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Combined Thermal and Air Leakage Performance of Double Windows

The heat flow pattern through a prime and storm window with temperature difference and air pressure difference causing infiltration or exfiltration is complex; so the total heat loss through a combined storm and prime window has usually been determined by calculating the transmission loss due to temperature difference and adding to that loss the heat required to heat the infiltrating air from outdoor temperature T_o to the indoor temperature T_i. Fig. 1 shows the model representing this case. The ASHRAE Guide has implied that such a model may be used since it publishes the data for infiltration rates in a different section from that which the heat transmission data is published, and uses the infiltration data obtained at no temperature difference in combination with heat transfer data obtained with no infiltration or exfiltration of air. Published works from various sources are available on air leakage rates through numerous window types, at no temperature difference, and overall heat transfer coefficients, and surface heat transfer coefficients and surface temperatures of vertical air spaces under natural convection,[1,2,3,4] with no air leakage.

Fig. 2 represents a model which is closer to the heat loss pattern in an actual window. Outdoor air at temperature T_o infiltrates into the space between the windows where it is heated to temperature T_a by the transmission loss through the prime window created by the temperature difference ($T_i - T_a$). The infiltrating air then enters the space at T_a rather than at the temperature T_o in the case of considering the transmission losses and infiltration losses separately. Also a considerable portion of the heat loss is transmitted by radiation between the panes. Calculations show that less heat losses occur with the more realistic model since there is a greater resistance to heat flow in this case, with the window being the equivalent of a counterflow heat exchanger. The glass and its surface coefficients act as resistance to the heat flow heating the infiltrating air. Furthermore, if exfiltration occurs on a window the air temperature in the space between the windows is higher than in the case of infiltration or no infiltration and the overall U value is lower than that obtained by tests of temperature differences only.

This paper reports the results of a laboratory investigation on combined storm and prime double hung windows with simultaneous air pressure differences and temperature differences across the windows. Measurements were taken of air leakage quantities, surface temperatures, air space temperatures and heat input requirements with various types of double hung windows. The windows were installed in a guarded hot box calorimeter with a system for creating a pressure difference across the window. Tests were run to determine the overall U value, air leakage rates and the temperature index, that is, the surface or air space temperature difference compared to the overall temperature difference.

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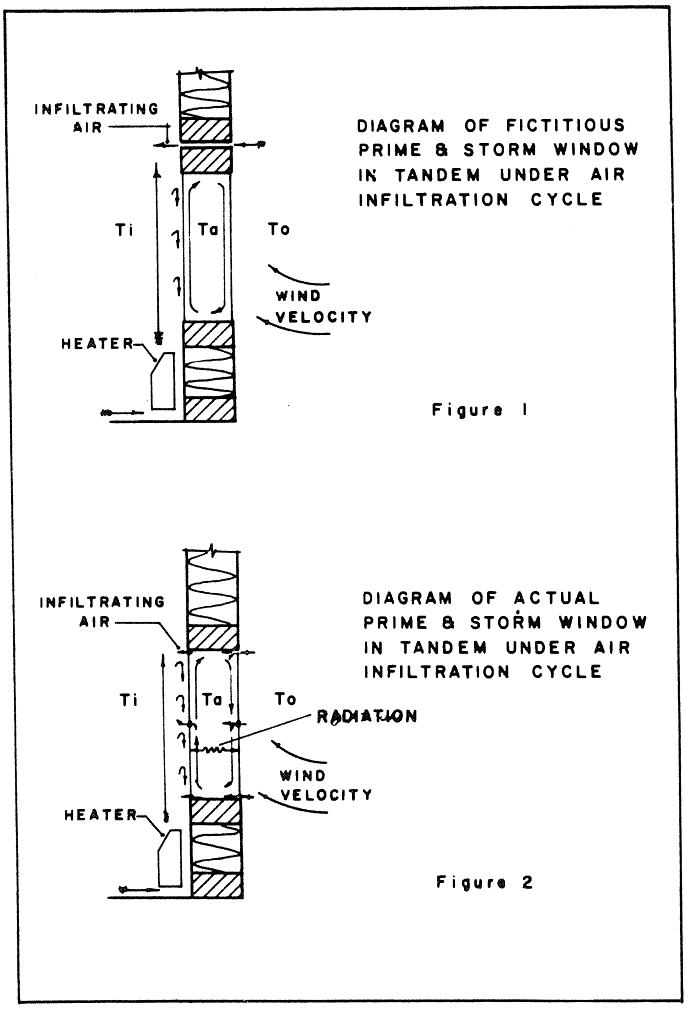


Figs. 1 and 2 Window models.

HEAT TRANSMISSION THROUGH DOUBLE WINDOWS

The overall heat loss of a combined storm and prime window using the model of Fig. 1 is given by

$$H_t = U\,A(T_i - T_o) + I \times \rho \times .24(T_i - T_o) \qquad (1)$$

Where $\rho = 0.075\ \text{lb}_m/\text{ft}^3$.

If the air between panes is assumed completely mixed at temperature T_a and radiation between panes is included in the heat transfer coefficients U_o and U_i;

where U_i = the overall heat transfer coefficient for the prime window

and U_o = the overall heat transfer coefficient for the storm window.

Then:

$$U_i\,A(T_i - T_a) = U_o\,A(T_a - T_o) + I \times \rho \times .24(T_a - T_o) \quad (2)$$

If U_i, U_o, and I are known and the assumption is made that the air temperature T_a is then mixed air temperature resulting from the air infiltrating or exfiltrating into the space between the windows the space temperature T_a can be calculated and the overall heat loss determined. Fig. 3 shows the differences between the results of Eq 1 and the measured heat input. This investigation attempted to determine if the mixed air temperature T_a could be accurately determined in order to find the values U_i and U_o that could be applied over a range of air infiltrating rates. The investigation revealed that the infiltrating air does not mix in the air space and considerable stratification takes place because the heat transfer takes place mainly by natural convection, and separation of radiation, air leakage and other effects to determine U_o is very difficult. It becomes apparent then, that because of the complex heat flow pattern through a window it is very difficult to determine separate components of heat loss for air leakage and transmission, and precludes the satisfactory mathematical description of the model in Fig. 2. Therefore another equation is proposed.

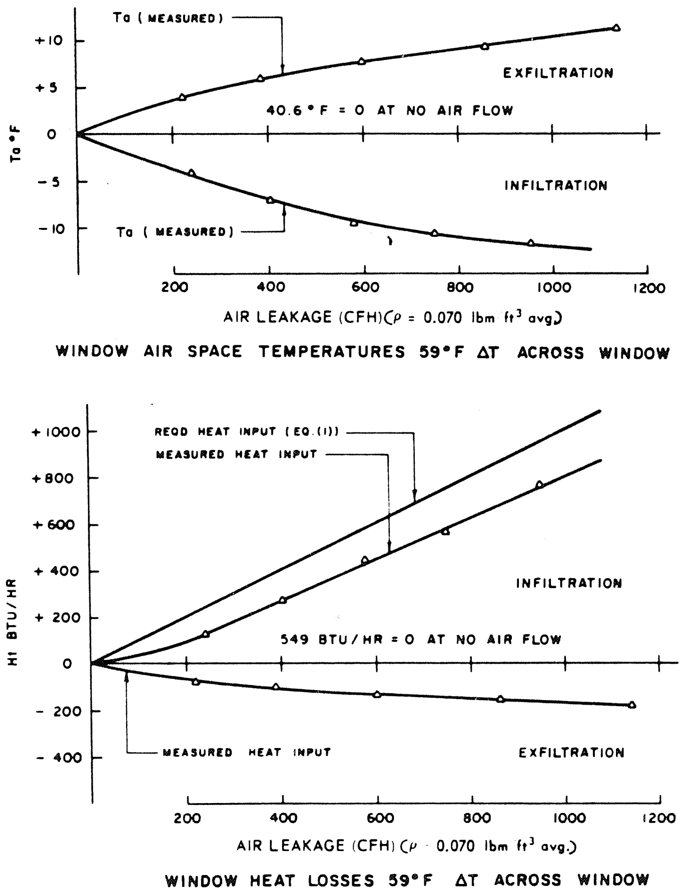


Fig. 3 Experimental heat losses through windows.

$$H_t = U_t\, A(T_i - T_o) \tag{3}$$

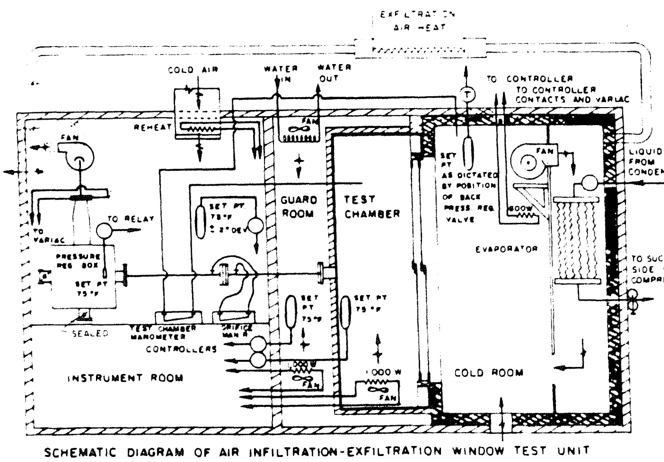
where H_t = the measured heat flow in Btu's/hr
U_t = apparent coefficient of heat transfer, Btu/ft^2 F.

U_t is an apparent heat transfer coefficient combining heat transmission and infiltration or exfiltration effects. The equation has the disadvantage that the U_t is dependent upon air infiltrating or exfiltrating rate.

TEST APPARATUS

Fig. 4 shows the test unit. Outside weather conditions were simulated in the cold room, and normal room temperatures 75 F were maintained in the test chamber, which was surrounded by a guarded hot box of the same temperature. Positive or negative pressures were maintained in the test chamber, depending on the air leakage cycle being run, and the air flow into or out of the test chamber was measured by the use of a suitable orifice.

The test chamber size of 7 ft 0 in. high, 5 ft 3 in. wide by 2 ft 10 in. depth was indicated by the size of windows which were expected to be tested. The chamber was constructed from 12 gauge galvanized sheet metal and welded throughout to eliminate the possibility of leaks developing after the work was in progress. The chamber was pressure tested under 7 in. of water before installation, and after installation the box was put under a positive and a negative pressure of 1 in. of water and all joints, inserts, and the outlet flange were tested with a soap and water solution, both inside and outside. Furthermore, the guard room was filled with a



SCHEMATIC DIAGRAM OF AIR INFILTRATION-EXFILTRATION WINDOW TEST UNIT

Fig. 4 Test Equipment.

dense smoke and the test chamber put under a negative pressure of 1 in. of water as an additional check to be sure of no air leakage other than through the window opening.

The test chamber was insulated as follows: the part of the chamber adjacent to the guard room was covered with 1 in. of urethane (conductivity = .15 Btu/hr ft^2 F/in.), the front of the chamber facing the cold room was insulated with 6 in. of urethane. The area immediately back of the window frame inside the test chamber was filled with three 1½ in. batts of mineral wool (density 4.0 lbs/cu ft and conductivity = .27 Btu/hr ft^2 F/in.) and this was covered with 1 in. of urethane.

It will be noted from observation of this figure that a small wall heat loss occurs through the insulated part of the test chamber, and a small heat gain through the window frame into the window air space. By the use of thermocouples the wall heat loss and frame heat loss was estimated to be of the order of 1.5% to 4% during air infiltration. For air exfiltration the wall heat loss was estimated to be of the order of 5% to 6% and the frame heat loss was estimated to be of the order of 2% to 4%.

Circulating fans were used in the guard room and a test chamber, limiting the differential reading between the lowest and highest thermocouples to a maximum of 1 F. The approximate transverse air velocity in the test chamber was 65 fpm, and the inside surface heat transfer coefficient was 2.0 Btu/hr ft^2 F, slightly higher than for purely natural convection. Air velocity at the test window on the cold side was approximately 7 mph in an upward direction and a convective heat transfer coefficient of 4.4 was determined.

The heating equipment installed in the test chamber consisted of one 1000 watt electric baseboard heater 3 ft 6 in. long by 3 in. wide by 8½ in. high installed beneath the window. The heater inputs to all chambers were varied to suit conditions, using an autotransformer and were controlled with thermistor sensing probe capable of maintaining a set point to ±0.1 F.

Heating equipment for reheat control was installed in the cold room in front of the circulating fans to the evaporator. The set points for the test chamber were held constant at 75 F while the set point of the cold chamber was varied from 75 F to -20 F. Heating was also provided in the pressure regulating box for use in heating the air on the exfiltration cycle to a constant temperature of 75 F. The heating load is switched on or off as required by the action of a mercury thermoregulator and relay with a differential of ± 0.02 F. To prevent frost formation on the window during exfiltration, a pipe returned the air from the cold chamber to a pre-heater where the air was heated to approximately 74 F before the returning to the fan and pressure regulating box.

For infiltration the fan connections were reversed, and the air discharged back into the cold chamber.

Air pressure in the test chamber was regulated by means of a variable speed fan connected to the box by a pipe line containing a standard ASME orifice. The piping and orifice arrangement consisted of 4 in. seamless steel schedule 40 piping with faced companion flanges and suitable thin plate orifices for flow measurement, all constructed in accordance with Chapter 4, Flow Measurement, of the American Society of Mechanical Engineers Code (ASME) on Instrument and Apparatus.[5]

Thermocouple locations on prime and storm windows are shown in Fig. 5. Air space temperature in the test chamber and cold room were measured in 3 locations opposite each other, using 24 gauge solid copper-constantan shielded thermocouples. The outside air thermocouples were located 12 in. from the inside edge of the rough opening.

The window air space thermocouples were located at the centerline of the air space in the position shown on the figure. The error of positioning was approximately ±1/32 in. The thermocouples were shielded and in each case extended some 6 in. from the vertically supporting rod so that flow patterns in the air space were not influenced by the vertical rod itself. The accuracy of all readings should be well within the limits of the recording instruments which is ± ¼% of the range −50 F to +250 F plus ¼ F and amounts to approximately ±1 F. Two thermocouples were constantly maintained in an ice bath at 32 F to indicate any instrument drift.

The glass surface thermocouples were located at the centerline of the window in the positions shown on the figure. The thermocouples were flattened to approximately .015 in. thick to 1/8 in. wide by ½ in. long and securely taped to the glass.

Thermocouples were also placed on the metal surface at the inside of the rough opening and on the wood frame of the outside of the rough opening for estimating heat loss through wall and frame.

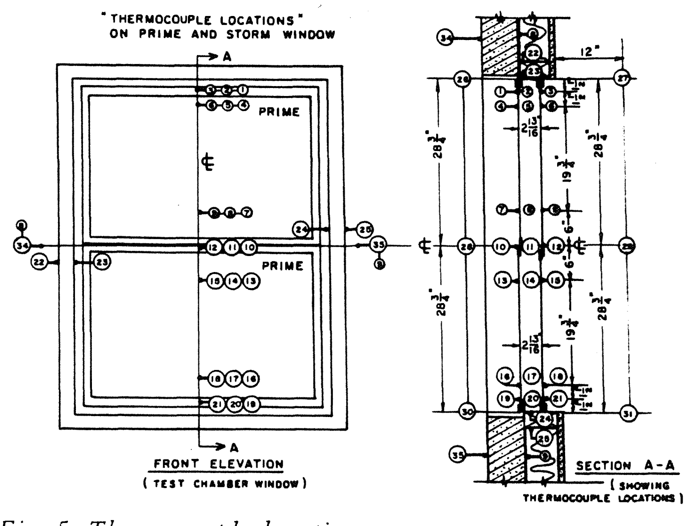


Fig. 5 Thermocouple locations.

The static pressure in the test chamber was measured by the use of an inclined manometer connected in an averaging manner to 4 suitably located ¼ in. diameter pressure taps in the test chamber. The differential pressure across the orifice was also measured by the use of a similar manometer. Both manometers can be read to .00124 in. of water per division.

TEST PROCEDURE

The window was tested on an air infiltration and air exfiltration cycle at cold room temperatures of 75 F, 37 F, 16 F and −20 F. The guard room and test chamber were in each case maintained at approximately 75 F by averaging 2 thermocouple readings in the former and 3 thermocouple readings in the latter.

Static pressures were maintained in the test chamber corresponding to wind velocities on the face of the window of 30 mph, 25 mph, 20 mph, 15 mph, and 10 mph by use of the formulae

$$h = .00482\ V\ \text{in.}^2 \qquad (4)$$

where h = static pressure in H_2O
V in. = wind velocity in mph.

Conditions were allowed to stabilize at each new static pressure reading for a minimum of 8 hrs, after which time electrical inputs were recorded for a minimum of 8 hrs. Thermocouple readings were recorded for 1 hr, and differential pressure across the orifice were read and recorded every 10 minutes for 1 hr and then averaged and the flows calculated.

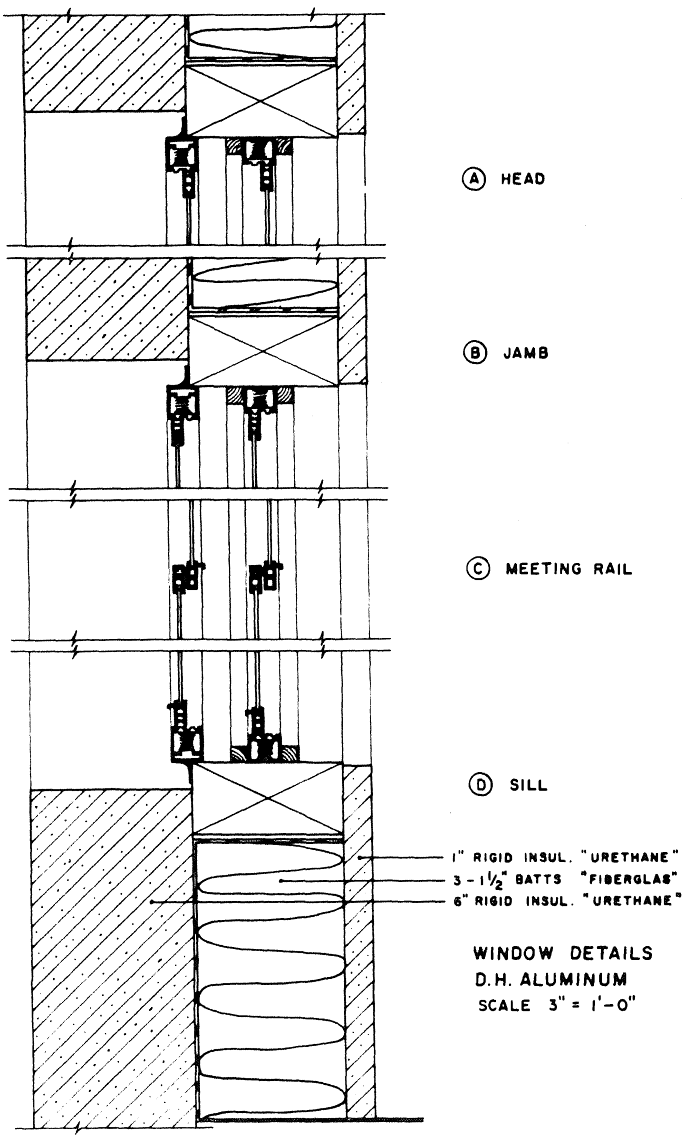


Fig. 6 Window details.

Air space temperatures and electrical inputs were also recorded for the no exfiltration or infiltration case for each of the cold room temperatures previously mentioned. For this test a blank was inserted in place of the orifice.

WINDOWS TESTED

Three double hung prime windows were tested in combination with various storm windows. Fig. 6 shows the detail of one of the sets tested consisting of aluminum prime and storm windows. Two of the sets had wooden prime windows, but all storms were constructed of aluminum. All except group 0 windows were tested for infiltration and exfiltration at no temperature differential by a manufacturer before being installed. Infiltration tests and exfiltration tests were repeated at no temperature differential, and a U value determined at no pressure difference. Table 1 gives test results for air leakage at no temperature difference.

The U values determined by the tests at no pressure differential are shown in Table I. These values are close to the values published in the ASHRAE GUIDE AND DATA BOOK. Infiltration rates for the window tests were greater than the values established by the manufacturer, with the differences quite large in some cases. The windows were shipped from the United States after air leakage testing, and installed according to the manufacturers instructions. Extreme care was taken to determine if the values we determined were valid and one apparent discrepancy arose with the locking device. If the window was locked in a certain position the infiltration was less than the value in another position. The variation of infiltration values establishes the fact that there would be considerable variation in the values for windows of the same manufacturer and same design because of installation techniques, variation in manufacture and methods of closure.

WINDOW AIR LEAKAGE RESULTS

It has been shown[6] that air flow rates through short capillary tubes plotted against the pressure drop ΔP through tubes of various length to ratios (L/d ratios) reduced to an equation of the type

$$Q = \text{constant}\ (\Delta P)^N \tag{5}$$

where Q = infiltration cfh
ΔP = pressure difference in. H_2O

for a limited range of Reynold's numbers. The exponent N increases from a value of 0.5 for an L/d approaching 0, to a value of unity for very large L/d's. The exponent N had values in the range of 0.6 to 0.75 for all double windows tested for both infiltration and exfiltration. This is the same range of values found by other investigators for infiltration.[4] From an examination of the infiltration air leakage results in Fig. 7 it will be noted that the air leakage increases with increasing temperature differential. Since the calculations of variation of properties of air did not account for the increase, it is surmised that this air leakage quantity increase is caused by an increase in the window cracks due to both the large temperature differentials and pressure differentials. However crack

TABLE 1

GROUP	AIR LEAKAGE CFH/FT P= .3" 25 MPH, T=0, ρ .075				U. OF S.[1] "U" (TESTED) P=0	REMARKS [2]
	INFILTRATION		EXFILTRATION			
	U. OF S.	MFG.	U. OF S.	MFG.	BTU/HR FT² °F	
051 CFH	610				.52	Tight fit aluminum prime and storm
CFH/FT	30.5					
052 CFH			760		.52	
CFH/FT			38.0			
151 CFH	630	504			.52	Tight fit wood prime and tight storm
CFH/FT	31.5	26.2				
152 CFH			680	595	.52	
CFH/FT			34.0	29.75		
131 CFH	980	604			.54	Tight fit aluminum prime and loose storm
CFH/FT	49.5	30.2				
132 CFH			800	684	.54	
CFH/FT			40.0	34.2		
251 CFH	810	570			.52	Average fit aluminum prime and tight storm
CFH/FT	40.5	28.5				
252 CFH			870	740	.52	
CFH/FT			43.5	37.0		
231 CFH	Did not test	960			.52	Average fit aluminum prime and loose storm
CFH/FT		48.0				
232 CFH			Did not test	830	.52	
CFH/FT				41.5		
351 CFH	790	637			.54	Loose fit wood prime and tight storm
CFH/FT	39.5	31.85				
352 CFH			930	575	.54	
CFH/FT			46.5	28.75		
331 CFH	1800	1601			.54	Loose fit wood prime with loose storm
CFH/FT	90	80.05				
332 CFH			1780	1034	.54	
CFH/FT			89	51.7		

[1] Adjusted to inside and outside heat transfer coefficients of 1.50 and 6.0 btu/hr °F ft² respectively.
[2] All Storms are Aluminum.

changes which occurred due to pressure differences should remain fairly constant regardless of the temperature differential across the window, while change due to temperature should increase with increasing temperature differential. Therefore, the constant in Eq (1) is influenced mainly by the temperature changes.

The same line of reasoning may be applied to the windows on air exfiltration, but as the window sash contraction would be much smaller in this case, the difference in air leakage quantities is not as significant. (Fig. 8)

The increase of infiltration with temperature difference is linear as is shown in Fig. 9, therefore a relationship can be set up as

$$I = I_o + S.I_o.\Delta t \qquad (6)$$

where I = infiltration rate cfh/ft at specified Δt
I_o = infiltration rate at 0 F Δt
S = slope

The constant S determined for this series of tests is S = .005. Comparison of the computed results from the above equation and the tests show a maximum discrepancy of 15%. This error is acceptable in view of the number of unknown factors influencing the infiltration in windows. Exfiltration rates

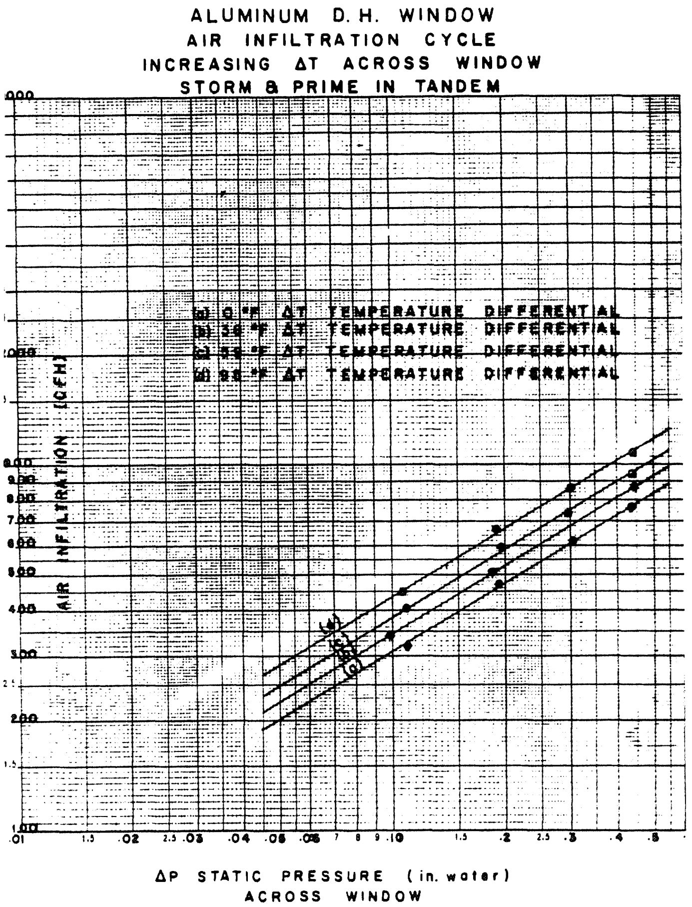


Fig. 7 ΔP static pressure (in. water) across window.

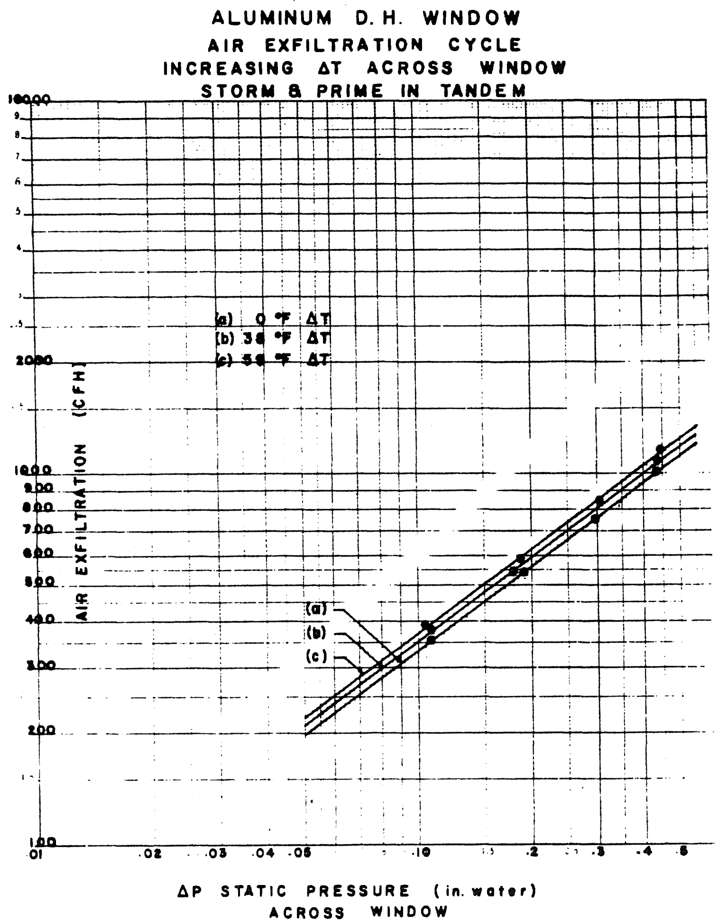


Fig. 8 ΔP Static pressure (in. water) across window.

changed only slightly with temperature difference, and no relationship was determined.

TEMPERATURE PROFILES

The results of the vertical centerline temperature measurements for the prime surface, air space, and storm surface are shown on Fig. 10 for air infiltration, and Fig. 11 for air exfiltration. These profiles are for the 59 FΔT and are similar to data at other temperature differentials. They are plotted non-dimensionally as the ratio of the difference in temperature between the surface, or air space, and the cold room, to the overall air to air temperature difference. Height is given as the ratio of the distance from the bottom of the window to the total height of the window.

The work in Ref 1 shows that for natural convection, the vertical air space temperature profiles, when bounded by isothermal surfaces, varies with the air space thickness. The profile depends on the heat flow "regime" existing in the air space. There are 3 distinct regimes of heat in the air space, which depend on the value of the Grashof Number,

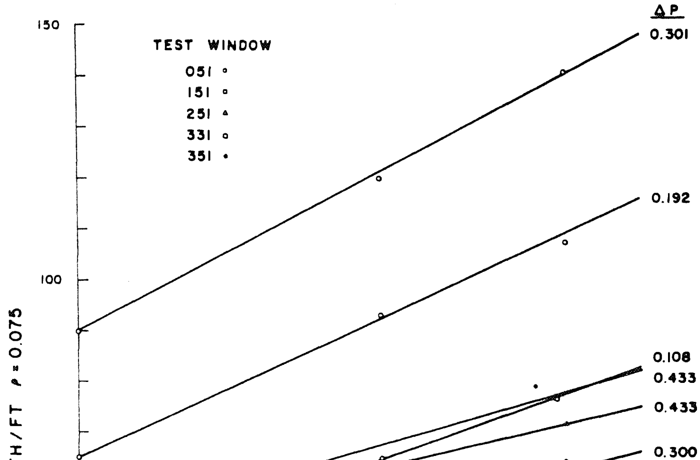


Fig. 9 ΔT F.

(Gry) and the height to thickness ratio $\frac{Y}{L}$. The "conduction regime" occurs when Gry is small and $\frac{Y}{L}$ is large and heat is transferred in the air space mainly by conduction. The temperature and velocity profiles are fully developed, and the temperature drop from the hot to the cold surface is linear over the air space. Slight convective effects will appear at the lower edge of the hot surface and the upper edge of the cold surface.

When Gry is large and $\frac{Y}{L}$ is small, the heat is transferred in the air space mainly by convection and the temperature and velocity gradients are concentrated in the boundary layers along the vertical surfaces. The temperature of the core is essentially constant along horizontal lines. Conditions under which such a temperature and velocity field exist are referred to as the "boundary layer regime."

For intermediate values of Gry and $\frac{Y}{L}$, the heat is transferred in the air space by both conduction and convection, and is referred to as the "transition regime."

At no air leakage for a $\frac{Y}{L} = 20.5$, (air space dimensions of 2 13/16 in. wide by 59½ in. high), these profiles exhibit characteristics similar to that shown in Fig. 11A for a $\frac{Y}{L} = 14.6$ (2 sheets of glass with air space dimensions of 4 in. wide by 60 in. high), where the natural convective flows inside the air space were determined to be of a boundary layer

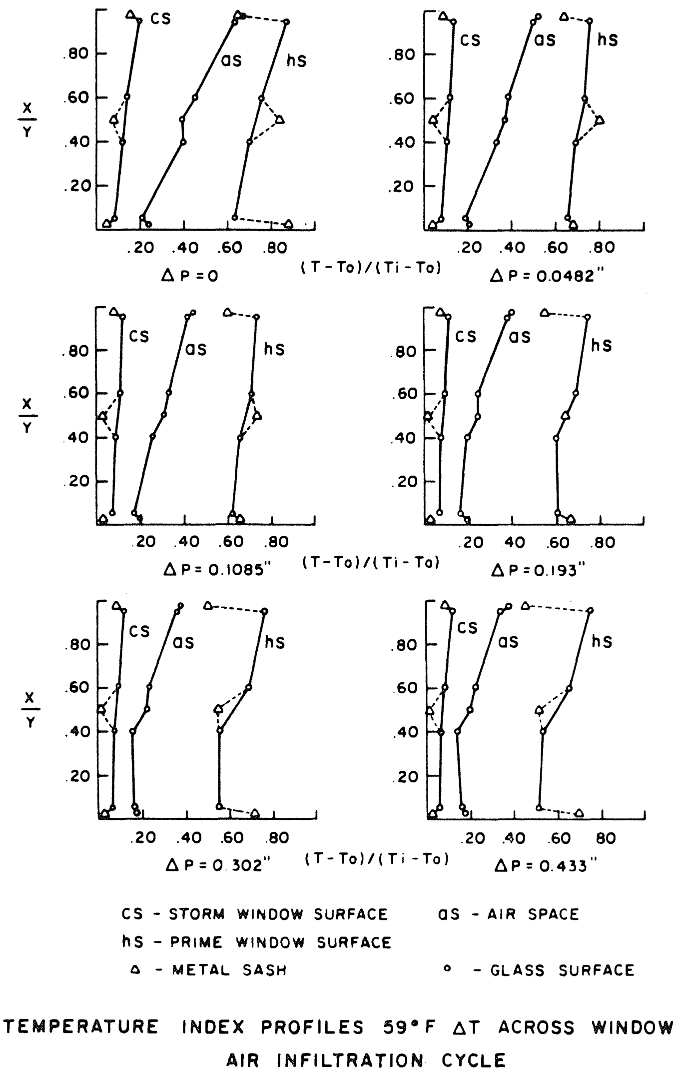


Fig. 10 Window Surface Temperature Profiles with Infiltration.

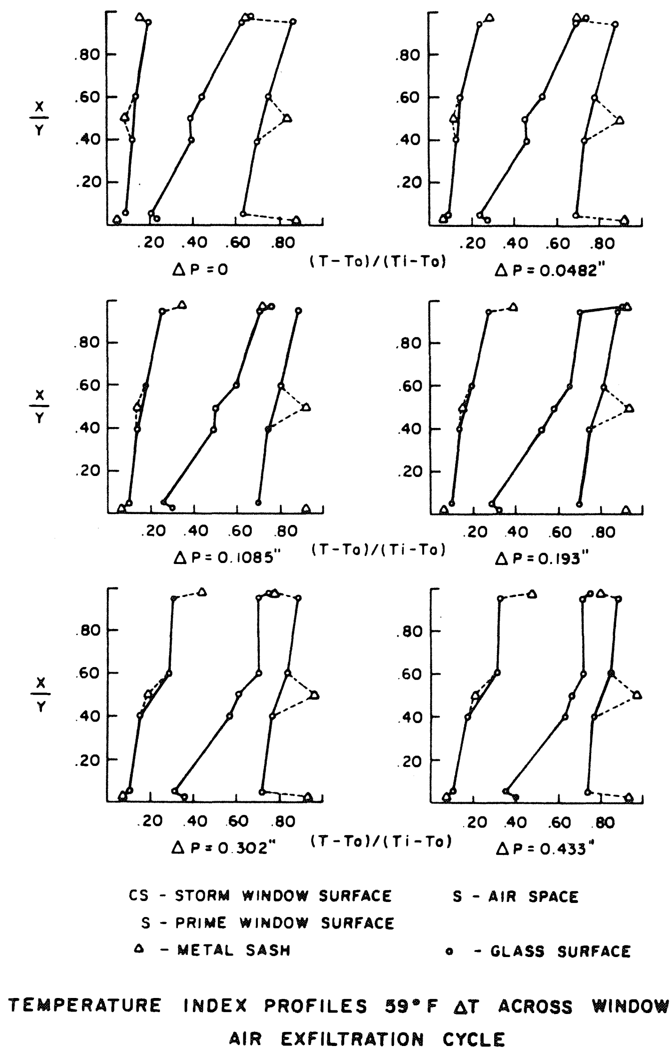


Fig. 11 Window Surface Temperature Profiles with Exfiltration.

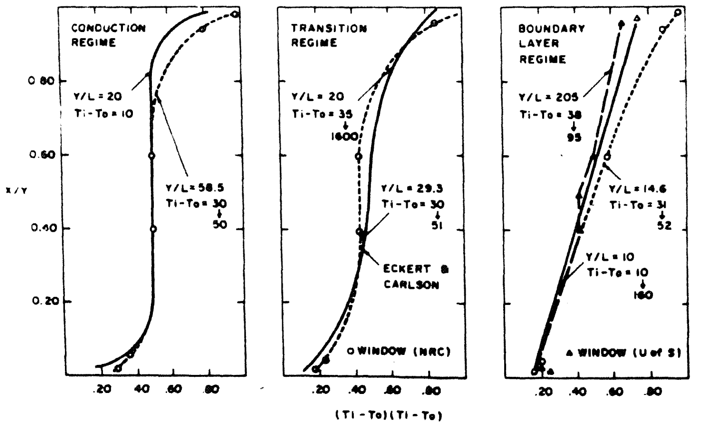


Fig. 11A Temperature Profiles at Vertical Centerline of Air Space with No Air Leakage.

nature. The slight temperature inconsistancies at midheight of the window are due to the irregularities of the double hung window sash. In the boundary layer regime the vertical temperature profile is practically linear. According to results given by Eckert and Carlson,[1] generally, air space of ¼ in. or less will fall within the "conduction regime," air space between ½ in. and 1 in. will fall in the transition regime and air spaces above 2 in. are in the boundary layer regime.

It will be noted that the largest change in shape (Fig. 10) for the warm surface profile occurs on the lower pane of the prime window, for increasing air infiltration. While on the cold surface profile the maximum change in shape occurs on the upper pane of the storm window. This is to be expected when air motion between the surfaces is due primarily to natural convection.

When the air space is subjected to increasing air exfiltration (Fig. 11a) the profiles are again similar to the basic "no air leakage" profile. However, the shape of the air space profile changes from a nearly linear profile to a curve with increasing air exfiltration. The shape of the hot surface profile remains very nearly constant with increasing air exfiltration.

The "temperature index" profiles for the hot surface, cold surface and air space, is essentially independent of overall temperature difference for the range of outside temperatures tested. However, they are very dependent on the rate of air infiltration or exfiltration, due, principally, to the interference with free convective heat transfer and the introduction of air at varying temperatures in the air space.

Double windows permit higher inside relative humidities, due to the higher minimum prime surface temperatures. As might be expected, the temperature index decreases, as well as the allowable space relative humidity, with increasing air infiltration. The reverse is true for increasing air exfiltration. The lowest temperature index occurred on the metal sash during maximum infiltration, and points to the obvious need of incorporating thermal insulating barriers in the metal frames of windows, if the highest possible relative humidities are to be maintained in the space.

HEAT TRANSMISSION THROUGH DOUBLE WINDOWS

The mechanism of heat transfer through a prime and storm window with temperature difference and pressure difference causing air leakage is quite complex and involves the conduction, convection and radiant modes of heat transfer. Therefore a U_t value (apparent overall heat transfer coefficient Btu/hr/ft^2 F) was determined for the window tested.

The term U_t is defined by Eq 3 and was determined experimentally for each type of window,

$$H_t = U_t\, A(T_i - T_o)\,, \tag{3}$$

where H_t = the net measured heat flow in Btu/hr. Shown in Fig. 12 are the U_t values for one window with infiltration or exfiltration.

The U_t values, based on an infiltration rate/sq ft of window, are shown in Fig. 13. For comparison with the case where the infiltration air is considered separately from the heat transmission, a U_t value based on indoor to outdoor temperature difference is derived as

$$U_t = U_o + I_f\, \rho\, c_p \tag{7}$$

where U_o = the overall heat transfer coefficient at no infiltration;

I_f = infiltration cfh/sq ft of window area at A density of 0.075 lbs/cu ft;

and c_p = specific heat of air, 0.24 Btu/lb/F

It may be seen from the plot that these values are 10 to 20% greater than the measured U_t values. An effect which is explained by the counter flow heat exchanger effect achieved with the combined effect of infiltration and heat transmission. The results of the tests in the laboratory show a linear relationship for the U_t values in the range of infiltration tested. The maximum deviation between an average fitted curve and any test point was 9%. In view of the unknowns in the infiltration rates of installed windows this graph is a practical method of finding the overall heat loss through a window. The procedure would be to find the infiltration rate/sq ft of a window determined from the present data (0 F temperature difference) and corrected by Eq 6 to the appropriate temperature difference. The infiltration rate thus determined is used to find the U_t value which is applied in Eq 3 to find the total heat loss of the window H_t.

Example: Double hung window, 17 sq ft, 50 FΔt, 17.5 ft of crack, infiltration 40 cfh/ft of crack at 0 FΔT. Infiltration at 50 FΔt = 40 + .005 $\times$ 50 $\times$ 40 = 50 cfh/ft, cfh/sq ft = $\frac{17.5 \times 50}{17}$ = 51.5 cfh/sq ft.

From Fig. 13 U_t = 1.14 Btu/hr ft^2 F. Total heat loss, infiltration plus transmission

$$H_t = 1.14 \times 17(50)$$
$$= 969 \text{ Btu/hr.}$$

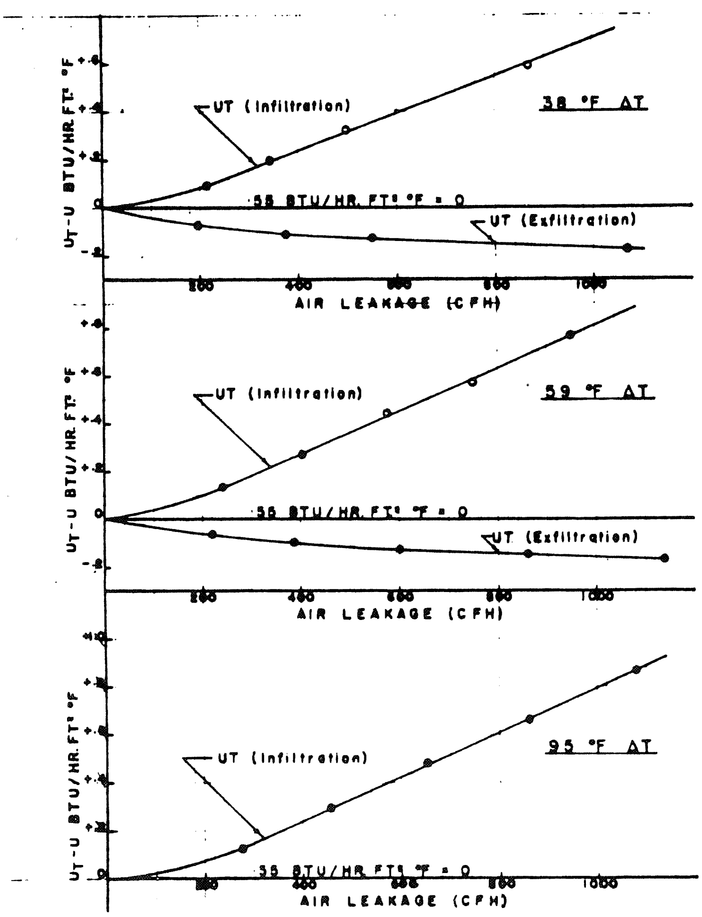


Fig. 12 Apparent Overall Heat Transfer Coefficients.

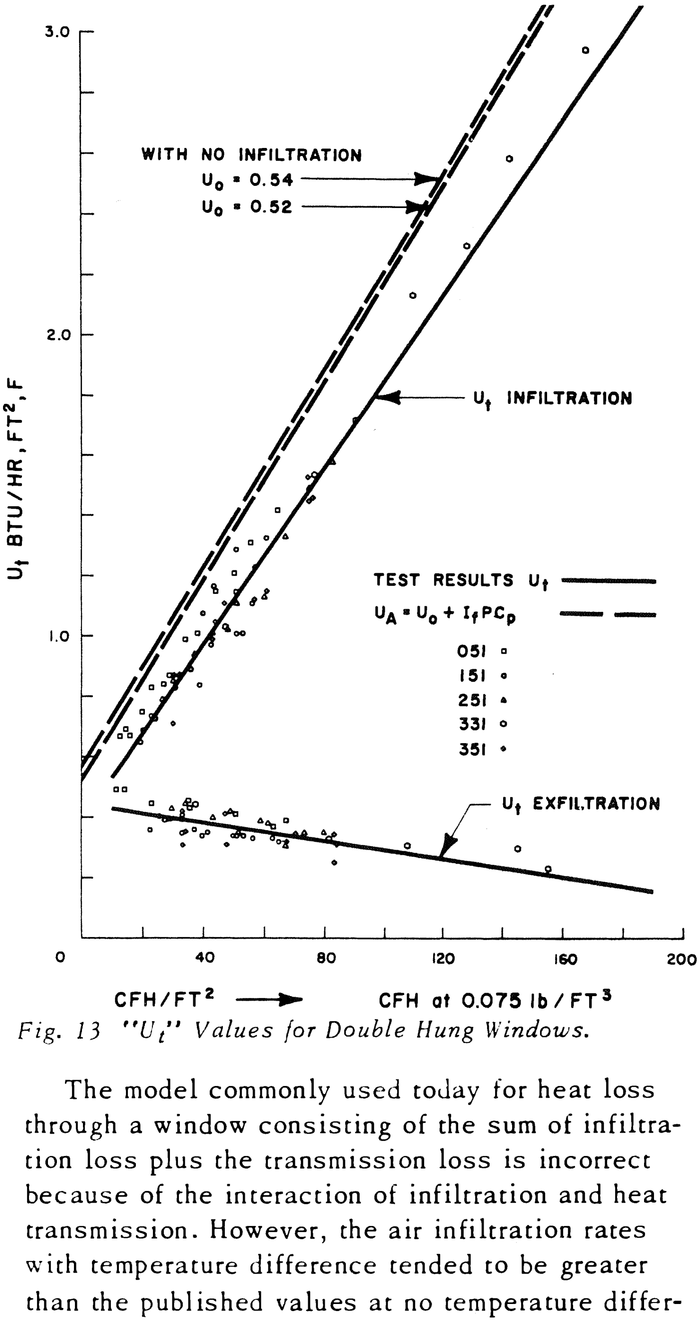


Fig. 13 "U_t" Values for Double Hung Windows.

The model commonly used today for heat loss through a window consisting of the sum of infiltration loss plus the transmission loss is incorrect because of the interaction of infiltration and heat transmission. However, the air infiltration rates with temperature difference tended to be greater than the published values at no temperature difference, and the transmission of heat with the combined effect (Fig. 3) was less than that calculated as necessary by the use of Eq 1 with published values of U for prime windows plus storm windows. The net result for the combined effect was compensating and probably explains why the use of Eq 1 gave reasonably good results in the past.

CONCLUSIONS

1). For the double hung windows tested in this work, the mass flow rate of air increased significantly with increasing temperature difference on air infiltration and increased to a less degree on air exfiltration.

2). Providing the air leakage quantity is correct for the window concerned, calculation of infiltration and transmission losses separately through double windows will give higher heating requirements that are necessary.

3). When there is exfiltration, the space between the windows has a higher air temperature than with no exfiltration and the heat loss through the window is lower than the heat transfer rate now used based upon no exfiltration. Where exfiltration could be depended upon such as in a pressurized building these values may be utilized to determine the heating requirements of a room.

4). The concept of an apparent overall heat transfer coefficient U_t for each window, which includes the effects of air leakage, provides all of the essential information required for design heat loss calculation of double hung windows.

5). The window surface temperatures decreased markedly with infiltration and increased with exfiltration. The temperature index is essentially independent of overall temperature difference for the range of outside temperatures tested, with no air leakage, however, the temperature index varied considerably with increasing air leakage.

6). A test of a window with combined heat and air leakage is complex and time consuming and it is difficult to conceive of an industry standard at this time.

The work reported in this paper is part of a research program being carried out at the University of Saskatchewan, Saskatoon, Canada, for ASHRAE on RP73 on the thermal and air leakage performance of contemporary double windows to provide additional design information for heat loss calculations and to develop standards.

The National Research Council of Canada supported the project in its initial phases and its aid is gratefully acknowledged.

NOTATION

A – Area of window ft^2

ΔP or h – differential heat across orifice or window in. of water

Gry – Grashof Number

hpi – inside prime surface heat transfer coefficient Btu/hr ft^2 F

hpo – outside prime surface heat transfer coefficient Btu/hr ft^2 F

hsi – inside storm surface heat transfer coefficient Btu/hr ft^2 F

hso – outside storm surface heat transfer coefficient Btu/hr ft^2 F

H_t – total heat flow Btu/hr

I – infiltration rate – cu ft/hr/ft of crack

I_f – infiltration rate – cu ft/hr/sq ft of window area

I_o – infiltration rate – cu ft/hr/ft of crack at 0 FΔt

L – thickness of window air space (in.)

N – characteristic pressure drop exponent

T_a – window air space temperature (F)

T_i – air temperature test chamber (F)

T_o – air temperature cold room (F)

U – overall heat transfer coefficient Btu/hr ft^2 F (without air leakage)

U_i – overall heat transfer coefficient across prime Btu/hr ft^2 F

U_o – overall heat transfer coefficient across storm Btu/hr ft^2 F

U_t – apparent overall heat transfer coefficient Btu/hr ft^2 F (including the effects of air leakage)

V – Wind velocity (mph)

X – distance from bottom of window to thermocouples (in.)

Y – height of window (in.)

ρ – density of air .075 lb_m/ft^3

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DISCUSSION

T. FAISON, (National Bureau of Standards, Washington, D.C.): In your exfiltration tests was the air dry or moist? From your test results the rate of air penetration increases with a change in temperature. It has been my experience with exterior wall systems that as the temperature difference increases (the outdoor temperature is lowered) the rate of moist air penetration through the wall is lower because of condensation and freezing within the structure.

MR. BURSEY: Many of our exfiltration tests were performed in the summer with humidities that were fairly high in the range of 60 to 85%. When we tried to use this type of moist air without re-cycling it, we had condensation and frost formation problems on the storms and to a lesser degree on the prime. This, of course, reduced the exfiltration and introduced a variable which depended on the day to day fluctuations in the rh. Therefore, we decided to recycle our exfiltration air from the cold room in order to keep the dew point temperature of the air below the surface temperature of the glass. Consequently, our results are temperature dependent, but only very slightly dependent on the rh because there was no condensation involved in the results of this paper.

D. HOLMAN, (Cleveland, Ohio): Many designers pressurize buildings postulating that this offsets the infiltration. In your tests did you find anything to substantiate this concept?

MR. BURSEY: There is no doubt that pressurization of a building in relation to the outside pressure will eliminate air infiltration and cause air exfiltration. This is accomplished by the normal stack effect on the upper floors of high rise buildings above the neutral zone and during the winter season. When exfiltration occurs the heat transferred through the window itself is less because the overall U value is less than for other conditions. The volumes of air required for the degree of building pressurization depends on the tightness of the building itself and we do not have any average figures.

MR. HOLMAN: Do you have any pressurization figures to accomplish this?

MR. BURSEY: No. Of course, in our tests our main pressure from an exfiltration point of view was that corresponding to this pressure on the 30 mph wind which would be .433 in. wider; but no doubt, as you say, pressurization of the building would certainly reduce the infiltration.

G. WILSON, (National Research Council, Ottawa, Ontario, Canada): With regard to Mr. Holman's comment in connection with the possibility of pressurizing the building to take advantage of the decreased U value that Mr. Bursey suggested, it should be pointed out that there is a problem of increased condensation on windows of the type discussed if humidification is provided in winter.

MR. BURSEY: Yes. This is very true. As stated in my earlier reply to Mr. Faison's question, we encountered this problem.