to be transportable and to accommodate different measurement scenarios.

Finally, the module should be designed to be durable and to accommodate different measurement scenarios. The module should be designed to be transportable and to accommodate different measurement scenarios.
identical test rooms, each with a removable exterior wall and roof panel. This allows direct comparative measurements between horizontal or vertical fenestration systems exposed to the same exterior weather conditions. Varying the climate is achieved by moving the MoWiTT.

Realistic interior conditions are achieved by making the test room dimensions and construction as nearly like those of a room as possible. The interior dimensions—2.44 m parallel to the removable wall, by 3.05 m perpendicular to it, 2.34 m high—provide a space of the correct height and reasonable proportions, although smaller than typical of a residence. The walls are of plywood-faced polyurethane panels, providing a thermal time constant similar to light-frame residential construction. Thermal mass can be added to simulate a higher-mass structure. Wall, ceiling, and floor surface treatments may be varied to achieve the correct emissivity and reflectivity, or to study the effect of these parameters on fenestration performance. The climate-control system for each test room is self-contained and may supply heating and cooling.

After realism, the key consideration in the MoWiTT design was measurement accuracy. Since both high-resistance and low-shading-coefficient fenestration systems are of interest, the ability to measure the performance of a 1 m² fenestration system with $R = 10$ or [SC] = 0.1 to an accuracy of 10 percent was a design goal. Fenestration systems with claimed $R$ or [SC] values in this range have already appeared on the market (e.g., insulating shutters and reflective venetian blinds). From the standpoint of national energy consumption, extremely small changes in the average $R$ or [SC] of the nation’s building stock have a significant impact; thus, it is of interest to determine whether high-$R$ or low-[SC] devices actually perform as expected. In addition, questions about the difference in performance between two different fenestrations with considerably lower $R$ value (for example, low-emissivity sealed-insulating-glass units, for which $R = 3$, with different values of emissivity) will require a comparable degree of accuracy.

Experimental flexibility is achieved by a large data-recording capacity and a flexible computer system for collecting and manipulating data from up to 150 sensors from each test room, with an additional 50 sensors per room mountable on the fenestration’s exterior side. These are connected through a multiplexer to an LSI-11 computer. Data from temperature sensors, anemometers, radiometers, or other instrumentation may be collected and recorded on disc. From the field, data may be sent to the laboratory either on floppy disc or by telephone. The computer can also control devices inside the test rooms (for example, blinds during an experiment on window management) or modify chamber or guard conditions.

**Measurement Accuracy in the MoWiTT**

Examining the error sources for the passive cell in Table 2 (which is the same size as a MoWiTT test room) points up the magnitude of the measurement accuracy problem. Even with the high level of envelope insulation, a 1 percent measurement accuracy on the area-integrated envelope heat flow is necessary for nighttime measurements. Considering that heat fluxes are spatially inhomogeneous due to radiation and convection effects, it seems unlikely that such accurate measurements could be made.

This problem is solved in the MoWiTT by surrounding the test rooms with a guard plenum through which controlled-temperature air is circulated as shown in Fig. 3. This decouples the envelope heat flow from the external temperature and greatly reduces its magnitude during nighttime measurements. It also makes all envelope surfaces (other than that containing the test sample) effectively interior surfaces, which better simulates commercial and residential spaces (other than corner rooms) than does a passive cell. The contribution to the fractional error in the fenestration heat flow due to $H$ becomes:

$$\frac{\delta W}{W} = \frac{R_R}{R_E} \left( \frac{A_E}{A_W} \right) \frac{\Delta T_G}{\Delta T} \left( \frac{bH}{H} \right),$$

where $\Delta T_G$ is the temperature difference between the guard air and the test room air. The sensitivity of the fractional error ($\delta W/W$) to the heat flow measurement accuracy ($\delta H/H$) is reduced by a factor $(\Delta T_G/\Delta T)$. By maintaining the guard temperature close to the test room air temperature, we make this factor small. We have taken it to have a value of 0.1 in making error estimates.

The guard reduces the effect of errors from a number of sources by the same factor. Table 3 summarizes the contribu-

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**Table 3: Estimate fractional error magnitudes in MoWiTT**

<table>
<thead>
<tr>
<th>Source</th>
<th>Nighttime</th>
<th>Daytime</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air Heat Content</td>
<td>$\frac{F_c}{\tau U_0} \frac{R}{A_w} \frac{\Delta T}{T_A}$</td>
<td>$\frac{F_c}{\tau U_0} \frac{R}{A_w} \frac{\Delta T}{T_A}$</td>
</tr>
<tr>
<td>Envelope Heat Flow</td>
<td>$\alpha (\frac{T_H}{T})$</td>
<td>$\alpha (\frac{T_H}{T})$</td>
</tr>
<tr>
<td>Infiltration</td>
<td>$\frac{C}{U_0} (\frac{\Delta T}{T_A}) R \delta a$</td>
<td>$\frac{C}{U_0} (\frac{\Delta T}{T_A}) R \delta a$</td>
</tr>
<tr>
<td>Climate Control System</td>
<td>$(1 + \frac{R}{A_w} \frac{\Delta T}{T_A}) \frac{\delta H}{L_c}$</td>
<td>$(1 + \frac{R}{A_w} \frac{\Delta T}{T_A}) \frac{\delta H}{L_c}$</td>
</tr>
</tbody>
</table>

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**Fig. 3** MoWiTT guard system, showing circulation of forced-flow, temperature-controlled air around the two test rooms.
In the concluding discussion, we concentrated on the following computer conditions of object recognition and time-frequency processing.

- **Computer Conditions:**

  - The conditions of object recognition and time-frequency processing were considered.
  - A model of computer recognition and time-frequency processing was developed.
  - The conditions of object recognition and time-frequency processing were evaluated.
  - The model was validated against real-world data.

- **Conclusion:**

  - The conditions of object recognition and time-frequency processing will continue to improve.
  - Further research is needed to develop more sophisticated models.

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(1) \[ L(t) = \sum_{k} a_k \delta(t - kT) \]

(2) \[ L(t) = \frac{1}{T} \int_{t-T/2}^{t+T/2} L(\tau) d\tau \]

(4) \[ L(t) = \frac{1}{T} \int_{t-T/2}^{t+T/2} L(\tau) d\tau \]

(5) \[ L(t) = \frac{1}{T} \int_{t-T/2}^{t+T/2} L(\tau) d\tau \]

(6) \[ L(t) = \frac{1}{T} \int_{t-T/2}^{t+T/2} L(\tau) d\tau \]
the daytime, which is the origin of the value of 0.4 used for $\alpha$ in the simplified discussion above. Both curves for the MoWITT and the $L_C(t)$ curve for the passive cell were multiplied by the 5 percent assumed accuracy to produce the time-dependent absolute errors, $\delta L_C(t)$ and $\delta H(t)$. For the passive cell, the values of $H(t)$ during daylight hours were corrected for the effects of the moving patch of sunlight, as described above. These are shown as points in Fig. 4 (e), with the derived errors $\delta H(t)$ shown as error bars on the points. The points show sizable deviations from the BLAST-calculated curve (assumed to be the true value), which are considerably larger than expected for random errors. This is due to the incorrect weighting of essentially point measurements of the wall heat flux as the patch of direct sunlight moves around the wall. Each small heat flow sensor must be taken to represent a much larger area of the envelope; if direct sunlight strikes the sensor but only partially illuminates the envelope area that the sensor represents, then that entire area is taken to have the high absorbed flux characterizing the direct sunlight. Conversely, if the sunlight misses the sensor but partially illuminates the area, then the area will be characterized by a much lower absorbed flux than the true one. Only the size of the deviations is significant; a different sun angle or arrangement of the sensor grid would produce a different pattern of deviations from the curve—possibly even in the opposite direction. Whether these deviations would cancel out in measurements taken over a number of days depends on the arrangement of the sensor grid and the range of sun angles in the measurement period; it can not be assumed a priori. This graphically demonstrates the type of systematic error that may arise in daytime measurements made with an inadequate measurement system.
The flow diagram in the figure shows the steps involved in the measurement process of the WMIT. The diagram illustrates the sequence of events from data collection to final measurement output. Each step is represented by a box, with arrows indicating the flow of data and processes.

**References**


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