Thermal Performance of Vinyl Covered Fiberglass Solar Screens

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ABSTRACT

During the past decade, vinyl covered fiberglass solar screening has evolved from insect screening with a capability, as judged by shading coefficients, of eliminating up to 75 percent of the summer solar heat gain.

Calculated results of heat transfer coefficients (4 values) for a single pane window system in winter heating conditions have suggested some dramatic reductions in thermal transmittance with the addition of solar screens.

This paper addresses the thermal performance of a porous screening applied to fenestration products and its effect on the annual energy balance. The program utilized the latest technology for evaluating thermal performance under accepted laboratory controlled environments for summer and winter conditions, both daytime and nighttime.

INTRODUCTION

The operators of most commercial buildings throughout the nation are now finding it increasingly difficult to provide a reasonably comfortable environment for their occupants with today's energy limitations.

With recently set summertime and wintertime mandatory thermostat settings to cope with, building managers are now faced with unfortunate choices.

The purpose of this paper is not to delve into the causes for this increasingly costly set of problems, but to present one possible partial solution to this complex situation.

While clear double glazed and storm windows provide improved protection against winter thermal conditions, they do little to reduce solar heat gain in the summer when it is a serious problem requiring air-conditioning. In office buildings, as they exist in the present-day-world, the winter solar heat gain is almost invariably a disaster from the occupant comfort standpoint, whether they have double or single glazed clear windows.

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Lighting in residences are a minimal part of energy consumption. In high rise structures it is considerably greater. Solar screens permit daylight to enter in the proportion described by the shading coefficients including all of the direct sun as well as from ground reflected light. Solar screens can also be removed for the winter months from residences to permit the beneficial winter solar heat gain and the additional natural lighting when darkness comes earlier. Architects have rarely designed buildings in the past 30 years to minimize electrical lighting or maximize daylighting. There seems to be little tendency to do so in the future as can be presently judged.

Vinyl covered fiberglass solar screens (also termed sun screens and/or plastic solar screens) are a part of a generic type of shading devices attached to the exterior of windows. In the November 7, 1979, Federal Register the DOE Residential Conservation Service Program refers to metal or plastic solar screens as a window heat gain retardants measure.

Any shading device on the window exterior is more effective than those on the interior. The generic classification of solar (sun) screens also include minitured louvered and aluminum as those with a long term (over 10 years) performance capability. The literature references to the miniture louvered type suggest that it was the first solar screen developed with data which was presented in the 1972 and 1977 ASHRAE Handbook of Fundamentals. While the development of vinyl covered fiberglass for insect screens dates back to the very early 1950's, their evolution into solar (sun) screens dates to the middle 1960's. The earliest solar properties evaluated were the very important shading coefficients. None of these properties have become part of the technical literature, although they were used in commercial literature.

At that time, it has been generally agreed, that the ASHRAE Handbooks use of heating and cooling degree days data was to determine the size of heat and/or air conditioning equipment rather than to determine energy efficiency in a building. After 1973 it should have become apparent that equipment sizing and energy efficiency or conservation in a building was not directly involved with saving and/or reducing energy. Fortunately we believe that that is now the case in 1979.

Berman and Silverstein (1) provided some exciting and intriguing thermal conductivity (U_c) calculations for single paned and double glazed windows at various wind velocities in winter conditions from 15 mph down to 2 mph. They speculated that "2 mph is a hypothetical value for a wind velocity at the glazing surface which can conceivably be realized, on the average, by the use of retrofitted porous wind-reducing screens fitted to the perimeter of existing windows." They also stated that this "speculation is not optimistically unreasonable, and the results serve to indicate that some engineering development along these lines should be done." The calculated "U" Value was 0.67.

Since solar screens fit very nicely and precisely into the definition of a porous wind-reducing screen, the desire to seek the suggested engineering development to confirm these calculations should be obvious.

John Yellott (6) also provided interesting information regarding reductions in thermal transmission (U Value) for windows shaded externally by louvered screens. He concluded that approximately 15 percent reduction is not only possible but may in some cases be conservative.

STANDARDS

Standards are always important in judging quality and reproducibility. The Screen Manufacturers Association Specification (SMA) 4001-1978, Proposed American National Standards Institute (ANSI) Specification for Solar Screening for Windows; Vinyl Coated Fiberglass for Energy Conservation and SMA 5001-1979, Proposed ANSI Specification for Tension Mounted Solar Screens are on record for this purpose, since the solar screens being evaluated in this paper are those made from vinyl coated fiberglass screening.

ENERGY CONSERVATION WITH WINDOWS

The FEA/NBS publication, "Energy Conservation with Windows" states, "Shading windows during the summer can eliminate the need for air-conditioning during much of the cooling season and drastically reduce the load on the air-conditioner when it is required. Shading is most effective if it occurs outside the window." Solar screens are normally retrofitted to the exterior of the window where they can intercept a major part of the solar heat gain before it reaches the window glass or glazing. Thus, in the summer, a major part of the electrical energy required for air-conditioning for reasonable occupant comfort can be conserved.

It is worthy of note that many of the non-residential buildings in our country's inventory were so designed that winter solar heat gain is a disadvantage causing occupant discomfort with too much heat on one side of the building while occupants in other areas are cold. It is as if architects designed their buildings and then told the HVAC mechancial engineers to make it work at the lowest initial cost and without design stage consultation. One solution is a shading device, and solar screens are one type. While the information in this report covers vinyl covered fiberglass sun (solar) screens attached permanently to the window exterior it should be noted that the Federal Energy Administration Report FEA/D-76/467, September 1976, "Identifying Retrofit Projects for Buildings," (3) stated (page D-6) "sun screens may be fixed permanently in position or can be arranged to slide in channels or made removable so that advantage can be taken of the sun's beneficial heating effect in winter." Reality suggests that these screens be permanently attached (with release for window washing) for high rise applications. New construction would most likely be indicated if sun screens were to be channeled. For residential use (one or two stories) solar screens can be removed in the winter, whether framed or tension screens are used. Thus, the winter solar heat gain can be utilized where the physical situation permits.

The question now becomes: "Could the effect of lower thermal conductivity (U_C) on heat loss or gain come close to equaling the potentially beneficial winter solar heat gain if the solar screens were left in place the year around?" "Potentially" is used above to indicate that, while the winter solar heat gain might be utilizable in homes, it almost always causes discomfort to an occupant who must remain near the windows in high rise office buildings. However, the questions deserve an effort to seek answers.

TEST EQUIPMENT

ASHRAE Paper PH 79-6 Rennekamp (4) described a new modified calibrated hot box capable of testing 4 foot by 5 foot windows. Figure 1 shows a pictorial view of this testing equipment. Among the facilities referenced, the National Certified Testing Laboratories (NCTL) was selected and its President, Dale E. Hein, is a co-author of this paper. The testing equipment has become known as the AAMA (Architectural Aluminum Manufacturers Association) test. The fact that NCTL has an unadvertised solar simulator was added inducement for this selection.

The unique simulator was based upon equipment developed by NASA - Lewis Research Center (5) which used air mass 2 with approximately 240 BTU/Sq Ft - ARHR to reproduce solar effects to approximate the earth's atmosphere. The device consisted of eighteen (18) 300 watt tungsten halogen lamps with dichroic coated reflectors evenly spaced to provide the proper intensity over the test area. The selection of the lamps used was based on this earlier NASA development, see Figure 1 showing the apparent curve relationship between the simulator and air mass 2.

We recognized that there had been relatively little data collected on this solar simulator. It was important that we seek both winter and summer solar test data with and without solar screens as well as winter nighttime evaluation.

Although calibration procedures for this type of equipment are well documented as part of AAMA Standard 1502.6, this procedure is not well known outside AAMA. Therefore, this procedure to calibrate the solar simulator is described in Appendix A.

With the chamber and solar simulator calibrated, as described in Appendix A, it is possible to evaluate the thermal transmission, and solar optical properties of fenestration and for fenestration-screening combinations, with confidence in the accuracy of the final test results.

When attempting to evaluate solar screening for thermal performance, it is necessary to first look at its basic application. It is obvious that retrofitted solar screening over a prime window must be considered in the same manner as a prime window by itself. In S. J. Rennekamp's (4) Paper, the need for a modified guarded hot box technique for determining thermal performance of window and/or window-screening combination was established. For this reason, it was decided to evaluate the solar control properties of solar screening using a modified guarded hot box to yield accurate, consistent and reproducible test results.

TEST PROCEDURE

Once the facility is adjusted and calibrated as described in Appendix A, it is then possible to proceed with determination of the solar control properties of solar screening retrofitted to fenestration products. A two phase program, performed on both 1/4" glass and a 1/4" glass/solar screen combination under both winter and summer conditions, was instituted. Phase 1 involved the determination of thermal transmittance (U) and Phase 2 was to determine the solar properties of the glass or glass/screen combination.

The testing was performed on a 4' x 5' test unit, consisting of one piece of 1/4" glass mounted in a 2 by 4 frame. After the test unit was installed in the chamber and prior to starting the test run, 13 premium grade 30 gauge copper constantan, Type T, thermocouples were applied to both the interior and exterior surfaces of the glass. The thermocouples were spaced so as to provide a representative temperature of the glass.

Once the thermocouples were in place, the equipment was made operational using winter environmental conditions, i.e., exterior temperature 18 degrees Fahrenheit, interior temperature 68 degrees Fahrenheit, with 15 mph wind velocity. To eliminate the effects of infiltration, a static pressure was applied to the warm side of the test unit to balance out the dynamic pressure created by the 15 mph wind on the cold side, giving a zero pressure drop across the specimen. The test units were tested until steady-state stabilization occurred. Stabilization was considered reached when the thermocouple readings on three consecutive sets of temperature readings spaced 30 minutes apart were within one degree Fahrenheit. Once stabilization was reached, the test run was made as detailed in Appendix B.

DISCUSSION OF TEST RESULTS

The data obtained from the testing program, as summarized in Table No. 1, indicates the relative solar control properties between glass and glass-solar screen combinations. It is obvious that solar screens are very efficient in reducing the amount of heat admission during the summer months. However, those properties of the solar screen which are responsible for the improvement in summer heat gain, would, under winter conditions, tend to reduce the potentially beneficial solar heat gain. We should, therefore, consider the performance of a window glass/solar screen combination under year around conditions and all directional exposures.

It would be a monumental task to attempt to include in this paper the performance of window glass/solar screen combinations for all locations within the continental United States. Therefore, in order to illustrate the annual energy efficiency of these combinations, a location at 40 degrees North latitude, such as Philadelphia, Pennsylvania, was selected for the computation. A daily energy balance was calculated for a one square foot area using the test temperature conditions and incident radiation values for February 21 and August 21. National climate center (6) data was used.

Table 2 is a summarization of a calculation for the annual energy balance, using Equation 6, of the building indicated which is located in Philadelphia. To simplify calculations, these results are based upon the maintenance of 68 degrees Fahrenheit inside all year. These calculations indicate an unshaded window total BTU consumption of 28,638 for one square foot of glazing on each side of the building and 16,736 BTU consumption for solar screens under similar conditions. This represents about a 40 percent annual energy savings. When no air-conditioning is used between 68 Fahrenheit and 78 Fahrenheit, a 32 percent saving is indicated.

It should be noted that the calculations used to project the annual energy balance for a Philadelphia location are simplistic in nature and do not account for additional shading devices, number of occupants or HVAC operational variables or other locations.

As can be determined from Table No. 1, the addition of solar screens to the exterior of windows can provide, conservatively, a 15 percent reduction in the thermal transmission (U). This substantiates the findings of Yellott (2).

It was earlier noted that Berman (1) had presented an interesting and challenging opportunity. The results presented herein do not confirm the Berman calculations. It is possible that their assumption about a 2 mph laminar flow with a porous wind reducing screen is questionable. It is certainly not possible with a 15 mph perpendicular wind. The effects of a wind parallel to the window is not known and the NCTL test equipment did not have that capability.

It should be noted that window glass/solar screen combinations are highly effective in the summer. While the winter energy balances are not always positive from these test results (with a perpendicular wind and solar source), there are several conditions which can make winter situations with solar screens more positive. In the real world, the likelihood of a building being exposed continuously to a perpendicular wind and/or solar input is quite low. Thus, testing conditions reflecting both wind conditions and solar exposures at other than perpendicular will reflect improved performance as well as being closer to the actual environment.

ADDITIONAL CONSIDERATIONS

One consideration is a generalization based upon macro economics. The efficiency of conversion of the BTU's in a fossil fuel, coal for example, into electricity, is a maximum of 33 percent. The same amount of fossil fuel converted directly into space heat is estimated to be between 75 to 90 percent efficient.

The import of this is that a dollars worth of fossil fuel, such as coal, used for winter heating provides between 2 and 3 times the energy to the consumer as the same amount of fuel if it were used to generate electricity for air-conditioning. The net effect is a devaluation of the potentially beneficial winter solar heat gain. However, we are not suggesting that any potential energy conservation should not be strived for.

The results of this testing program plus the above noted leverages indicate that retrofitted solar screening over fenestration products definitely reduces energy consumption on an annual basis.

CONCLUSIONS

Based on all the work presented for this paper, it can be concluded:

1. Retrofitted vinyl covered fiberglass exterior mounted solar screening on fenestration products are definitely energy efficient as a shading device in the summer environment since a shading coefficient as low as 0.23 was determined by solar testing.

2. Test data indicates a net reduction in thermal transmittance of approximately 15 percent is possible with the addition of solar screens as determined under the worst possible testing conditions.

3. Wintertime benefits of retrofitted exterior mounted solar screens are complex and such determination would be based upon local climatic conditions, building orientation, internal heating and lighting loads, etc.

4. Net annual energy savings of approximately 40 percent are indicated when compared to an unshaded window based upon the worst possible testing conditions and results. When no air conditioning is used between 68 degrees Fahrenheit and 78 degrees Fahrenheit, a 32 percent savings is indicated.

5. Maximum cost savings can be affected with solar screening because the greatest reduction in energy consumption is in the summer months when energy costs per therm are at a maximum.

6. From an equal amount of fossil fuel twice as much heating can be produced as electricity due to their relative conversion efficiency. Thus for energy conservation it becomes twice as important to save electricity by reducing air conditioning as it does to save fossil fuel from heating.

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also

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Springfield, VA

(2) John I. Yellott
ASHRAE Paper No. 2227, New Orleans, January, 1972
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(3) "Identifying Retrofit Projects for Buildings"
Report FEA/D-76/467, September, 1976

(4) S. J. Rennekamp
ASHRAE Paper PH 79-6 (January, 1979-Philadelphia)
U-Value Testing of Windows Using a Modified Guarded Hot Box Technique

(5) K. Yass and H. B. Curtis
NASATM X-3059 Lewis Research Center (June, 1974)
"Low-Cost, Air Mass 2 Solar Simulator

(6) "Local Climatology Data - Annual Summary Philadelphia"
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Ashville, NC

TABLE 1

SOLAR CONTROL PROPERTIES

	WINTER				SUMMER				SOLAR OPTICAL PROPERTIES			
	NIGHT U_c	DAY q_a	S.H.G.	S.C.	NIGHT U_c	DAY q_a	S.H.G.	S.C.	T_s	R_s	A_s	GLASS A_s
1/4" GLASS	1.17	143.65	202.15	.89	1.07	233.31	204.44	.89	81	6	13	100
1/4" GLASS w/SOLAR SCREEN	1.00	-14.47	44.03	.23	0.92	73.1	44.12	.23	18	2	80	1

1. Winter U_c = Thermal Transmittance due to conduction determined by Test using a modified guaded hot box (ASHRAE Paper 79-6 by Rennekamp) under conditions T_o = 18F, T_i = 68F, 15 mph wind. Perpendicular to test unit.
2. Summer U_c = Thermal Transmittance due to conduction determined by adjusting test results, using ASHRAE procedures, to conditions of T_o = 95F, T_i = 68F, 7 1/2 mph.
3. q_a = Total Heat Admission through glass. (BTU/HR-FT^2)
4. S.H.G. = Solar Heat Gain (BTU/HR-FT^2)
5. S.C. = Shading Coefficient
6. T_s = Solar Transmittance in percent
7. R_s = Solar Reflectance in percent
8. A_s = Solar Absorptance in percent
9. Glass A_s = Percent of Solar Absorptance in Glass
10. Solar Optical Properties tested in accordance with ASHRAE 74-73.

TABLE 2

TYPICAL ANNUAL ENERGY CONSUMPTION IN BTU'S (1)

		1/4" GLASS					1/4" GLASS WITH SOLAR SCREEN				
MONTH	MEAN TEMP(2)	NORTH	SOUTH	EAST	WEST	YEAR TOTAL	NORTH	SOUTH	EAST	WEST	YEAR TOTAL
JAN.	32.3	899	366*	565	565		834	559	761	761	
FEB.	33.9	817	412*	386	386		787	520	685	685	
MAR.	41.9	543	448*	62*	62*		585	373	454	454	
APR.	52.9	109	452*	567*	567*		267	155	130	130	
MAY	63.2	240*	292*	890*	890*		29	3*	111*	111*	
JUNE	72.3	533*	640*	978*	978*		186*	210*	318*	318*	
JULY	76.8	596*	740*	1223*	1223*		273*	322*	409*	409*	
AUG.	74.8	454*	978*	1099*	1099*		211*	323*	350*	350*	
SEPT.	68.1	154*	1138*	769*	769*		46*	247*	168*	168*	
OCT.	57.4	124	1061*	316*	316*		202	54*	104	104	
NOV.	46.2	509	727*	182	182		500	231	429	429	
DEC.	35.2	833	381*	559	559		767	503	708	708	
TOTAL		5811	7635	7596	7596	28,638	4450	3032	4627	4627	16,736

* Denotes Air Conditioning Load

1. Base on 1 FT^2 of area on each side of a square building with a true North Exposure, located in Philadelphia, Pennsylvania.
2. "Local Climatology Data - Annual Summary Philadelphia," National Climate Center, Ashville, North Carolina
3. Consumption based on maintaining 68 F interior temperatures the year around. Savings are indicated in the text for maintaining 68 F all year as well as for air-conditioning only over 78 F.

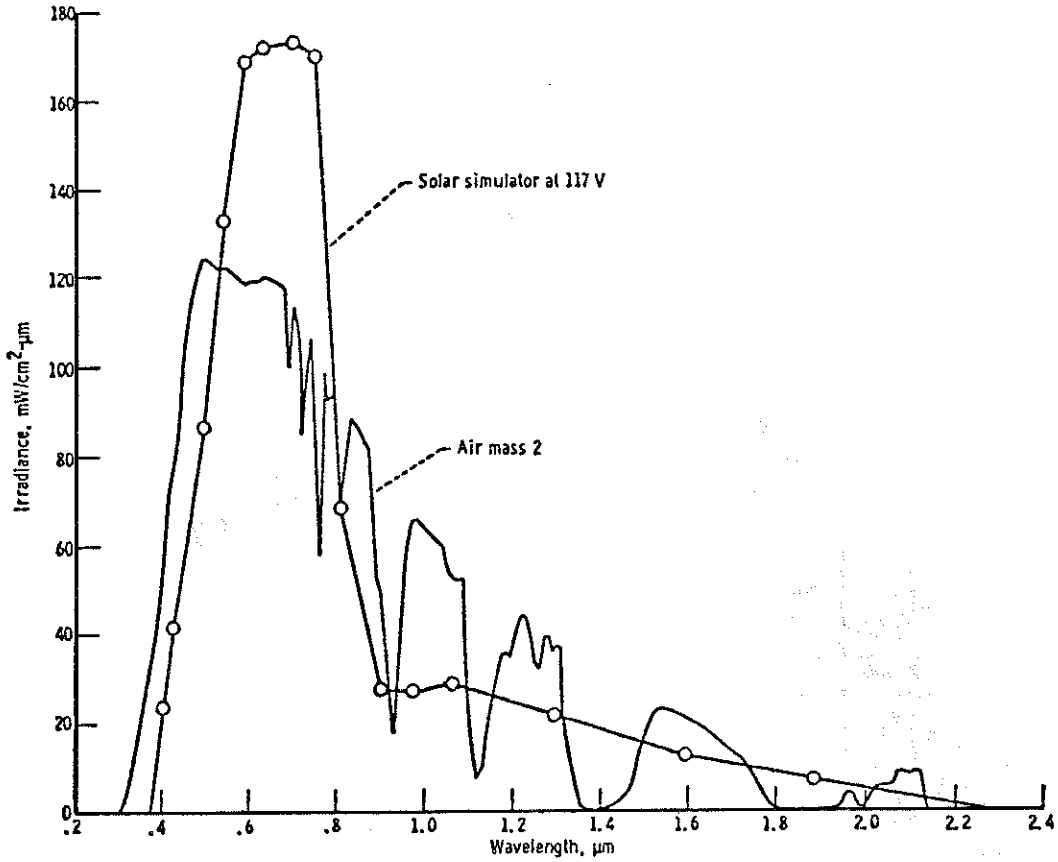
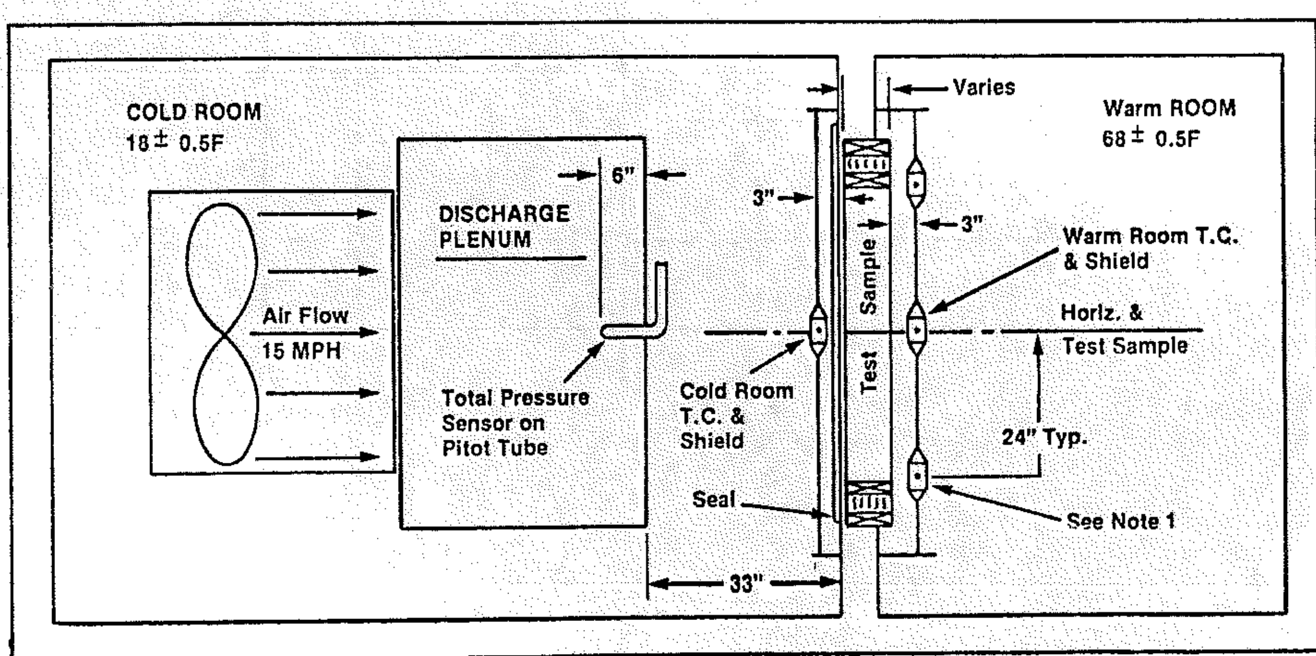


Fig. 1

THERMAL CHAMBER



Overall Chamber Length May Vary

Fig. 2

NOTE

1. *Thermocouples and shields on warm side movable to maintain 3" spacing to test sample*

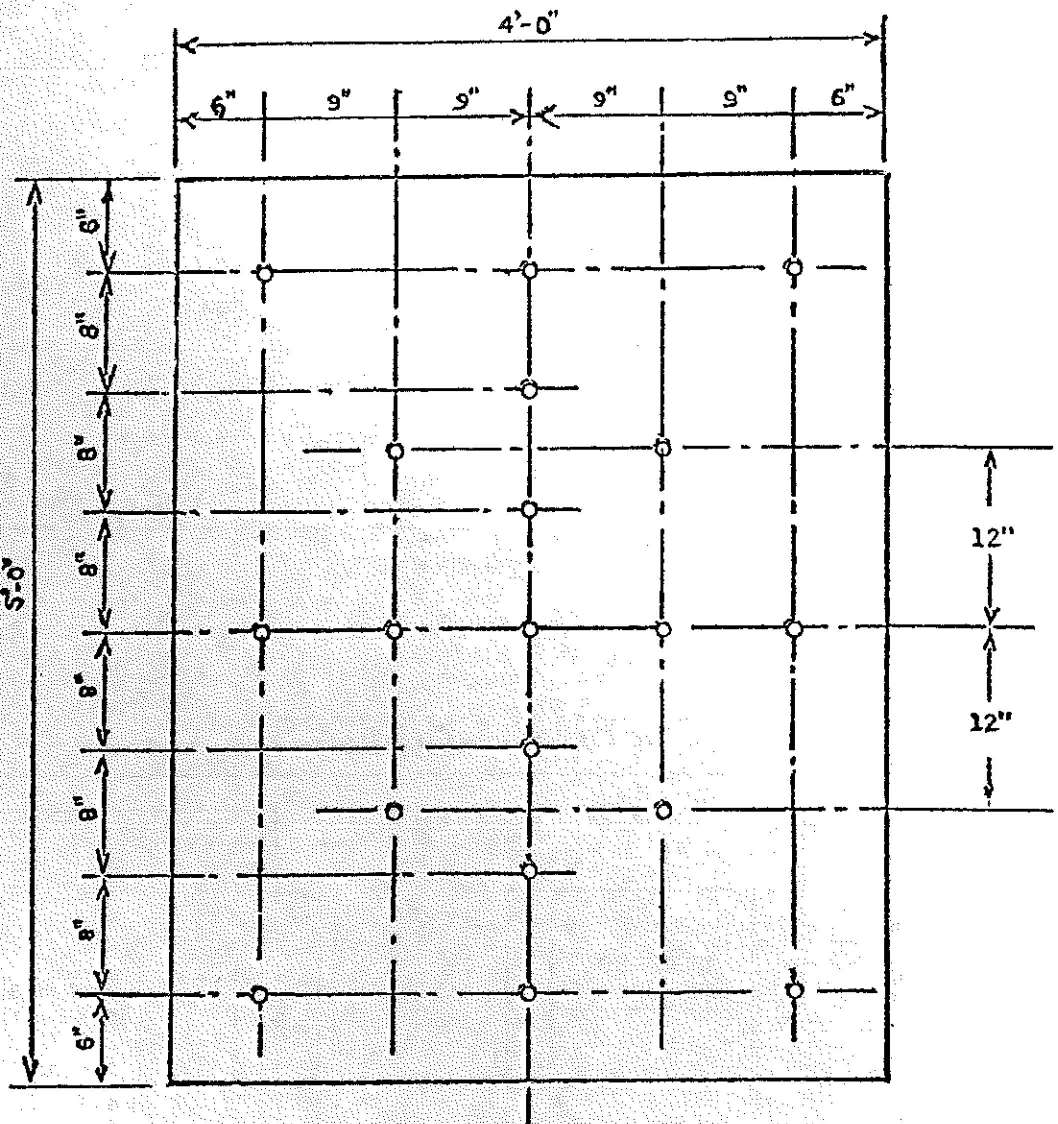


Fig. 3 Velocity profile measurement traverse location in plenum discharge

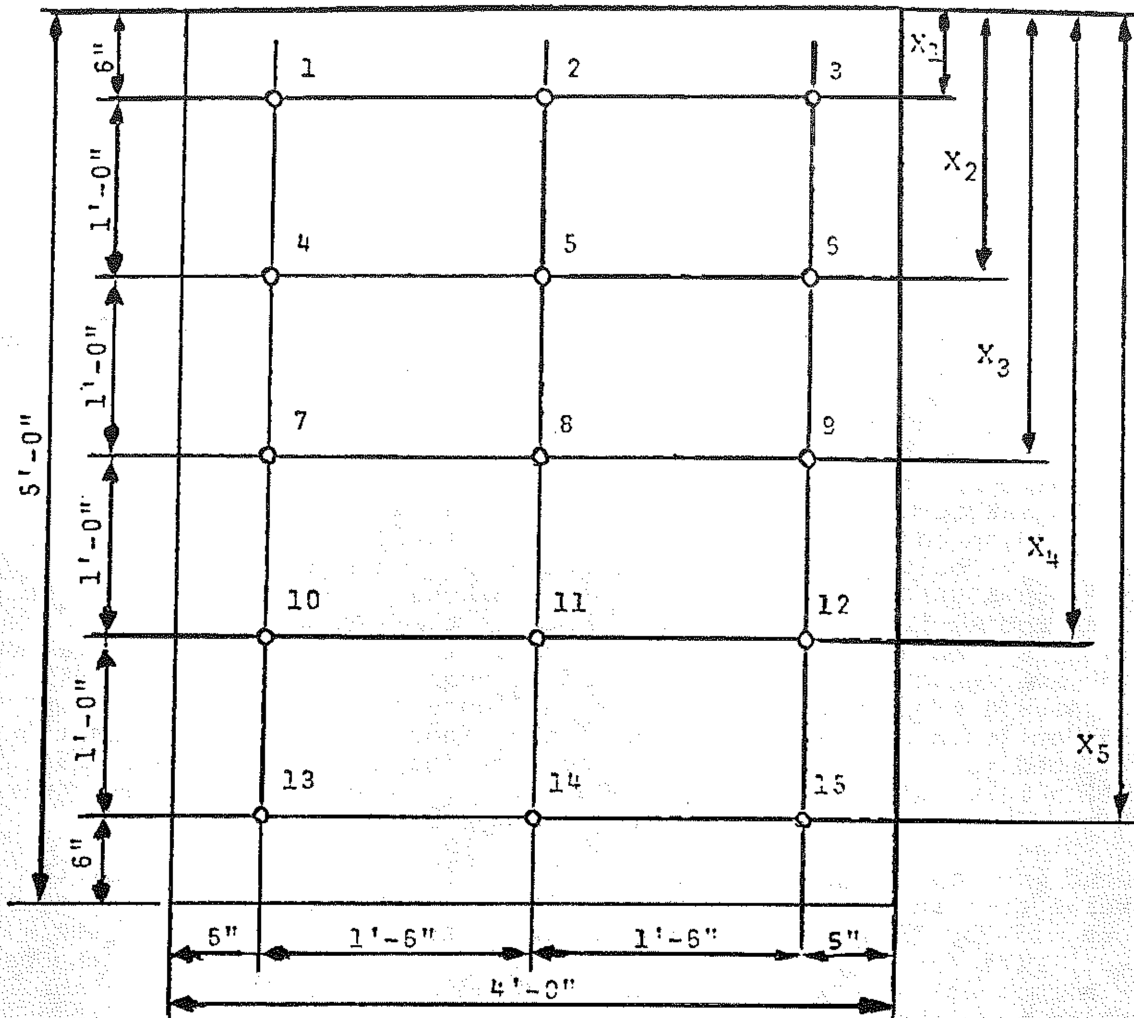


Fig. 4 Thermocouple location on calibration panel

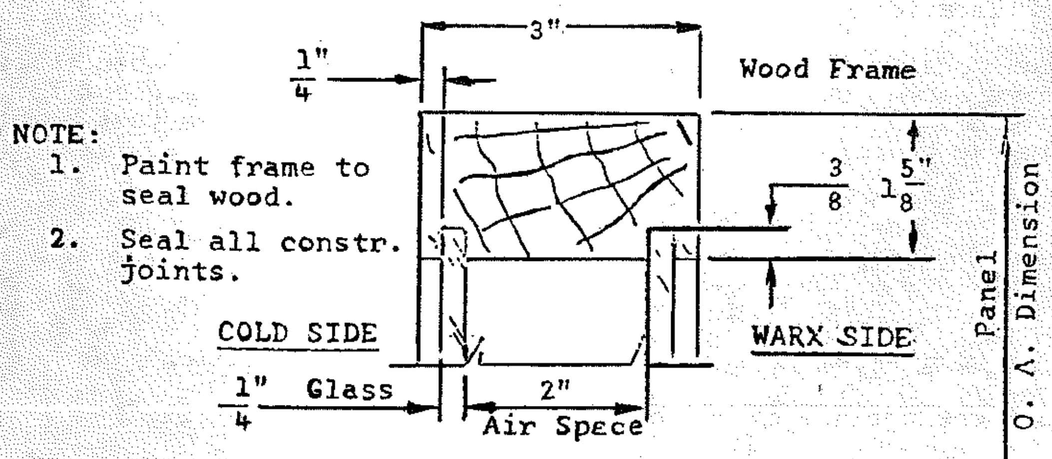


Fig. 5 Calibration panel frame

CALIBRATION PROCEDURE FOR THERMAL TEST EQUIPMENT

APPENDIX A

Like a Standard ASTM C236 test facility, the chamber has a warm side and a cold side with the window being bested located between and separating the two rooms. Figure No. 2 is a schematic of the test facility, which has a cross sectional area of approximately 10 feet high by 10 feet wide. The major difference between this facility and ASTM C236 facility is the addition of a large centrifugal fan and plenum arrangement capable of providing a uniform 15 mph wind directed perpendicular to the outside of the fenestration product. On the warm side, the facility is a series of fans and air flow directional equipment to simulate a free convection condition on the interior surface of the test specimen. Temperatures of 18$\pm$ 0.5 degrees Fahrenheit and 68$\pm$ 0.5 degrees Fahrenheit were selected as standardized conditions on the cold room and warm room sides of the test facility respectively.

The reliability of this test procedure is assured by determination of Standard ASHRAE surface coefficients under an actual test condition prior to testing of any product, the facility being calibrated to verify that the exterior air velocity, and the inside and outside surface coefficients agree with Standard ASHRAE conditions. With the discharge end of the air flow plenum located 33 inches from the separation between the warm and cold sides of the chamber, ajdustments were made to the trim baffles in the discharge plenum until the average air velocity, measured 6 inches inside the plenum discharge at the cross-sectional locations shown in Figure (2), were all 15$\pm$ 1 mph. The standard deviation of the population of the nineteen velocity observations was not permitted to exceed 1.1 mph. Corrections for air density were made at calibration temperature conditions. The center point velocity was required to be within 0.5 mph of the average air velocity.

The second important phase of the calibration is the verification of the inside and outside coefficients. To calibrate the facility, a standard all glass, four foot wide by five foot high calibration panel, as shown in Figure (4) was selected. Fifteen thermocouples were attached to the cold side of the panel and 15 thermocouples to the warm side as shown in Figure (3). With the calibration panel in place, the chamber was operated at the established 15 mph wind, with inside and outside temperature conditions as required for calibration. Adjustments are made and the chamber operated until conditions stabilized. Stabilization was defined as the point when all the thirty individual point temperatures obtained in three consective run spaces thirty minutes apart were within 1 degree Fahrenheit at each individual point location. When stabilization was reached, calibration runs were made.

A calibration run consisted of five sets of temperature readings spaced thirty minutes apart over a two hour period. The cold side surface coefficient for the calibration panel, h_{oc}, was calculated for each of the fifteen test points from data during the calibration run using the formula:

$$h = 0.975 \frac{(t_{ic} - t_{oc})}{(t_{ic} - t_{cc})} \quad \text{(Equation 1)}$$

Where:

h_{oc} = cold side surface coefficient for the calibration panel

0.975 = conductance of the calibration panel in BTU/hr.ft.°F

t_{ic} = warm side surface temperature of the calibration panel (°F)

t_{oc} = cold side surface temperature of the calibration panel (°F)

t_{cc} = cold side air temperature (°F)

Surface coefficients were determined from this formula at each of the fifteen locations for each of the five calibration runs. The outside surface coefficients for the five calibration runs were then averaged for the fifteen calibration locations.

The mean value of the fifteen average outside surface coefficients was required to be 5.8± 0.4 and the standard deviation of the population was not permitted to exceed 1.4. The 5.8 value is extremely close to the standard ASHRAE surface coefficient for a 15 mph wind condition.

The warm side surface coefficient for the calibration panel, h_{ic}, was calculated for each of the 15 test points from data taken during each of the five calibration runs using the formula:

$$h_{ic} - 0.975 \frac{(t_{ic} - t_{oc})}{(t_{wc} - t_{ic})} \quad \text{(Equation 2)}$$

Where:

h_{ic} = warm side coefficient for the calibration panel

t_{wc} = warm side air temperature (°F)

The warm side surface coefficient determined at each of the fifteen locations was then averaged for the five calibration runs. The average coefficients were found to be satisfactory when the d_{rms} was less than 0.15. The d_{rms} is given by the equation:

$$d_{rms} = \frac{\underset{i}{1,5} \ (h_{ave_i} - hs_i)^2}{5} \quad \text{(Equation 3)}$$

Where:

d_{rms} = root-mean-square deviation of warm side heat transfer coefficients.

h_{ave_i} = average calculated warm side heat transfer coefficients for elevation X.

X_i = location of thermocouple row from Figure (3).

hs_i = base warm side heat transfer coefficients for elevation X_i, as indicated in the following chart:

i	X_i (ft.)	hs_i (BTU/hr./sq./ft./°F)
1	0.5	2.10
2	1.5	1.50
3	2.5	1.45
4	3.5	1.40
5	4.5	1.30

The reason for the difference in the allowable inside surface coefficient values from the top to the bottom of the calibration panel is a consquence of the conditions which naturally occur on the inside surface of a window subjected to normal free convection conditions where the heat transfer at the top of the window occurs at a much faster rate than at the bottom of the unit, because the air is warmer at the top of the unit and drops as it falls along the surface of the window from the top to the bottom. This effect was compensated for by adjusting the allowable inside surface coefficient from the top to the bottom of the calibration panel. The center line values on the calibration panel were required to conform to the standard ASHRAE still air surface coefficients.

In order to completely evaluate the solar control properties of solar screening retrofitted to a prime window, the solar optical properties of the combination fenestration product must be determined. To this purpose, a solar simulator, based on earlier NASA (4) work, capable of providing spectral distribution of air mass 2 at approximately 240 BTU/hr. ft. has been constructed as an integral part of the modified guarded hot box facility. The solar simulator applies the radiation perpendicular to the test product and parallel to the wind flow.

Just as calibration is extremely critical in the modified guarded hot box, so is calibration of the solar simulator critical. The first phase of calibration requires the determination of spectural distribution of the various radiation wave lengths. Inasmuch as the basic concept and design of the simulator was based on previous NASA work, the spectural distribution for this simulator was assumed to be the same as previously determined by NASA.

The second important phase of this calibration is the verification of the radiation intensity at a plane located two inches from the warm side of the glass. Intensity readings were taken, using a Dodge Model 776 solar meter, over an area 24 inches wide by 36 inches high on a grid pattern 6 inches by 6 inches. The average of these 35 meter readings was used as the incident solar radiation in all calculations.

TEST PROCEDURE

APPENDIX B

By electronically regulating the heat input into the warm room, the temperature was accurately controlled within ± 0.1 degree Fahrenheit. The total heat input to the room was electronically metered, totalized and integrated into BTU consumed. All extraneous conduction losses from the warm room, including conduction through the test chamber to ambient, conductive loss through the separation wall of the cold room, and conductive loss through the wood frame were determined. Losses to ambient were determined using two thermalpile arrangements, one on the inside surface of the test chamber wall, and one on the exterior surface of the test chamber wall. By maintaining ambient conditions that provide identical temperatures on both the inside and the exterior surface of the test chamber, a net zero heat loss occurs. Using the temperature data recorded during the test run along with calibration data for the separation wall, the losses from the warm room to the cold room are calculated. Using standard ASHRAE procedures, the loss through the wood frame is also calculated. The total of all extraneous losses are then subtracted from the total BTU consumed to determine the actual heat loss through the test product.

While the thermal transmittance determined by actual test applies to winter conditions, we do not live in a world of one season. Therefore, it was necessary to determine thermal transmittance under summer conditions. ASHRAE identifies summer conditions as having 7½ mph wind velocities instead of 15 mph wind velocity. Because wind velocity has a direct effect on thermal transmission, it is necessary to adjust the tested U_c values to the value anticipated under summer conditions. The thermal resistance of the product as tested would be:

$$R_t = \frac{1}{U_c} \quad \text{(Equation 4)}$$

Where:

R_t = Total Thermal Resistance of Product

U_c = Tested Thermal Transmittance of Product (BTU/hr. ft.^{2} F)

By definition, the total thermal resistance of either a single homogeneous material, such as glass, or two materials such as glass and solar screening, separated by an air space, is the sum of the individual resistances. These resistances include not only the conductivities of the homogeneous materials, but also the inside and outside film coefficients. Therefore, R_t can be written:

$$R_t = R_1 + R_2 + R_3 \quad \text{(Equation 5)}$$

Where:

R_1 = Resistance of warm side film coefficient

R_2 = Sum of all resistance except film coefficients

R_3 = Resistance of cold side film coefficients

From the ASHRAE Handbook of Fundamentals (1977), the thermal resistance of the film coefficient for 7½ mph wind velocity is 0.25 and 15 mph wind is 0.17. By substitution, R_3 then becomes 0.25 and the new thermal transmittance for summer conditions can be determined by reversing the calculation process. Solar optical properties are an important part of the overall solar control performance of a fenestration product. These properties were determined in accordance with ASHRAE Method 74-73, under both summer and winter conditions. With these properties, determined by tests, it is possible to calculate the total heat admission through the test specimen using the following formula:

$$Q_a = T_s + N_i (A_s) + U_c (T_o - T_i) \quad \text{(Equation 6)}$$

Where:

Q_a = Total heat admission through the specimen (BTU/hr. ft.)

T_s = Transmitted radiation (BTU/hr. ft.)

A_s = Absorbed radiation (BTU/hr. ft.)

N_i = $\frac{U}{h}$

h_o = Cold side surface coefficient

T_o = Cold side air temperature

T_i = Warm side temperature

Table (1) is a summarization of the results from these tests for both winter and summer conditions. With this data, it is possible to consider the positive aspects of retrofitted solar screening when applied to fenestration products.