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MEASUREMENT OF FENESTRATION NET ENERGY PERFORMANCE: CONSIDERATIONS LEADING TO DEVELOPMENT OF THE MOBILE WINDOW THERMAL TEST (MoWiTT) FACILITY

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ABSTRACT

A detailed consideration of the energy flows entering the energy balance on a building space and the effect of random measurement errors on the determination of fenestration performance is presented. Estimates of the error magnitudes are made for a passive test cell and it is shown that a more accurate test facility is necessary for reliable measurements on fenestration systems with thermal resistance in the range 2-10 times that of single glazing or shading coefficient less than 0.7. A test facility of this type, the MoWiTT, which has been built at Lawrence Berkeley Laboratory, is described. The effect of random errors in the MoWiTT is discussed and computer calculations of its performance are presented. The discussion shows that, for any measurement facility, random errors are most serious for nighttime measurements, while systematic errors are most important for daytime measurements. It is concluded that, for the MoWiTT, errors from both sources are expected to be small.

NOMENECLATURE

a	Infiltration rate (air change per unit time)
B	Shading coefficient of the fenestration.
C	Volume-weighted average of ρC_p for all thermal mass contained in E (J/K).
C_p	Specific heat at constant pressure (J/kg.K).
E	Denotes an imaginary surface lying just below the physical inner surface of the exterior envelope of V; also, the area of that surface.
f	Fluid flow rate (m^3/s).
F	Area of fenestration (m^2).
F′	Fenestration area illuminated by sunlight.
G	Gross floor area of the building in question.
H	Envelope heat flow across surface E (W).
H_c	Heat flow by conduction/convection between the exterior envelope and the air inside E.
I	Heat transfer by infiltration into V (W).
J_o	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 system (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 radiative heat transfer from fenestration to interior surfaces (W).
R	Dimensionless thermal resistance of fenestration, defined as U_o/U.
R_E	Dimensionless thermal resistance of envelope.
S_w	Energy leaving the innermost surface of the fenestration as radiation in the visible and solar infrared bands (W).
T	Temperature (K).
T_A	Weighted mean temperature of all material inside E.

T_i Inlet fluid temperature.

T_e Exit fluid temperature.

t Time (s).

U Thermal transmittance (W/m^2K).

U_o Thermal transmittance of single glazing.

V Volume enclosed by surface E (m^3).

W Energy flow rate through the fenestration (W).

z_i Internal load per unit floor area (W/m^2).

α Fraction of solar energy incident on interior building envelope surface that flows across E.

δ Operator denoting "measurement uncertainty in"; e.g., δW denotes measurement uncertainty in W.

ΔT Difference between interior and exterior air temperatures (K).

ΔT_G Difference between interior and guard air temperatures.

ΔT_S Difference between interior air temperature and exterior sol-air temperature.

ϵ An infinitesimal distance.

ρ Density (kg/m^3).

ξ Parameter accounting for thermal lags between fenestration/envelope heat flows and space load.

$\tilde{\xi}$ Fraction of exterior envelope in sunlight.

τ Data sampling time period (s).

INTRODUCTION

There is a wide range of issues relating to the development and utilization of energy-efficient fenestration (i.e., window and/or skylight) systems which require a quantitative knowledge of fenestration thermal performance under realistic conditions. The current method of dealing with these issues utilizes calculations of average net energy costs/benefits which are based on the U-value and shading coefficient of the fenestration. These calculations, which are often embedded in building simulation models such as DOE-2 (1) or BLAST,(2) require numerous subsidiary assumptions and approximations to specify the actual conditions to which the fenestration is subjected and the way in which these interact with the adjacent building space. The method by which fenestration U-values should be measured is somewhat controversial,(3),(4),(5) and some systems, such as fenestrations with exterior venetian blinds, do not have a well-defined U-value. The validity of superposition of U-value and shading coefficient has been experimentally verified only for 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. To test this theory requires the ability to measure average net energy performance of fenestration systems under conditions representative of actual use.

An obvious method of making such measurements might be the use of a room-sized passive test cell with measured energy inputs to test the fenestration performance. This technique has been used, for example, to study passive solar heating (8) and to test the predictions of BLAST.(9) As commonly employed, this method has two limitations: it is not sufficiently accurate for studying high-performance (i.e., highly-insulating or low-shading-coefficient) fenestrations, and it employs a control volumne that emphasizes space loads rather than net heat flows, which makes it difficult to disentangle the fenestration performance.

The technique can, however, be extended by improving its accuracy and changing the control volume to treat fenestration net heat flows correctly. When properly made, these extensions have such major consequences that the resulting facility is quite different from an ordinary passive cell and is uniquely suited to the study of fenestration performance. Such a facility has been built at Lawrence Berkeley Laboratory. It is called the Mobile Window Thermal Test (MoWiTT) Facility, and its characteristics and expected performance are also presented.

The need for measurement accuracy follows from the way in which fenestration systems are optimized. In general, the optimal fenestration system will have (if possible) an average net heat flow which 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) utilization of daylight. The result of these often conflicting requirements is frequently that average net heat flows are kept small, either because all peak heat flows are made small, or because cancellation between daytime thermal gains and nighttime thermal losses is achieved through the use of thermal storage. From a measurement standpoint, this requires either measuring a small signal or averaging the difference between two large signals, which immediately raises the question of accuracy and sources of error.

In this paper, an error analysis is developed for measurement of the performance of a fenestration system adjacent to a building space. The results of the analysis 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 from energy conservation that the fenestration energy flow, W, is given by

$$W(t) = CV\frac{dT}{dt} - H(t) - I(t) + L_C(t). \quad (1)$$

(L_C is the rate at which heat is removed from the building space by the climate control system and includes internal loads such as lights. All other energy flows are defined as positive flowing into the building space.)

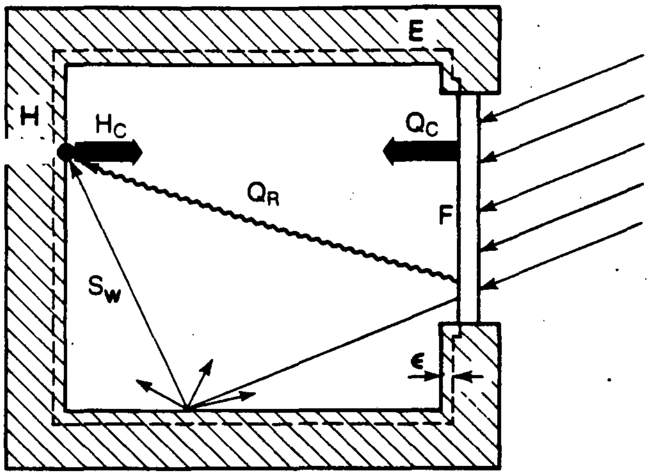


Figure 1. Components of Fenestration Energy Flow. Boundary E of control volume is located on infinitesimal distance ϵ inside wall-surface and completely encloses volume except for fenestration F. Long-wave thermal radiation is indicated by wavy arrow, conductive/convective heat transmition by heavy arrows and solar-optical radiation by light arrows. The absorbed solar radiation S_W is shown as first having undergone diffuse reflection from an interior surface. All heat flows are area-integrated.

It is instructive to consider some of the terms in this equation. 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 short-wave 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 a purely conductive flow since the surface E was taken to lie inside the solid comprising the envelope. If we consider the heat balance on the (infinitesimal) envelope layer inside E, we find that

$$H(t) = H_C(t) - Q_R(t) - S_W(t), \quad (2)$$

where H_C is the heat flow to the air by conduction and convection. Note that integration over the surface E, has 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 Eq. (1), is quite different in content:

$$CV\frac{dT}{dt} = H_C(t) + Q_C(t) + I(t) - L_C(t) . \quad (3)$$

It contains only Q_C, the conductive/convective part of the fenestration energy flow; the radiative and solar gain parts, Q_R and S_W, enter only partially and indirectly through H_C as determined by Eq. (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 Eq. (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_C. Any parts of the radiant heat flow which go through H rather than H_C will not be counted. Cavity back-reflection of solar-optical radiation will also go undetected.

ERROR ANALYSIS

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

$$\frac{\delta W}{W} = \left\{ \left[\frac{\delta(CV\frac{dT}{dt})}{W}\right]^2 + \left[\frac{H}{W}(\frac{\delta H}{H})\right]^2 + \left[\frac{I}{W}(\frac{\delta I}{I})\right]^2 + \left[\frac{L_C}{W}(\frac{\delta L_C}{L_C})\right]^2 \right\}^{1/2} , \quad (4)$$

where δW denotes the error in W, and similarly for the other quantities in the equation. The terms on the right-hand side of this equation arise from the heat capacity of the air (etc.) inside the building space, envelope heat conduction, infiltration, and climate-control system.

In order to estimate the magnitudes of the various terms in Eq.(4), we consider a simple model of the building space. We first parameterize the fenestration heat flows using (for nighttime heat loss) U_0, the U-value for single glazing, a dimensionless thermal resistance, R (see nomenclature), the fenestration area, F, and the inside-outside air temperature difference, ΔT:

$$W = -\frac{F}{R}U_0\Delta T . \quad (5a)$$

Similarly, for the daytime heat flow we use the shading coefficient, B, the heat flux through single glazing (solar heat gain factor), J_0, and the fenestration area receiving direct sunlight, F':

$$W = BJ_0F' . \quad (5b)$$

For simplicity, we neglect the comparatively small $U\Delta T$ term when the fenestration is in the solar gain mode. Nighttime envelope heat flows are analogously defined, neglecting the effects of thermal lags:

$$H = -E\frac{U_0}{R_E}\Delta T , \quad (6a)$$

where E is the total envelope area excluding the fenestration and R_E is the dimensionless envelope resistance. We assume that in the daytime envelope heat flow is dominated by fenestration heat gain, a fraction, α, of which flows into the envelope rather than into the air of the building space:

$$H = -\alpha BJ_0F' . \quad (6b)$$

Infiltration is parameterized using the air exchange rate per unit time, a:

$$I = -CVa\Delta T \ . \qquad (7)$$

Finally, the heat transferred by the climate control system is taken at nighttime to be

$$L_C = \xi(W+H) + I + z_I G \ , \qquad (8a)$$

where the parameter ξ is included to account for thermal lags, z_I is the internal load per unit floor area (from lights, etc.), and G is the gross floor area. The daytime space load is taken to be

$$L_C = (1-\alpha)BJ_0F' + \xi\Delta T_S(\frac{U_0}{R_E})E + z_I G \ . \qquad (8b)$$

Here ΔT_S is the temperature difference based on the sol-air temperature and ξ is the fraction of the envelope illuminated by sunlight.

Because the mean temperature of the air (and other thermal mass) inside the building space varies with time and is sampled only at finite intervals, there is an uncertainty associated with its heat content given by

$$\delta(CV\frac{dT}{dt}) = \frac{\sqrt{2}CV\delta T_A}{\tau} \ , \qquad (9)$$

where τ is the sampling period and δT_A is the RMS error for an individual measurement of T_A.

With these equations one can calculate the individual terms in Eq. (4), which are shown in Table 1. These are then added in quadrature 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 high (8 ft x 10 ft x 8 ft high), with a fenestration system mounted in a short side and facing south. A residential-sized fenestration of 1 m^2 area and a large fenestration filling the entire 2.4 m square are considered. The R value of the envelope is taken to be 40 and it is assumed that the cell is so tightly constructed that the infiltration rate is negligible. The magnitudes of the potential error sources are then shown in Table 2. We note that for the small window the fenestration area is 17% of the floor area, which, while high, is in a reasonable range for residential buildings. The large window is 80% of the floor area, which is atypically large for most kinds of construction.

The roughly equal importance of accuracy in measuring the climate-control system performance and the envelope heat conduction immediately emerges from the table. In the nighttime heating mode, in order to measure a residential-sized single-glazed window to 10% accuracy requires a 6% measurement of H; for an R-10 system one would need 0.6%, which is probably not possible. For the large window the situation is somewhat better; a 10% 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.

Table 1. Error Sources in Fenestration Heat Flow Measurement.

Source	Contribution to $\delta W/W$: (a) Nighttime	(b) Daytime
Space Heat Content	$R(\frac{V}{F})\frac{\sqrt{2}\ C\ \delta T_A}{U_0\ \tau\ \Delta T}$	$(\frac{1}{B})\ (\frac{V}{F'})\ \frac{\sqrt{2}\ CV\ \delta T_A}{J_0\tau}$
Envelope Conduction	$\frac{R}{R_E}(\frac{E}{F})(\frac{\delta H}{H})$	$\alpha(\frac{\delta H}{H})$
Infiltration	$R(\frac{V}{F})\frac{Ca}{U_0}(\frac{\delta a}{a})$	$(\frac{1}{B})(\frac{V}{F'})\frac{Ca\Delta T}{J_0}(\frac{\delta a}{a})$
Space Load	$\left\{\xi[1 + \frac{H}{W}] + \frac{I}{W} - R(\frac{G}{F})\frac{z_I}{U_0\Delta T}\right\}(\frac{\delta L_C}{L_C})$	$\left\{(1-\alpha) + \frac{I}{W} + (\frac{1}{B})\xi(\frac{E}{F'})\frac{\Delta T_S U_0}{R_E J_0} + (\frac{1}{B})(\frac{G}{F'})\frac{z_I}{J_0}\right\}(\frac{\delta L_C}{L_C})$

Table 2. Estimated Error Source Contributions to $\delta W/W$ for an R-40 Test Cell.

Source	(a) Small Window: Nighttime	(a) Small Window: Daytime	(b) Large Window: Nighttime	(b) Large Window: Daytime
Air Heat Content	$0.08R\ \delta T_A$	$0.01(\frac{1}{B})\ \delta T_A$	$0.01R\ \delta T_A$	$0.002(\frac{1}{B})\ \delta T_A$
Envelope Conduction	$1.0R(\frac{\delta H}{H})$	$0.4(\frac{\delta H}{H})$	$0.15R(\frac{\delta H}{H})$	$0.4(\frac{\delta H}{H})$
Climate Control System	$(1 + R)(\frac{\delta L_C}{L_C})$	$[0.6 + 0.02(\frac{1}{B})](\frac{\delta L_C}{L_C})$	$(1 + 0.15R)(\frac{\delta L_C}{L_C})$	$[0.6 + 0.003(\frac{1}{B})](\frac{\delta L_C}{L_C})$

This creates awkwardness when measuring window performance. Using a simple passive cell, it would appear that for residential-sized windows one can accurately study only low-thermal-resistance fenestration systems. To study the high-R systems which are of interest for improving building energy-efficiency, one must study large windows. This compromises the aim of studying realistic fenestration performance, since the glazing-to-floor area ratio will be atypically high (and therefore the importance of radiative heat transfers will be exaggerated).

These conclusions arise from the nighttime heat flows. A model which neglects thermal storage effects cannot adequately deal with daytime heat flows; in the above it has been assumed that $\alpha = 0.4$, which is a value made plausible by more detailed calculations presented below. In addition, the simplified model is purely one-dimensional, whereas the daytime heat flows arise from highly inhomogeneous distributions of solar flux on the interior surfaces. Spatial inhomogeneties are also present to a lesser degree in the nighttime heat flows, due to the radiative coupling to the fenestration.

These limitations of the model mean that Table 2 should be interpreted as presenting approximate lower bounds on the errors: effects left out of the model may add additional error, but will not greatly reduce those sources identified in the table.

A SPECIALIZED FACILITY FOR MEASURING FENESTRATION ENERGY FLOW

The foregoing considerations make clear the capabilities which a facility designed to measure fenestration performance should have. First, it should measure fenestration performance under conditions as representative of actual use as possible. This means that the fenestration should be exposed to outdoor weather conditions, since the combined effects of wind and radiation from the sun, sky and ground cannot be adequately simulated in the laboratory. It should be possible to measure fenestrations in different orientations and climates. The interior space should be room-like, with the correct height (since convective processes do not scale) and have a ratio of fenestration dimensions to room dimensions reasonably like those in a building (so that radiative processes have the correct weight). Surface reflectivities and emissivities on the interior should also be similar to those in a building, and it would be preferable to have them be variable. The fraction of solar-optical radiation absorbed in the interior envelope surfaces which is promptly transferred into the air should be comparable to that in a building. This means that the envelope should have a building-like thermal time constant, which ideally should be variable. The air temperature in the space should be kept within a reasonable comfort range, and humidity and forced-air velocities should be in a range representative of a building.

Second, the net energy flow, W, through the fenestration should be measurable with a time constant similar to the intrinsic response of the fenestration, i.e., very short. This means that the air infiltration rate must be very small or accurately measured, and heat added to or removed from the air by the climate-control system should be accurately monitored. Internal loads, if present, should be accurately measured. The area-integrated conductive heat flow through the interior surface should be accurately determined. The mean temperature of the air and any interior thermal mass should also be measured.

Third, it should be possible to do a wide variety of experiments in the facility, in order to relate the fenestration net energy flows to explanatory variables such as temperatures, solar intensities and wind speeds

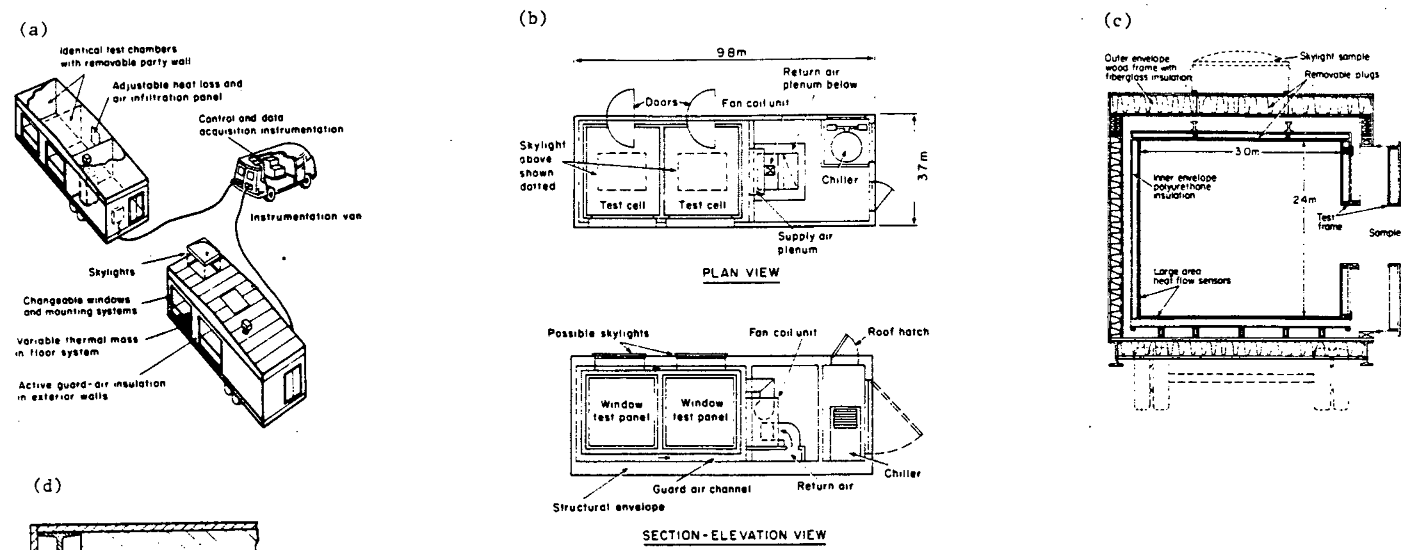
THE MOBILE WINDOW THERMAL TEST (MoWiTT) FACILITY

A measurement facility approximating these requirements has been built at Lawrence Berkeley Laboratory. It is called the MoWiTT (Mobile Window Thermal Test) facility and is shown in Fig. 2. It consists of one or more mobile measurement modules, together with a central instrumentation van for data collection. Each module contains a pair of identical test rooms, each with a removable exterior wall and roof panel. This allows direct comparative measurements between either horizontal or vertical fenestration systems exposed to the same exterior weather conditions. A variable climate is achieved by moving the MoWiTT to the climate of interest.

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 of 2.44 m parallel to the removable wall by 3.05 m perpendicular to it by 2.34 m high provide a space of the correct height and reasonable proportions, although the room is smaller than typical for a normal residence. The walls are of plywood-faced polyurethane panels, providing a thermal time constant similar to light-frame residential construction. The room is designed to permit the addition of thermal mass for simulation of higher-mass structures. Wall, ceiling, and floor surface treatments may be varied to achieve the correct emissivity and reflectivity, or, alternatively, to study the effect of these parameters on the fenestration performance. The climate-control system for each test room is self-contained and may supply either heating or 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-m^2 fenestration system with R = 10 or B = 0.1 to an accuracy of 10% was a design goal.

Experimental flexibility is achieved by having a large data-recording capacity together with a flexible computer system for collecting and manipulating the data. Provision has been made for bringing signals from up to 150 sensors out of each test room, with an additional 50 sensors per room mountable on the exterior side of the fenstration. These are connected through a multiplexer to an LSI-11 computer. Data from temperature sensors, anemometers, radiometers, or other instrumentation may be collected. The data are recorded on disc. When in the field, data may be sent back to the laboratory either on floppy disc or by telephone. The computer may also be used to control devices inside the test rooms (for example, the operation of blinds during an experiment on window management) or to modify the chamber or guard conditions.



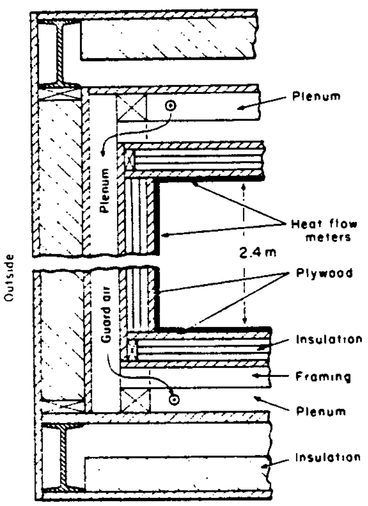


Figure 2. Design of the Mobile Window Thermal Test (MoWiTT) facility. (a) Planned field configuration. (b) Layout of a test module. (c) Cross-section through the center of a test chamber, showing mounting of alternative window or skylight systems. (d) Detailed envelope cross-section.

MEASUREMENT ACCURACY IN THE MoWiTT

Let us consider how the MoWiTT design accuracy is achieved. Examination of 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 problem. Even with the high level of envelope insulation, a 1% measurement accuracy on the area-integrated envelope heat flow would be necessary for nighttime measurements. Considering that heat fluxes will be spatially inhomogeneous, due to the effects of radiation and convection, it seemed highly unlikely that measurements could be made to this accuracy.

This problem is solved in the MoWiTT by surrounding the two test rooms with a guard plenum through which controlled-temperature air is circulated as shown in Fig. 3. This has the effect of decoupling the envelope heat flow from the external temperature and greatly reducing 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:

$$\left[\frac{\delta W}{W}\right]_H = \frac{R}{R_E}\left(\frac{E}{F}\right)\frac{\Delta T_G}{\Delta T}\left(\frac{\delta H}{H}\right), \qquad (10)$$

where ΔT_G is the temperature difference between the guard air and the air in the test room. As can be seen, 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 can make this factor small. We have taken it to have a value of 0.1 in making error estimates.

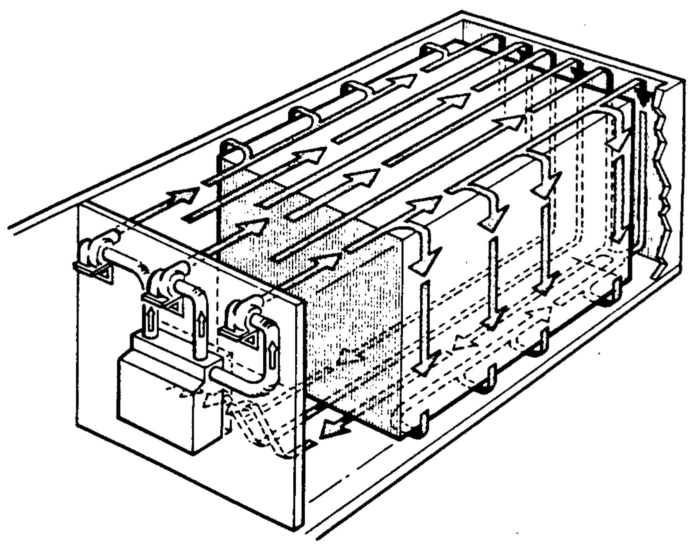


Figure 3. MoWiTT Guard System, showing circulation of forced-flow, temperature-controlled air around the two test rooms.

Table 3. Estimated Fractional Error Magnitudes in MoWiTT.

Source	(a) Expressions: Contribution to $\delta W/W$ — Nighttime	(a) Expressions: Contribution to $\delta W/W$ — Daytime	(b) Numerical Calculations: Contribution to $\delta W/W$ — Nighttime	(b) Numerical Calculations: Contribution to $\delta W/W$ — Daytime
Air Heat Content	$\frac{\sqrt{2}C}{\tau U_o \Delta T}(\frac{V}{F})R\delta T_A$	$\frac{\sqrt{2}C}{J_o}\tau(\frac{V}{F})\frac{1}{B}\alpha T_A$	$0.10\ R\ \delta T_A$	$0.014\ \frac{1}{B}\ \delta T_A$
Envelope Heat Flow	$(\frac{E}{F})(\frac{R}{R_E})(\frac{\Delta T_G}{\Delta T})(\frac{\delta H}{H})$	$\alpha(\frac{\delta H}{H})$	$0.11\ R\ \frac{\delta H}{H}$	$0.4\ \frac{\delta H}{H}$
Infiltration	$\frac{C}{U_o}\ (\frac{V}{F})\ (\frac{\Delta T_G}{\Delta T})R\ \delta a$	$C\frac{\Delta T_G}{J_o}(\frac{V}{F})(\frac{1}{B})\delta a$	$0.10\ R\ \delta a$	$0.015\ \frac{1}{B}\ \delta Ta$
Climate-Control System	$\xi\ [1+(\frac{E}{F})(\frac{R}{R_E})(\frac{\Delta T_G}{\Delta T})](\frac{\delta L_C}{L_C})$	$(1-\alpha)\ \frac{\delta L_C}{L_C}$	$(1 + 0.11\ R)\ (\frac{\delta L_C}{L_C})$	$0.6\ (\frac{\delta L_C}{L_C})$

The presence of the guard reduces the effect of errors from a number of sources by the same factor. Table 3 summarizes the contributions to $\delta W/W$ from each of the four sources of error. From this table it can be seen that, with the guard, achieving the nighttime design goal requires a 5% accuracy for the climate-control system and the envelope heat flow measurement, knowledge of the air infiltration rate to an accuracy of $\pm$ 0.05 air changes per hour, and knowledge of the interior mean temperature to $\pm$ 0.05°C. These are achievable requirements.

Measurement of the area-integrated envelope heat flow, H, is achieved by lining the interior surfaces of each test room with large-area heat-flow sensors, as shown in Fig. 1(c). These sensors were specifically developed for this application (10), (11) and provide about 90% coverage of the interior surfaces. Tests on prototypes reported in Ref. 11 indicated that the sensors would have adequate accuracy, and preliminary tests on the full-size production models are promising.(12)

All electrical inputs to each test room are monitored using specially constructed, accurate AC wattmeters that are insensitive to phase angle or waveform. This allows a measurement of the power delivered both to the electric heater and the circulating fan which has an accuracy better than 1%. Since the test room will not generally operate in the cooling mode for winter nighttime measurements, the 5% requirement will usually not apply; Table 3 indicates that daytime measurements require an accuracy of 10 - 20%. While this is not a difficult requirement when loads are large, it becomes more so for small loads. In order to achieve good accuracy in measuring the heat extracted by the cooling system, the MoWiTT extracts heat from each test room with a liquid-to-air heat exchanger. The flow rate, f, of the cooling fluid together with the fluid temperature where it enters (T_i) and leaves (T_e), the test room are measured, and the extracted heat is computed from:

$$L_C = \rho\ C_p f\ (T_e - T_i), \tag{11}$$

where ρ and C_p are the density and specific heat of the fluid, respectively. The percentage error arising from this measurement system is:

$$\frac{\delta L_C}{L_C} = \left(\left[\frac{\delta(\rho C_p)}{\rho C_p}\right]^2 + \left[\frac{\delta f}{f}\right]^2 + 2\left[\frac{\delta T}{T_e - T_i}\right]^2\right)^{1/2} . \tag{12}$$

One can see from this that accuracy from this system gets progressively worse as loads become small, since either f or $(T_e - T_i)$ becomes small while the measurement error does not. With the present MoWiTT measurement system, design accuracy can be maintained down to a cooling load of around 50W; for smaller loads, improvement in accuracy will be needed.

Through careful sealing of the test rooms, inadvertant air infiltration rates are reduced considerably below 0.05 air exchanges per hour, eliminating this source of uncertainty. Since there is a considerable pressure difference between the guard and each test room, sealing is quite important, and gasketing of the access doors and sample holding frame has been carefully engineered. For the same reason, the infiltration rate through the room envelope is independent of the outdoor pressure.

Through use of calibrated thermistors, individual temperature measurement accuracies better than 0.05°C are attainable. Measurement of an accurate mean interior temperature, T_A, then becomes a question of correct placement of thermistors and sampling of temperatures. Since the MoWiTT has the capacity to record many thermistors and to sample them frequently, this requirement presents no insuperable problems.

COMPUTER CALCULATION OF MoWiTT PERFORMANCE

In the foregoing discussion we have concentrated on nighttime measurements, with daytime estimates relying on the ad hoc parameter, α, the fraction of solar gain conducted through the envelope of the test room, which was taken without justification to have a value of 0.4. This procedure was used because a simple model such as the one used above is completely inadequate for calculating daytime performance of the test space.

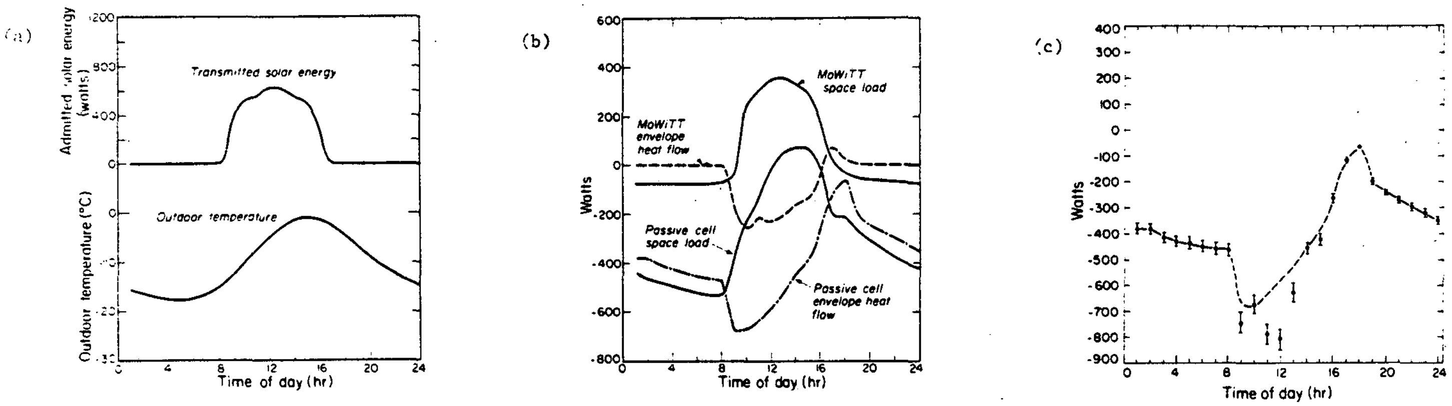
We next turn to a computer simulation of the MoWiTT performance. This is done for two reasons: First, we wish to check the conclusions about accuracy reached on the basis of the simple model. Second, we would like to know how well the MoWiTT, with its active guard and large-area heat-flow sensors, performs in comparison to a more modest and conservative system.

We have therefore simulated the performance of two measurement facilities: (a) one test room of the MoWiTT, and (b) a passive test cell of identical size and construction, but without the active air guard space and large-area heat-flow sensors. As Eq. (1) shows, it is not possible to construct the window net energy flow without a knowledge of H(t), the envelope heat flow. Accordingly, we add a network of commercial heat flux sensors to the hypothetical passive cell. These are arranged on a rectangular grid on each interior surface, with a vertical spacing of 1.2-m (4 ft) and a horizontal spacing of 0.6 m (2 ft). On the floor and ceiling the 0.6 m spacing is along the direction perpendicular to the fenestration. (This network requires some 55 commercial heat flow sensors).

The program BLAST was used to perform the simulation because it does an hourly net heat balance and calculates the heat fluxes into each interior surface. Both the MoWiTT and the passive cell were assumed to have a triple-glazed window mounted in the sample wall. A cold, clear design day (Dec. 20) at Donner Summit, in the Sierra-Nevada mountains of California, was assumed. The transmitted solar energy and outdoor temperature assumed in the calculation are shown in Fig. 4(a).

The purpose of this calculation was to simulate the measurement process in each facility, assuming that the loads and envelope heat fluxes calculated by BLAST are the true ones. Infiltration and changes in air heat content were neglected. It was assumed that L_c could be measured to 5% accuracy in both facilities, and both the large-area heat-flow sensors and the commercial heat-flow sensors were also assumed to have 5% accuracy.

For the passive cell, one additional step was needed in the calculation. BLAST treats each envelope surface as a one-dimensional problem, by averaging solar and radiative fluxes over the entire surface. While this is a reasonable approximation for the MoWiTT, where the area-integrated heat flow is measured directly, it does not treat correctly the discrete heat-flow sensor network of the passive cell. Accordingly, for each hour of daylight the location of the moving patch of directly transmitted solar gain was computed by hand and it was determined which heat-flow sensors were directly illuminated. Approximate values of the heat flux passing through those heat-flow sensors were computed from the transmitted solar intensity and the surface heat flux computed by BLAST. The values of the heat flux seen by the other sensors on the illuminated wall were corrected for the fact that part of the solar radiation was concentrated in the directly illuminated spot. The area-weighted sum of the heat fluxes was taken to be the contribution to H(t) from that surface. Corrections to the radiative heat balance, due to the fact that surface temperatures in the directly illuminated spot will be higher than the mean temperature used by BLAST, were neglected for both the MoWiTT and the passive cell.



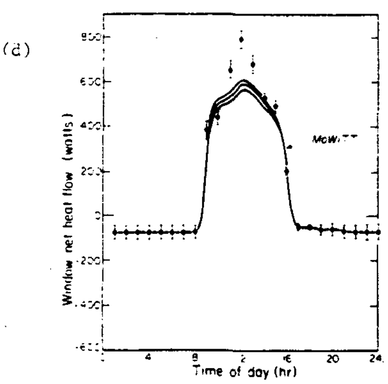


Figure 4. BLAST Simulation of a triple-glazed window measurement comparing the MoWiTT and a passive test cell. (a) Assumed outdoor temperature and solar energy transmitted through the window. Indoor temperature is assumed to be a constant 20°C. (b) Calculated space loads, $L_c(t)$, (solid curves) and envelope heat flows, H(t), (dashed curves) for the MoWiTT and for the passive cell. (c) Measurement of envelope heat flow in the passive cell. Dashed curve: BLAST calculation of the envelope heat flow; points with error bars: envelope heat flow, which would be measured with the heat-flow meter grid described in the text. (d) Derived values for the net heat flow, W, through the window. Solid curves are the mean, + 1 standard deviation, and -1 standard deviation, for measurements by the MoWiTT. Points with error bars are the corresponding quantities for the passive cell with heat-flow meter grid.

The results of the calculation are shown in Figs. 4 (b), (c) and (d). Fig. 4 (b) shows the BLAST calculation of $L_C(t)$ and $H(t)$ for the MoWiTT and the passive cell. In both cases, during the daytime $H(t)$ is approximately 40% of the total solar gain, which is the origin of the value of 0.4 taken for α 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% assumed accuracy to produce the time-dependent absolute errors, $\delta L_C(t)$ and $\delta H(t)$. For the passive cell, during the daylight hours the values of $H(t)$ were corrected for the effects of the moving patch of sunlight as described above. These are shown as points in Fig. 4 (c), with the derived errors $\delta H(t)$ shown as error bars on the points. As can be seen, the points show sizable deviations from the BLAST-calculated curve (assumed to be the true value) which are considerably larger than the range 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. 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. This is a graphic demonstration of the type of systematic error that may arise in daytime measurements attempted with an inadequate measurement system.

In Fig. 4(d), the values $L_C(t)$ and $H(t)$ are combined using Eq. (1) to produce the window net energy flow, $W(t)$. The errors $\delta L_C(t)$ and $\delta H(t)$ are added in quadrature to produce the measurement error $\delta W(t)$. For the MoWiTT these results are shown as a curve surrounded by an error band (which is too small to be visible during nighttime hours); for the passive cell they are represented as points with error bars.

This calculation reveals no surprises for the MoWiTT, which maintains approximately 5% accuracy throughout the day. This is because, for this sample and design day, one effect--solar gain during the day, transmissive loss at night--clearly dominates. For the case of a north-facing window one might see degraded accuracy during the daytime. For the passive cell, however, two effects may be observed which point up the advantage of the MoWiTT: First, during the night measurements the accuracy of the measurement is degraded to the approximate range $35\% \leq (\delta W/W) \leq 50\%$. This is because the nighttime measurement of $W(t)$ involves taking the difference between measurements of two large numbers, as can be seen from Fig. 4(b). Second, large systematic errors of up to 30% occur during the daytime measurement. Since these are much larger than the random error expected, measurements with this facility would result in erroneous conclusions about both the magnitude and the shape of the curve $W(t)$.

CONCLUSIONS

We conclude that direct measurement of the net energy flow through fenestrations of moderate complexity under realistic conditions is a difficult undertaking requiring a specialized measurement facility. One such facility, the MoWiTT, is designed to be capable of accurate measurements on fenestrations with thermal resistance up to 10 times that of single glazing and shading coefficient down to 0.1. This represents a significant advance in fenestration measurement. The first module of the MoWiTT, undergoing calibration at LBL, is shown in Fig. 5.

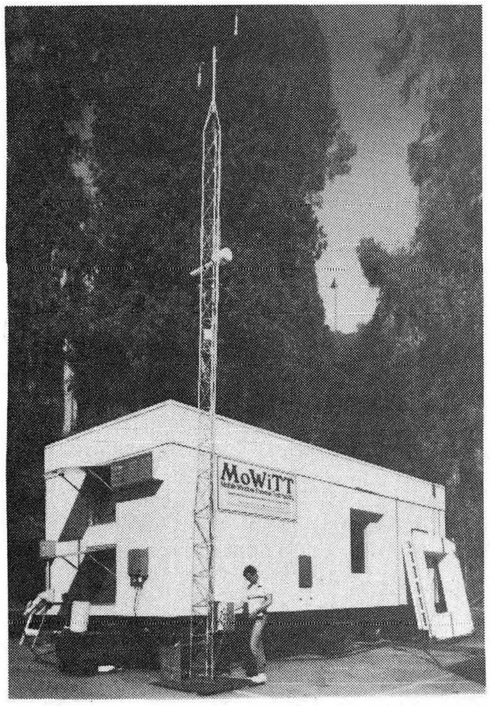


Figure 5. The first MoWiTT measurement module during calibration at Lawrence Berkeley Laboratory.

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