DOCUMENT RESUME

ED 035 257 EF 003 915

AUTHOR Christensen, G.; And Others
TITLE Thermal Performance of Idealized Double Windows, Unvented. Research Paper No. 223.
INSTITUTION National Research Council of Canada, Ottawa (Ontario). Div. of Building Research.
PUB DATE Jul 64
NOTE 12p.; Presented at the American Society of Heating, Refrigerating, and Air Conditioning Engineers 71st Annual Meeting, Cleveland, Ohio, 1964

EDRS PRICE EDRS Price MF-$0.25 HC-$0.70
DESCRIPTORS Air Flow, Building Design, *Climate Control, Climatic Factors, Construction (Process), Controlled Environment, *Design Needs, Evaluation, Glass Walls, Performance Specifications, *Research, Temperature, Testing, *Test Results, Thermal Environment, Ventilation

ABSTRACT

The testing plans, procedures, and results of an experiment are revealed concerning the thermal performance and variable factors of unvented double windows, their heat transmission and inner surface temperature. Data are given to help improve the design and development of standards for the thermal performance of windows. Building humidity, window arrangement and condensation problems are discussed along with criteria for air-space, window frames and sashes. Test measurement charts, drawings and a reference list are included. (TG)

NATIONAL RESEARCH COUNCIL
CANADA

DIVISION OF BUILDING RESEARCH

THERMAL PERFORMANCE OF IDEALIZED DOUBLE WINDOWS, UNVENTED

BY

G. CHRISTENSEN, W. P. BROWN AND A. G. WILSON

PRESENTED AT THE AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR - CONDITIONING ENGINEERS 71ST ANNUAL MEETING, CLEVELAND, OHIO, 1964.

RESEARCH PAPER NO 223
OF THE
DIVISION OF BUILDING RESEARCH

OTTAWA

PRICE 25 CENTS JULY 1964 NRC 8085

This publication is being distributed by the Division of Building Research of the National Research Council. It should not be reproduced in whole or in part, without permission of the original publisher. The Division would be glad to be of assistance in obtaining such permission.

Publications of the Division of Building Research may be obtained by mailing the appropriate remittance, (a Bank, Express, or Post Office Money Order or a cheque made payable at par in Ottawa, to the Receiver General of Canada, credit National Research Council) to the National Research Council, Ottawa. Stamps are not acceptable.

A coupon system has been introduced to make payments for publications relatively simple. Coupons are available in denominations of 5, 25 and 50 cents, and may be obtained by making a remittance as indicated above. These coupons may be used for the purchase of all National Research Council publications including specifications of the Canadian Government Specifications Board.

G. CHRISTENSEN

W. P. BROWN

A. G. WILSON
Member ASHRAE

Thermal performance of idealized double windows, unvented

Windows usually provide less resistance to heat flow than other components of a building enclosure. In winter, the lowest inside surface temperatures generally occur on windows, and the relative humidity that can be maintained in buildings is limited by condensation on these surfaces. Double windows are widely used in northern climates to reduce window heat losses and to increase the allowable relative humidity.

Heat transmission coefficients and inner surface temperatures of double windows depend principally on the geometry of the air space, the heat transfer conditions at inner and outer surfaces, and on the design of frames and sash. Extensive measurements have been made of average heat transmission characteristics of vertical air spaces.[1] Similarly, information is available on the average surface coefficients for natural convection and various degrees of forced convection.[2]

With this information, it is possible to calculate the overall heat transmission and a mean value of the inside surface temperature. The actual inside surface temperatures, however, are variable with height, due to vertical variations in the air space temperature and in surface heat transfer coefficients. With natural convection, or low air velocity at the inner surface, temperatures near the bottom may be significantly lower than those determined on the basis of average conductances. Published information on the nature of the inside surface temperature variations is very limited.

Where double windows incorporate metal frames, sash or track, the temperatures of the inner surfaces of these components can be considerably lower than those of the inner pane and can thus place a serious limitation on building humidity, unless the unit is carefully designed. In addition, the overall heat flow through the window may be increased significantly. In order to utilize fully the advantage of double windows in permitting higher interior humidity the temperature of all components of the window should be as high as the minimum temperature on the inner pane.

With the recent tendency toward humidified buildings, it has become increasingly important to define the interior surface temperature characteristics of windows so that the windows, or the humidity, can be selected so as to avoid serious condensation problems. The thermal performance of a double window can be no better than that of the glass-enclosed air space, so that this becomes a basic yardstick in designing windows and in their rating.

This paper reports the results of measurements of inside surface temperatures on a basic double window arrangement consisting of two sheets of glass surrounded by insulated construction. Principal variables were air space width and height and overall temperature difference. Carefully controlled natural convection conditions were provided on the warm side, with forced convection on the cold side. The surface temperatures are related to temperatures measured in the air space and to the surface heat transfer conditions. In addition, results are given of the average surface-to-surface thermal conductance of each configuration.

G. Christensen is with the Danish National Building Research Institute, Copenhagen, Guest worker with Building Services Section, DBR/NRC, 1961-1962. W. P. Brown is Research Officer, Building Services Section, DBR/NRC, Ottawa, A. G. Wilson is Head, Building Services Section, DBR/NRC, Ottawa. *This paper was prepared for presentation at the ASHRAE 71st Annual Meeting, Cleveland, Ohio, June 29-July 1, 1964.*

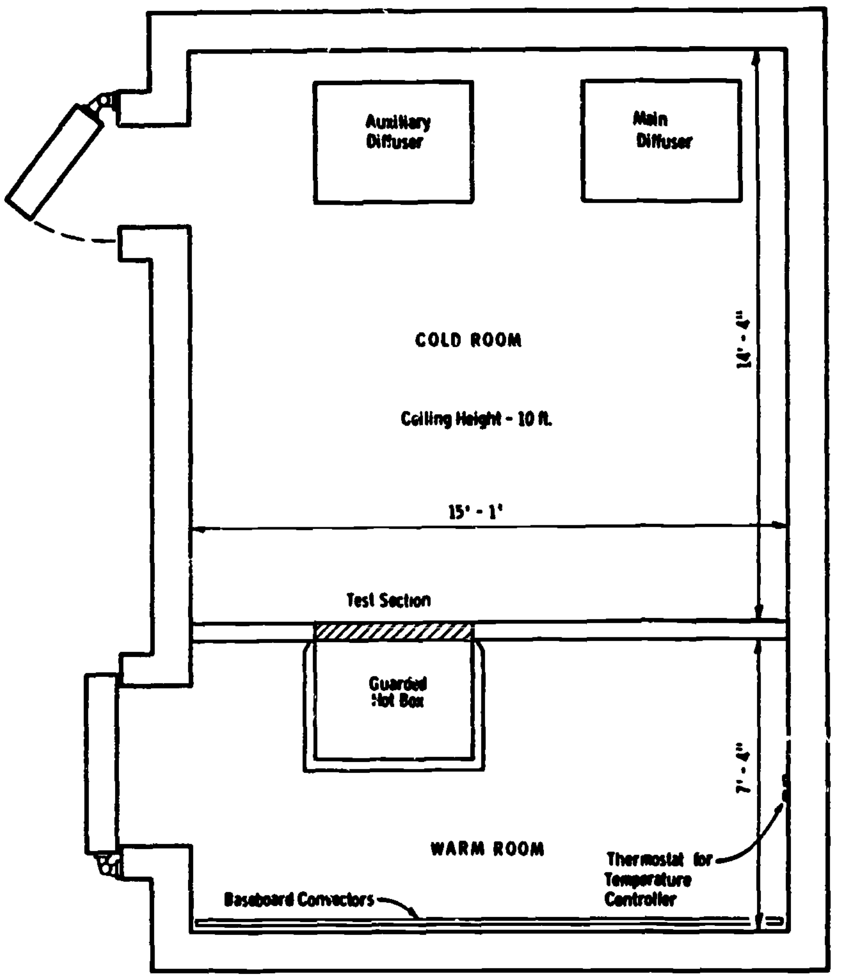

Fig. 1 Plan view of cold room

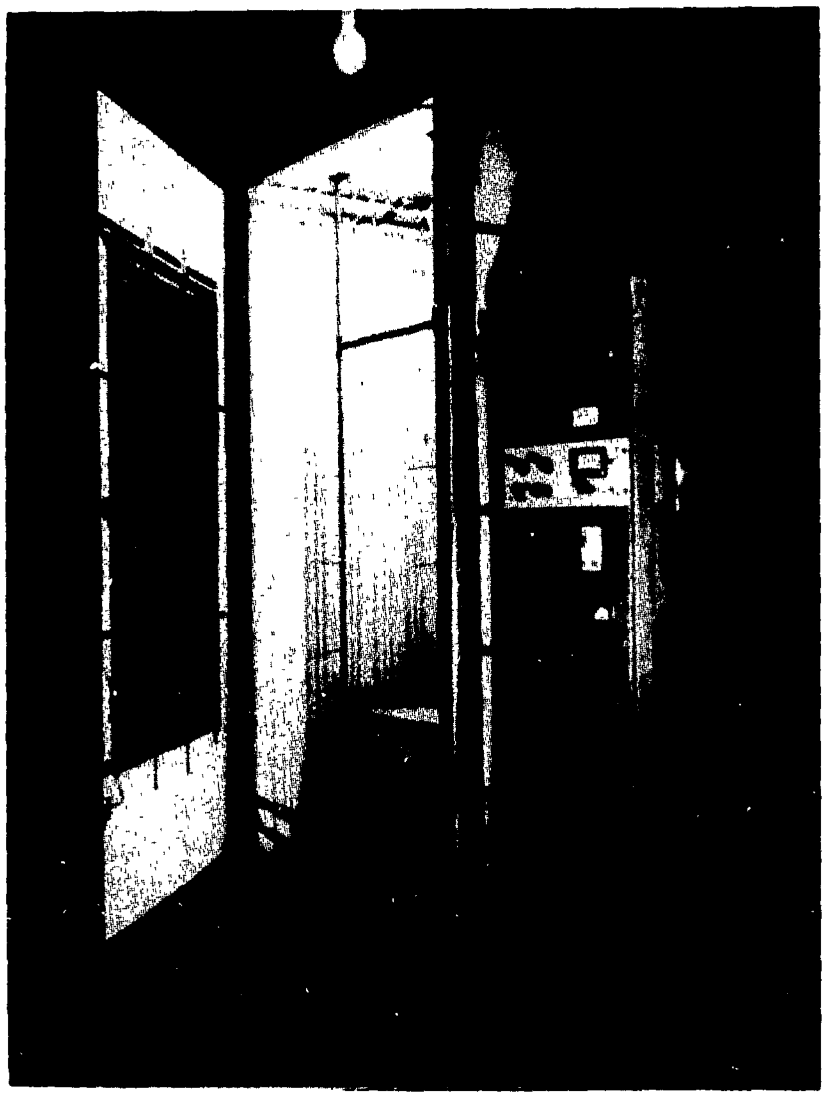

Fig. 2 Guarded hot box beside window arrangement

TEST ARRANGEMENT

All tests were carried out with a guarded hot box and associated cold room facility,[3] shown in Fig. 1. The cold room is divided by a wall into two compartments. The cold side can be controlled at temperatures from +40 F to —60 F with air temperature variations at any point less than ±0.1 F, and a floor to ceiling gradient of less than ±0.5 F. Air is discharged horizontally at ceiling level from the diffuser in use and returns through an opening in the front of the diffuser near the floor. Air velocity at the test wall is approximately 5 mph in a downward direction. The warm side is heated by electric gravity baseboard convectors and is controlled to ±0.25 F at the thermostat.

The guarded hot box (Fig. 2) is designed to measure heat flow through a 4-ft-wide by 8-ft-high test section placed between the warm and cold sides. The box is lined with radiant heating panel through which constant temperature liquid is circulated to provide uniform surface temperatures. In these tests air circulation in the box was by natural convection. At the greatest heat input required, the maximum air temperature gradient from top to bottom was about 4½ F, although the maximum air temperature variation over the height of the test window was less than 3 F. The average air temperature was approximately 3 F lower than the box surface temperature. The average heat transfer coefficients at the specimen surface provided by the apparatus are close to the value for vertical surfaces given in the ASHRAE Guide And Data Book for natural convection conditions.

The test window consisted of two sheets of 3/16-in. glass, separated by an air space, mounted in an opening, 3 ft wide by 5 ft high, which was centraily located in a panel of insulated construction, 4 ft wide by 8 ft high (Fig. 3). The inner pane was permanently mounted flush with the inside surface of the panel. The outer pane was movable so that the air space thicknesses could be changed without moving the hot box. Air space thicknesses of ¼, ½, ¾, 1, 2, 4, and 6 in. were investigated for the 5-ft-high window. Measurements were also carried out on a 2½-ft-high window, for which the upper half of the opening was filled with rigid insulation; air space thicknesses were ¼, ½, ¾, 1, 2, and 3 in. Inner and outer panes were sealed into the opening with tape to prevent air leakage. A series of temperature measurements were made with a light plywood frame around the interior perimeter of the 5-ft-high window configuration, projecting 3 in. and 8 in. from the inside surface, so as to simulate recessing of a window into the wall in which it is installed.

Air temperatures on the warm and cold sides and in the air space were measured with 30-gauge copper-constantan unshielded thermocouples, positioned vertically as shown in Fig. 3. On the warm and cold sides two thermocouples were located at each level, 1 ft on either side of the vertical centerline. For the 2½-ft-high window the air space thermocouples were located at the same relative heights as for the 5-ft-high window.

The air space thermocouple junctions were supported by horizontal ⅛-in. diameter solid brass rods soldered to a vertical ¼-in. dia brass tube, 8 in. away, through which the lead wires were carried. In this way flow conditions near the junctions were not affected by the vertical support. The support was held in place by compression against the top and bottom surfaces of

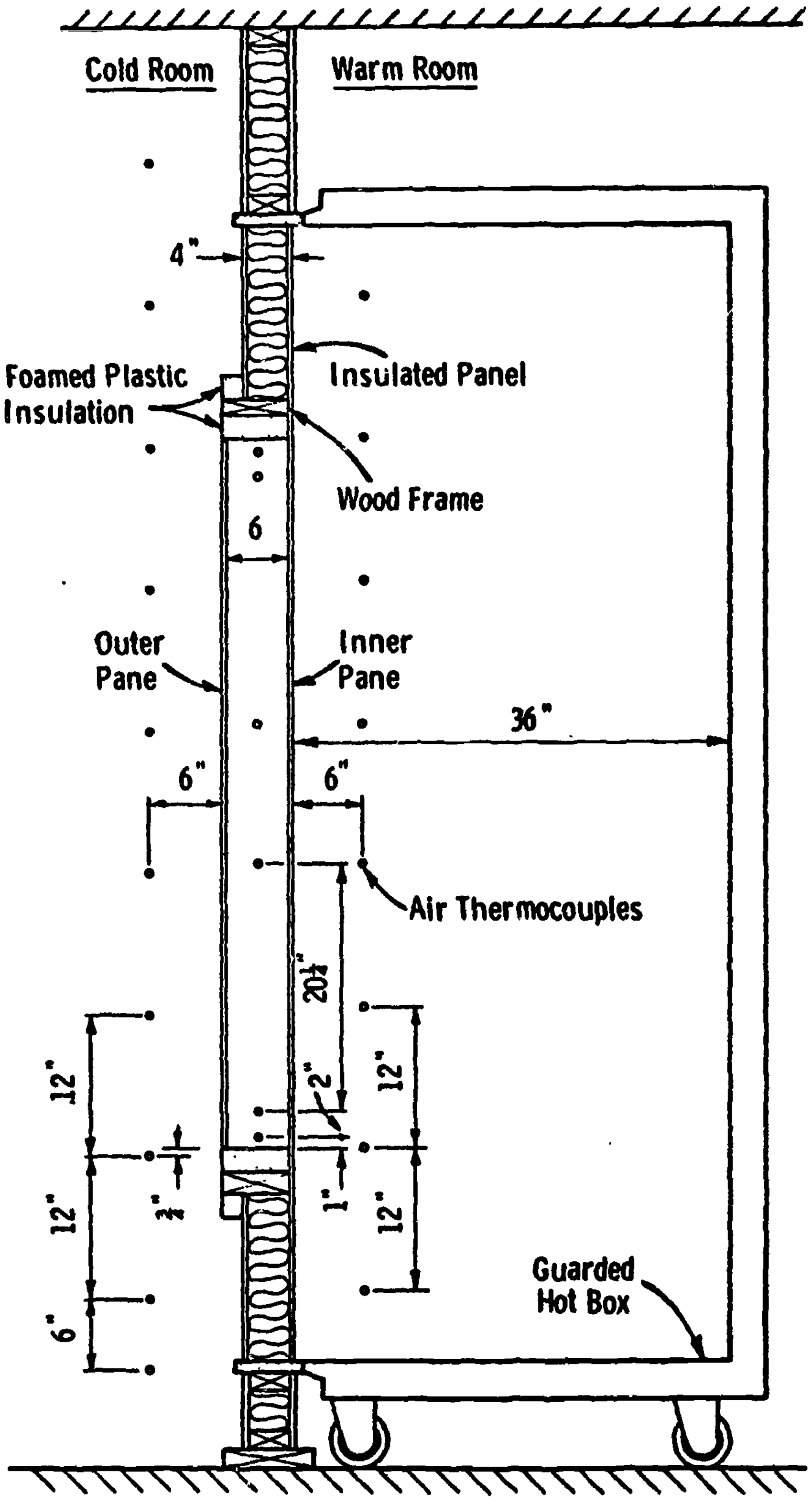

Fig. 3 Test arrangement

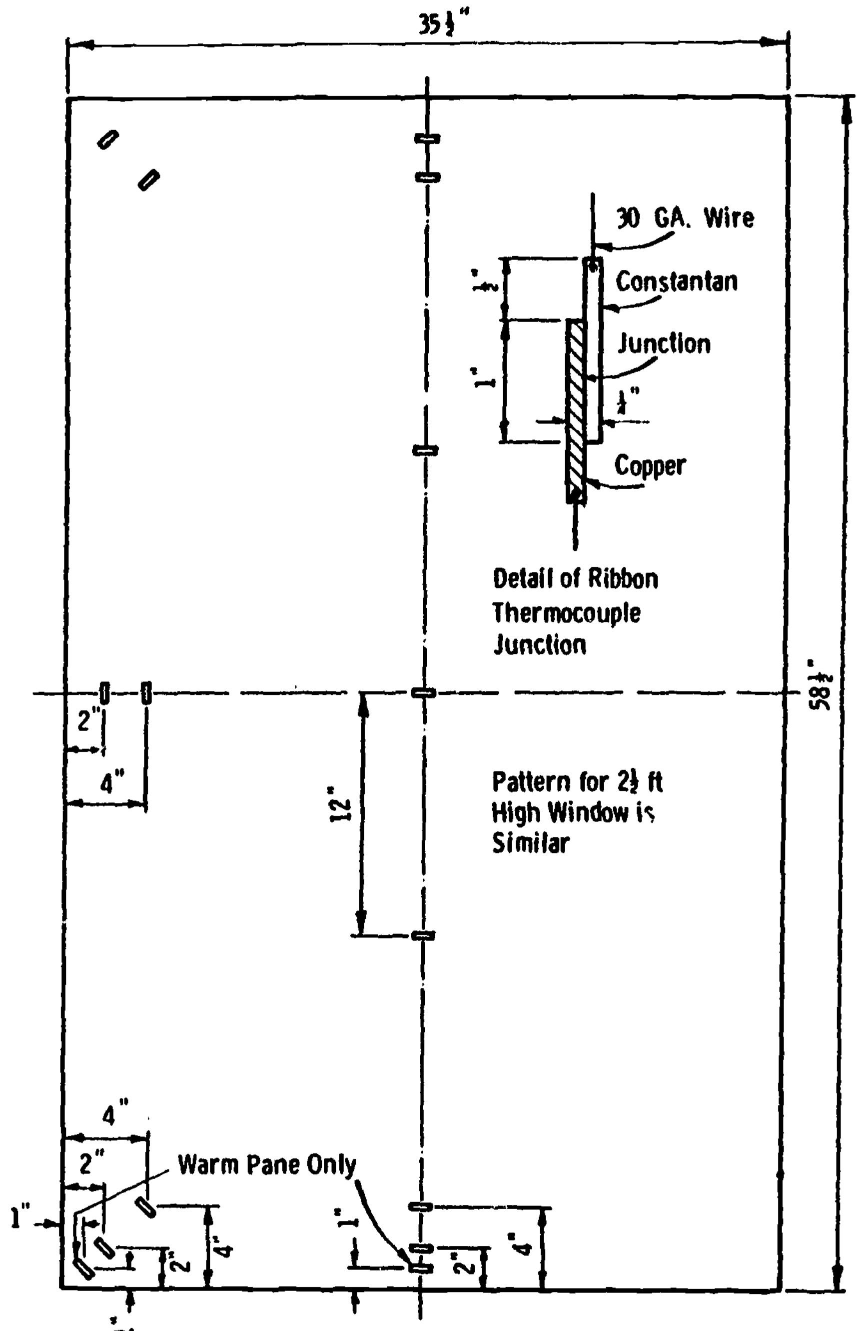

Fig. 4 Surface thermocouple locations on 5-ft window, viewed from warm side

the opening and was positioned so that the junctions were at the center of the air space. Air space temperatures are reported only for spaces 1 in. and wider, since discrepancies due to positioning of the junctions became excessive at smaller air space thicknesses.

Because of net radiation heat exchange between the thermocouple junctions and adjacent glass surfaces, air temperatures measured on the warm side could have been as much as 1 F lower than true air temperature at the lowest cold room temperature. The error in air temperatures measured on the cold side was much less than this due to smaller temperature differences between air and glass surface and forced convection at the junctions. Errors in air space temperature measurements were due both to radiation exchange with adjacent glass surfaces and to inaccuracies in positioning of junctions. The error in positioning the junctions was about $\pm 1/32$ in. and the effect of this was more significant with the 1- and 2-in. air spaces in which horizontal temperature gradients at the center were large. Measured temperatures could have been as much as 1.5 F too low at the top of the air space and 1.5 F too high at the bottom for the lowest cold room temperature.

Glass surface temperatures were measured with thermocouple junctions of 0.002-in.-thick copper-constantan ribbon fabricated as shown in Fig. 4. The junctions were soldered to 30-gauge copper-constantan lead wire. Junctions and connections were glued to the outside surface of each of the panes in the pattern shown. The error with this type of junction was determined and found to be small; using measured corrections it was possible to reduce it to less than ± 0.25 F for surface temperatures measured on the warm side.

All temperatures were measured with an electronic self-balancing indicator capable of giving temperature difference with an accuracy of ± 0.2 F.

TEST PROCEDURE

All 5-ft-high window configurations were tested at cold room temperatures of +10 F, —10 F and —30 F and the 2½-ft-high windows at +10 F and —30 F. The guarded hot box, which was positioned against the warm side of the insulated panel containing the window arrangement, was maintained at a constant temperature of approximately 70 F. The box was sealed to the panel surface with tape to prevent air leakage. To

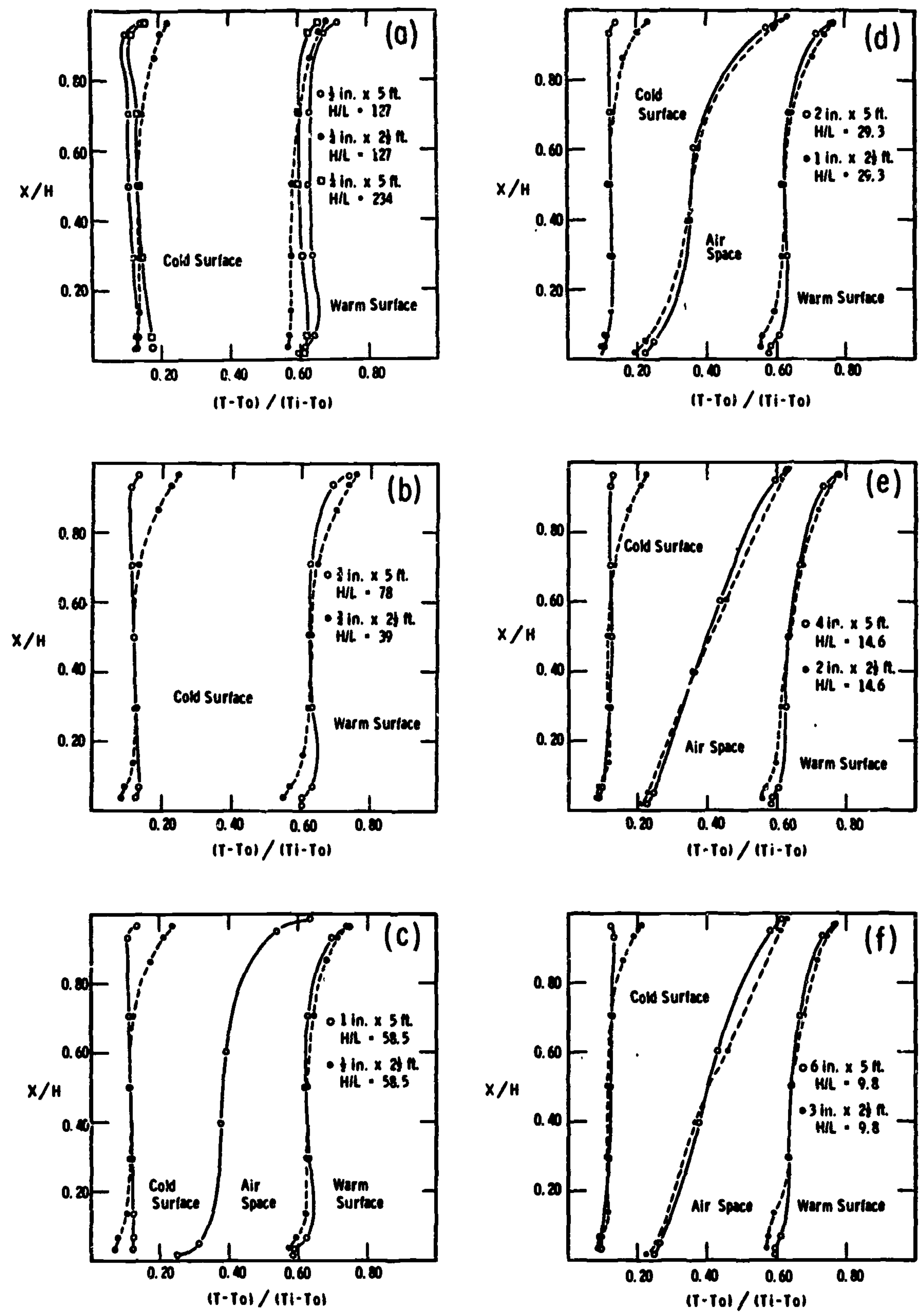

Fig. 5 Temperature index values at vertical center line

minimize edge heat leakage the average warm room air temperature surrounding the guarded hot box edge, measured with thermocouples, was controlled to within 0.5 F of the average temperature of the air in the box.

For each test condition measurements were made of all air and surface temperatures and power input to the guarded hot box. All measurements were made at steady-state heat flow conditions. The duration of steady-state prior to any reading was approximately 10 hr for all tests. During this period the power input to the guarded hot box was recorded. At steady-state, variations in power input were less than 2 watts; an average taken over several hours, therefore, provided good accuracy.

Prior to the window tests the insulated panel containing the window opening was calibrated to obtain the heat flow through it as a function of air-to-air temperature difference. This was done by installing a 4-in.-thick uniform blank of rigid insulation in the opening and measuring the heat flow through the panel-blank combination with the guarded hot box at the same cold room temperatures as used for the window tests. The thermal conductivity of representative samples of blank material had been determined in a guarded hot plate over a range of mean temperatures. The average temperature difference across the blank was measured with 11 thermocouples on each surface, and the heat flow through the blank was calculated from this temperature difference and the known conductivity.

The heat flow through the insulated panel containing the blank was then determined for each cold room temperature by subtracting the heat flow through the blank, from the heat flow through the panel-blank combination. In the tests on the window arrangements the heat flow through the window was determined by subtracting the heat flow through the insulated panel, calculated from the calibration data, from the total heat input to the guarded hot box. Heat flow through the insulated panel was 14 to 16% of the total heat flow through the panel-window combination.

The warm surface temperatures on the insulated panel during all window tests were observed to be the same as during the calibration tests except within 3 in. of the window opening. Surface temperatures on the panel adjacent to the opening were measured at top, bottom, and one side and were found to be lower than over the remainder of the panel. These lower temperatures were due primarily to lateral heat flow from the panel across the boundary of the opening. During the calibration tests this lateral flow of heat was into the blank and around it to the cold room. With the narrow air space window arrangements the lateral heat flow largely bypassed the window; with the widest air spaces there was heat flow from the panel to the air space near the warm pane and from the air space to the panel near the cold pane; at intermediate air space thicknesses the lateral heat flow introduced a net heat gain to the air space.

Because of these differences in lateral heat flow the heat flow into the panel adjacent to the window opening was greater during the window tests than during the calibration tests. This heat flow was calculated from the temperature measurements and the average surface conductance. The average surface conductance was about 1.5 for the panel and about 1.35 Btu per (hr) (sk ft) (deg F) for the windows. A mean value was assumed for the vicinity of the opening. The differences in heat flow into the panel adjacent to the opening amounted to 6% of the heat flow through the 5-ft window with 6-in. air space, increasing to 7% for the 1/4-in. air space. For the 2 1/2-ft-high window the differences varied from 8 to 10% of the window heat flow.

The extent to which the lateral heat flow influenced the heat transfer through the window is indeterminate. In calculating the window heat flow as outlined above, the heat flow through the insulated panel, based on the calibration tests, was increased by an amount equal to one-half the calculated additional lateral heat flow during the window tests. This procedure limits the error in window heat flow due to lateral heat flow to about ± 3 to 3 1/2% for the 5-ft window and ± 4 to 5% for the 2 1/2-ft window.

The total error in the measurement of heat flow through the window arrangement depends upon the error in measuring heat flow through both the panel-blank and panel-window combinations as well as the error in determining heat flow through the blank. The total error is also dependent on whether the errors are systematic or random. Errors in heat flow measurement using the guarded hot box are due mainly to heat leakage at the edge of the test area. When warm room air temperature around the edge of the box is approximately equal to box air temperature (within 1/2 F) it has been shown that edge heat leakage provides a systematic error of approximately 3%.[3]

Since the box was operated at similar conditions for both window and calibration tests the systematic errors due to edge heat leakage are subtractive. Random heat flow measurement errors in the guarded hot box were approximately 1 to 1 1/2%. It is estimated that the error in the determination of heat flow through the blank was approximately 3%, although this error is not important since the heat flow through the blank was small compared to that through the window. Based on these considerations it is estimated that the error in the determination of heat flow through the window arrangement, exclusive of errors due to lateral heat flow in the insulated panel, were less than 2 1/2% for the 5-ft window and 4 1/2% for the 2 1/2-ft-high window.

TEMPERATURE RESULTS

The results of the vertical center line temperature measurements for the various basic window configurations are shown in Fig. 5. 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 results given for any one configuration are the average of the results obtained at the three cold room temperatures; the temperature index values at —10 F are very close to the average values; those at —30 F and +10 F differ from the average by less than 1%.

All of the inside surface temperature gradients exhibit an S-shaped profile and reflect to some degree the temperature gradients in the air space. The variations in both surface and air space temperatures from top to bottom are greatest for the larger air space thicknesses. It will be noted that the shape of the vertical air space temperature profiles varies with air space thickness. Eckert and Carlson have shown,[4] for air spaces bounded by isothermal surfaces, that the shape of these profiles depends upon the non-radiative heat flow regime that exists in the air space.

These regimes, in turn, depend on the value of the Grashof number, G_{rL}, and on the height-to-thickness ratio, H/L. In the "conduction regime," obtained when G_{rL} is small and H/L is large, heat is transferred mainly by conduction and the temperature drop from the hot to the cold surface is linear over the central part of the space; convective effects occur at upper and lower edges of the layer only. When G_{rL} is large and H/L small, the heat is transported mainly by convection and the temperature gradient is concentrated in thermal boundary layers on both vertical surfaces. The temperature of the core is essentially constant along horizontal lines and conduction heat transfer is therefore very small. Conditions under which such a temperature field exists are referred to as the "boundary layer regime." For intermediate G_{rL} and H/L values the boundary layers appear to come together and the horizontal temperature profiles are curved over the width of the space at all levels; the heat transport is by both conduction and convection and the condition is called the "transition regime."

AIR SPACE TEMPERATURES

Typical vertical air space temperature profiles obtained by Eckert and Carlson for the different heat flow regimes with isothermal bounding surfaces are given in Fig. 6. Some comparable window air space temperature gradients obtained in this study are also shown. They are plotted non-dimensionally as the ratio of the difference in temperature between air and cold surface to the surface-to surface temperature difference across the space.

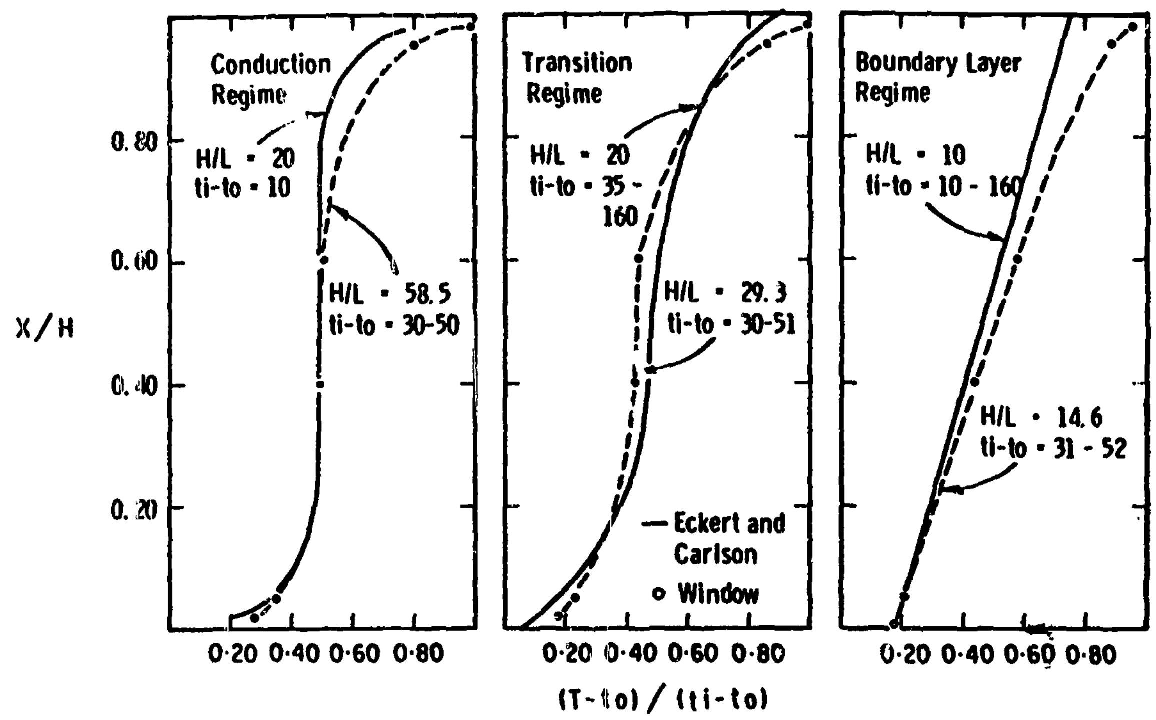

Fig. 6 Temperature profiles at vertical center line of air space

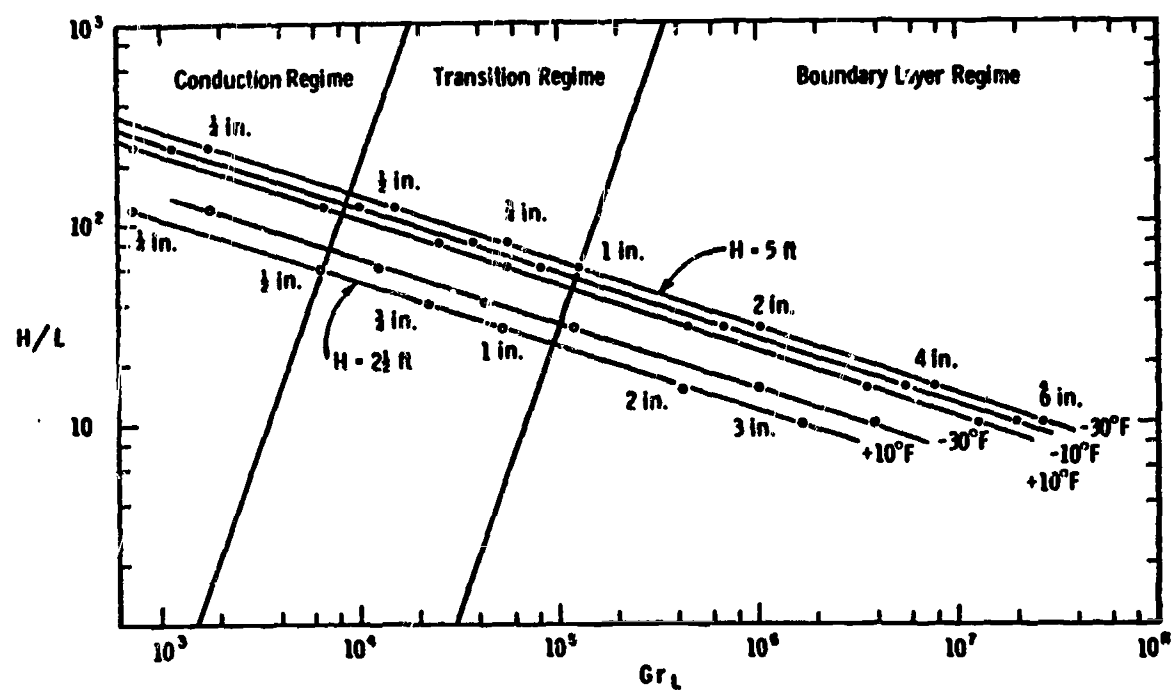

Fig. 7 Test results relative to flow regimes, after Eckert and Carlson

These center line temperatures are essentially constant in the conduction regime for some distance above and below mid-height with temperature variations at the upper and lower boundaries only; the upper temperatures move toward the warm surface and the lower temperatures toward the cold surface temperature. These regions of varying temperature increase in size with increasing G_{rL} until they meet at mid-height in the transition regime. In the boundary layer regime the vertical temperature profile is practically linear. The limits of the various regimes in terms of G_{rL} and H/L as determined by Eckert and Carlson are given in Fig. 7, together with the results of the window air space temperature measurements.

According to the limits given by Eckert and Carlson, the results for the ¼-in. air space fall within the conduction regime, those for the ½-, ¾-, and 1-in. spaces fall generally within the transition regime, and those for the 2-, 3-, 4-, and 6-in. spaces are within the boundary layer regime. The actual shape of the centerline profiles for the 1-in. air space suggests that it is closer to the conduction regime than is indicated by Fig. 7 when 5 ft high (Fig. 5c), but is definitely in the transition regime when 2½ ft high (Fig. 5d). The 2-in. air space, 5 ft high, which is geometrically similar to the 1-in. air space, 2½ ft high, is also in the transition regime according to the temperature profiles. The shape of the profiles obtained for the 2-in.-thick by 2½-ft-high and the 4-in.-thick by 5-ft-high-air spaces (Fig. 5e) on the other hand indicate that they are within the boundary layer regime. The 3-in.- and 6-in.-thick air spaces (Fig. 5f) are also within the boundary layer regime. It would seem that the transition regime in window air spaces extends to greater air space widths than in air spaces bounded by isothermal surfaces.

SURFACE TEMPERATURES

It has been pointed out that the inside surface vertical temperature profiles generally reflect the air space temperature pattern and result in inside surface temperatures being lower at the bottom than at the top. This applies even to the narrow air spaces with heat transfer in the conduction regime where convection effects, nevertheless, occur at the top and bottom. In this connection Eckert and Carlson point out that at the bottom edge of the warm surface the local heat transfer coefficient in the air space is significantly higher than in the central region and is lower at the corresponding top edge. The

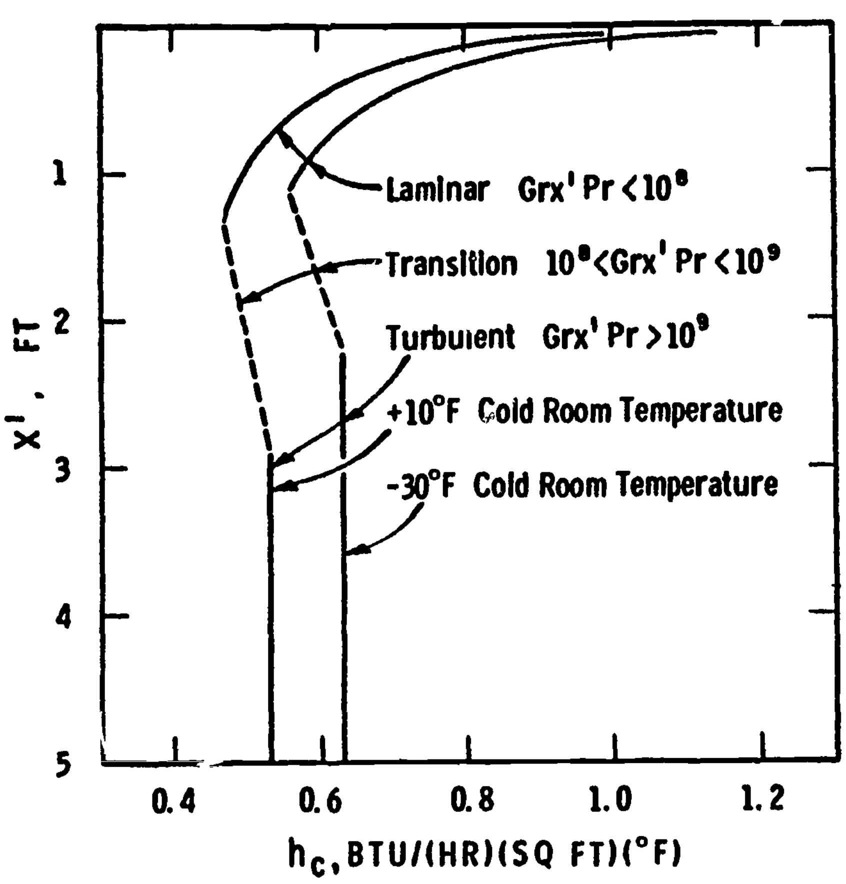

Fig. 8 Convection heat transfer coefficient over inside surface of ¼-in. x 5-ft window

situation is reversed at the top and bottom edges of the cold surface. This further explains the inside surface temperature gradients in the vicinity of upper and lower edges of the window, as well as at the outside surface. It can be calculated[4] that the height of the convection region from the bottom of the window is less than 1 in. for the ¼-in. air space, but increases significantly with increasing air space thickness. This is evident in the difference between the surface temperature gradients at the bottom of the ¼- and ½-in. air spaces.

In the case of the thicker air spaces, with heat transfer in the transition and boundary layer regime, the thickness of the laminar flow boundary layer in the air space adjacent to the warm surface increases with distance from the lower edge; similarly for the cold surface it increases with distance from the upper edge. The result is higher than average heat transfer coefficients at the bottom adjacent to the warm surface, which are reflected in the large warm surface temperature gradients near the bottom for the wider spaces. At the very bottom, however, the gradient decreases. It is assumed that this is due to radiation exchange between the glass near the bottom and the adjacent insulated boundary which is relatively warm. With the narrow air spaces, the angle of view between the glass and the insulation is smaller and radiation exchange reduced.

Since this effect is not evident in the surface temperatures at the bottom edge of the narrow air spaces it is assumed that conduction into the glass from the insulation is not significant. It will be noted that the inside surface temperature for the narrow air spaces is fairly constant over a small central region consistent with the air space temperature profile for the conduction regime (Fig. 6). With the 5-ft height, however, the temperature increases slightly from the center downwards until the lower edge is approached. This rise can be explained by consideration of the convection conditions on the warm side.

Natural convection air flow occurs in a vertical downward direction over the inside window surface. Natural convection can be described in terms of N_{ux}', G_{rx}', and P_r, where X' is the vertical distance from the top of the pane. The variation in the P_r number is small over the range of air temperatures and can be assumed constant, equal to 0.71. When $G_{rx}' . P_r$ is less than about 10^8 the flow is laminar and the convection heat transfer rate is described by the following equation, taken from Reference (5):

$$N_{ux}' = 0.378 . (G_{rx}')^{1/4}, \text{ or } h_c = \frac{K}{X'} . 0.378 . (G_{rx}')^{1/4} \quad (1)$$

When $G_{rx}' . P_r$ is greater than about 10^9 the flow becomes turbulent and in this range the heat transfer coefficient is described by the following equation, taken from Reference (5):

$$N_{ux}' = 0.1 (P_r . G_{rx}')^{1/3} \text{ or } h = \frac{K}{X'} 0.1 (P_r . G_{rx}')^{1/3} \quad (2)$$

Values of the convective component of the surface conductance for the inside surface of the 5-ft window with ¼-in. air space, calculated from Eq (1) and (2) for cold room temperatures of 10 F and —30 F, are shown in Fig. 8. This indicates very high values near the top edge, rapidly decreasing in the downward direction with transition from laminar to turbulent flow between 1 to 3 ft from the top, and a constant value when turbulent flow is established.

The rise in heat transfer coefficient with turbulent flow causes the rise in surface temperature from the center downwards with the narrow air spaces of 5-ft height. This rise is most pronounced with the ¼-in. air space where it can be assumed that there is little vertical temperature gradient in the air space, except adjacent to upper and lower boundaries, but is evident with air spaces up to 2 in. in width.

At greater air space widths, the effect of the transition from laminar to turbulent flow in the boundary layer at the inside surface is masked by the influence of the large air space temperature gradients. No rise in inside surface temperature occurs toward the bottoms of the narrow air spaces of 2½-ft height since laminar flow occurs over most or all of the height of the window. The large temperature gradients of warm surfaces at the top edge with all air space thicknesses would appear to be due in part to the high convective heat transfer coefficient at this edge, which rapidly decreases in the downward direction.

On the cold side, forced convection flow was provided in a downward direction over the window surface at a velocity of approximately 5 mph. Outside surface conductances were about 3.5 for the 5-ft window and about 3.2 Btu per (hr) (sq ft) (F) for the 2½-ft window. It is almost certain that flow over the window was turbulent because of the turbulence of the free stream approaching the window due to the fan and impingement against the test partition. This would indicate a high value for the convection heat transfer coefficient at the top edge of the cold surface, decreasing in a downward direction, and would tend to cause a temperature rise in a downward direction on this surface.

This is evident in the case of the narrow 5-ft-high air spaces except at the upper edge, where a local increase in surface temperature occurs due to recessing of the cold pane in the insulated opening. With the 6-in. air space, 5 ft high, where there is no recessing of the cold pane, the cold surface temperatures increase in a downward direction at the very top, as would be predicted. Over the remaining height of the window, however, the temperatures decrease toward the bottom due to the temperature gradients in the air space and on the warm surface. This also applies to the other wider air spaces.

MINIMUM SURFACE TEMPERATURES

The value of double windows in permitting higher inside relative humidities is determined by the minimum inside surface temperatures. For the basic window arrangements in this study temperatures on the vertical centerline were always lower than those measured at corresponding levels at other locations shown in Fig. 4. In Fig. 9, vertical center line surface temperatures near the bottom of the various window configurations are compared with those at the center and with the weighted averages calculated from all surface temperatures. Included are the inside surface temperatures measured near the bottom of the 5-ft-high windows with the simulated recessing of 3 in. and 8 in.

It will be noted that the lowest temperatures occur at the bottom of the window in all but the ¼-in. air space with the 5-ft height. The difference between temperatures 2 in. and 1 in. from the bottom is small. Furthermore, the effect of air space thickness on the value of the minimum surface temperature is not great; with the simulated recessing the differences become insignificant. Lower minimum surface temperatures occur on the 2½-ft windows than on the 5-ft ones, even though center and average temperatures are generally as high, or higher. Minimum inside surface temperatures are lower with the simulated recessing due to interference with convection conditions. The center temperature is lower than the average by about 2% of the overall temperature difference.

The surface temperature parameter used in Fig. 9 and in preceding figures is a convenient one for comparison of window condensation performance and has been referred to as a "condensation index."[6] It must have a value between 0 and 1. The values of this temperature index for the basic windows of this study provide a useful yardstick for rating the performance of proprietary windows. The surface temperatures of a number of such windows have been measured under conditions similar to those in the present investigation. Several of these were horizontal sliding double windows, 2½ to 3-ft high and with air space thicknesses from 2 to 4½ in., having aluminum frames incorporating optimum thermal insulating breaks. These specimens had minimum inside surface temperatures corresponding to a temperature index of 0.55 to 0.57, which is in close agreement with values in Fig. 9 for the 2½-ft window.

Providing optimum surface temperatures becomes more difficult with narrow air spaces due in part to the relatively high conductance of sash or spacer materials.

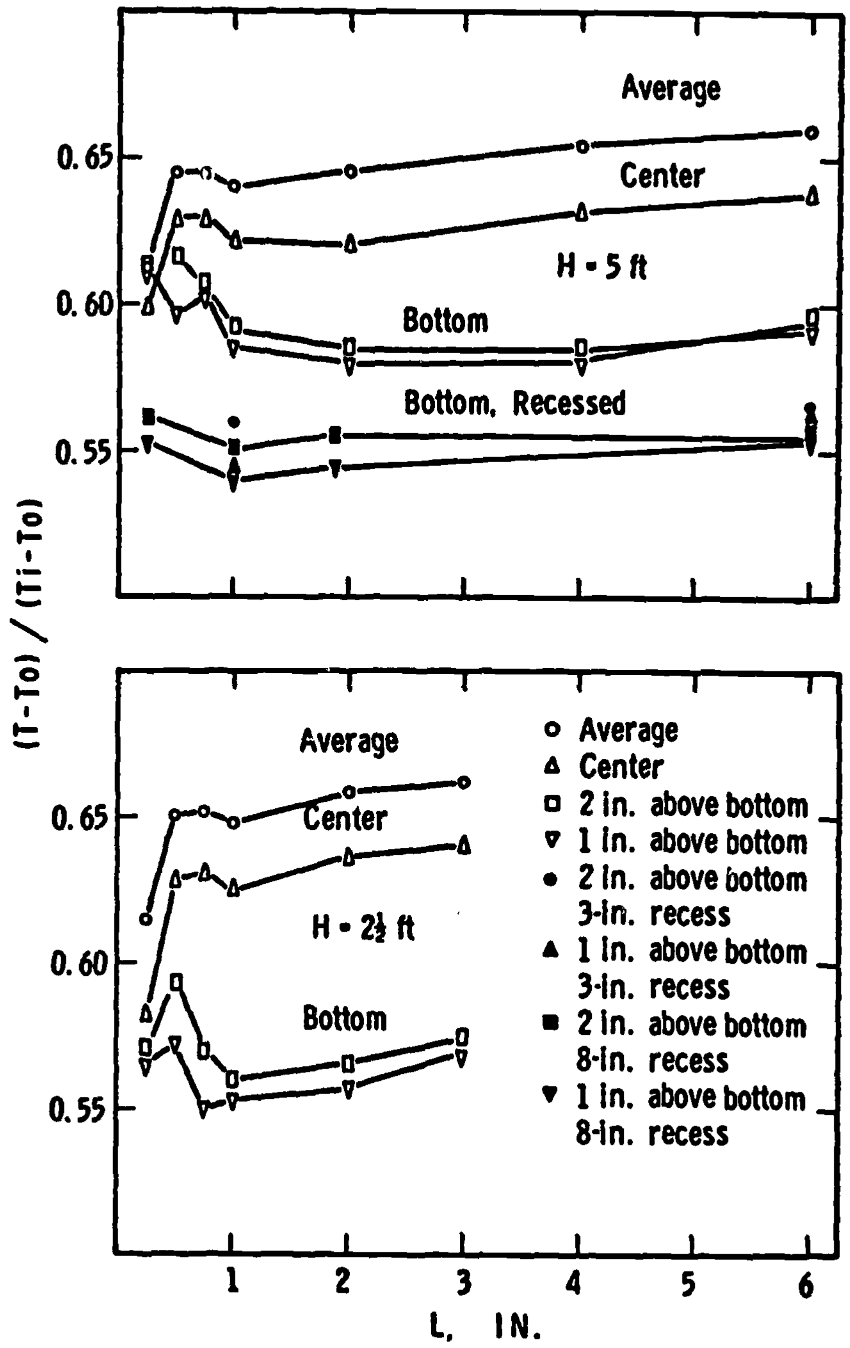

Fig. 9 Temperature index values for inside surface

The minimum temperature index of one specimen with ½-in. air space and all vinyl sash was 0.50, while that for another with ½-in. sealed double glazing enclosed in vinyl sash was 0.47. Published values[7] of surface temperatures for a number of proprietary sealed double glazing units, about 20 in. high, in wood frames gave a minimum value of the temperature index of 0.33 and lower, at the junction of glass and frame; in this case there was a recess of 1½ to 2 in. at the point of temperature measurement. The surface temperature gradients were very large near the bottom and top so that, 1 in. above the lower frame member the temperature index was about 0.53.

The temperature index of all windows can be improved by forced convection over inside surfaces but this of course increases overall heat transmission. Low surface temperatures are difficult to overcome adjacent

Table I. Relative Humidity for Condensation vs. Temperature Index

$T_i = 70$ F, $T_o = 0$ F

Index	RH, %
0.65	41
0.60	36
0.55	31.5
0.40	27
0.45	24

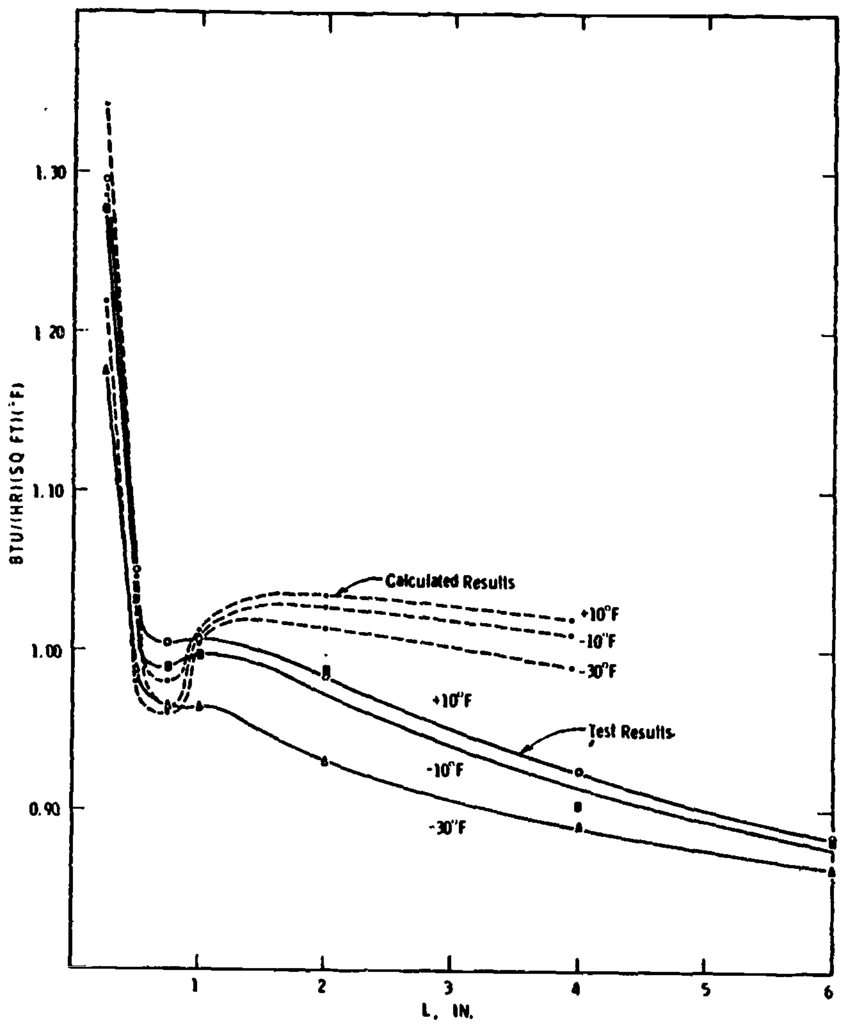

Fig. 10 Thermal conductance of 5-ft-high air space

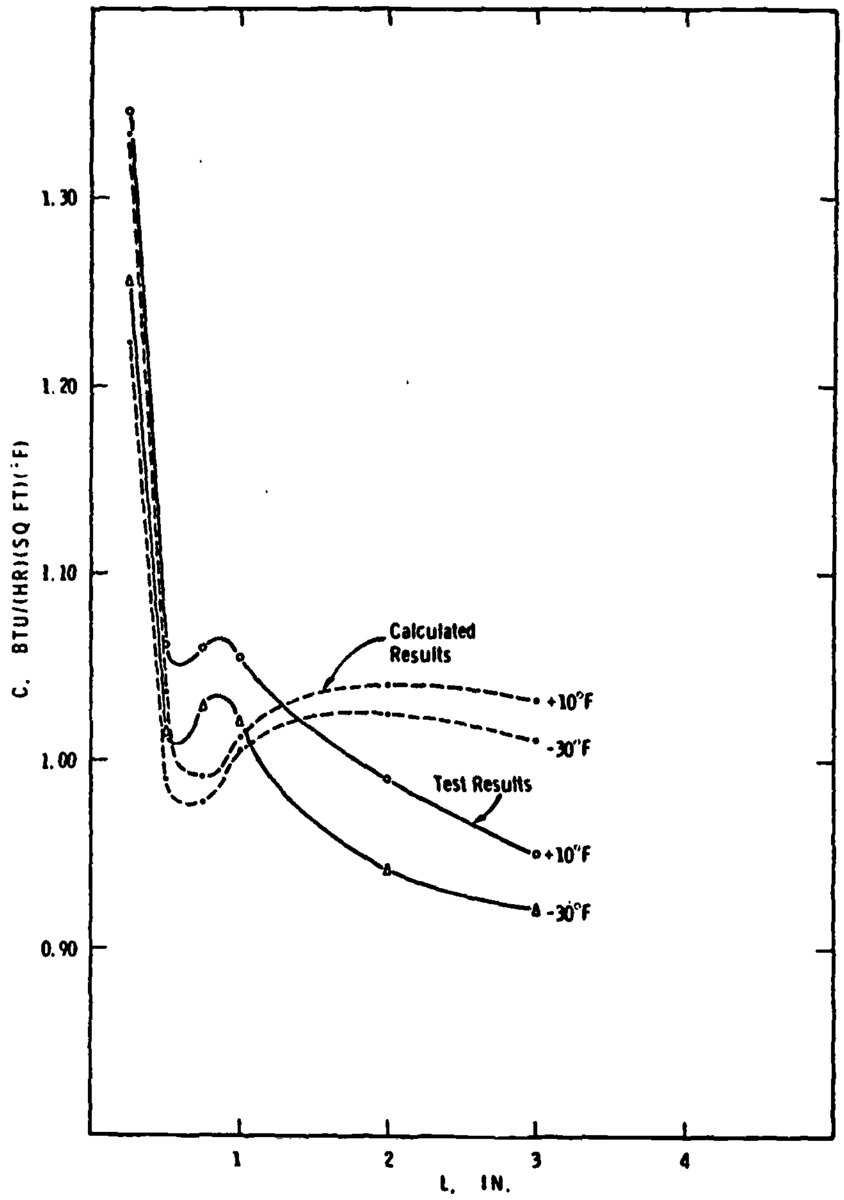

Fig. 11 Thermal conductance of 2½-ft-high air space

to a recess because of the interference with convection heat transfer.

Values of the surface temperature index for the center of the various proprietary windows already referred to correspond closely to the values shown in Fig. 9 for the basic window; in some instances the values were slightly higher due to increased heat flow through the frames into the air space. It is common practice to base the allowable building humidity on window surface temperatures calculated from average thermal conductance or U values. These calculated average surface temperatures correspond to a temperature index of about 0.65 which is close to the temperature index based on average surface temperatures in Fig. 9, compared with a minimum value of 0.55 for a well designed double window. Table I gives values of the humidity at which condensation occurs for various values of the temperature index, with inside air at 70 F and outside air 0 F. It will be noted that the relative humidity that can be carried is significantly overestimated when based on average surface temperature.

THERMAL CONDUCTANCES

Air space conductances obtained for the 5-ft and 2½-ft window configurations are given in Figs. 10 and 11. These are compared with values calculated from published data.[1] The conductance values for this present study were calculated from the window heat flow values and from average values of the measured surface temperatures. In determining the average surface temperatures the measured temperatures were weighted for area on the basis of plotted isotherm patterns. The conductances were adjusted to account for the thermal resistance of the glass.

The variation of conductance with thickness for the basic window follows the same general pattern as the calculated values. Agreement in absolute values for the narrower air space widths is well within the limits of error; the large discrepancy for the ¼-in. space is probably due to a greater uncertainty in average air space thickness. The decrease in conductance with thickness at air space thicknesses greater than 1 in. is larger for the basic window. The significance of this is not known.

Air space conductance calculated from a relationship in Reference (4) for the boundary layer regime, using the average of measured surface temperatures for the basic window with the cold room at —30 F, are essentially constant with thickness at a value of 0.99 Btu per (hr) (sq ft) (F) for the 5-foot height. The relationship in Reference (4) is based on measurements with essentially isothermal surfaces and it may be that the slope in the values for the basic window is related to the non-isothermal nature of the surfaces. Tests were carried out on 5-ft window with 6-in. air space with forced convection in the guarded hot box in order to provide more uniform warm-pane temperatures. This resulted in an increase in air space conductance of about 7%.

CONCLUSION

Under winter conditions, with natural convection on the warm side, basic double windows exhibit a significant vertical temperature gradient on the inside surface, with lowest temperatures at the bottom. The shape of the surface temperature gradient depends primarily on the air space temperature distribution and on the surface heat transfer conditions. The air space temperature conditions depend upon the type of heat flow regime, which is a function of the height-to-thickness ratio and the Grashof number based on thickness. The inside surface heat transfer coefficient for natural convection is dependent on the distance from the top edge of the window.

Surface temperatures calculated from heat transmission coefficients in the ASHRAE Guide And Data Book[2] are in close agreement with average surface temperatures obtained for the basic window in this study. The calculated values also approximate closely the surface temperatures at the center of the basic window and many proprietary windows. The relative humidities that can be tolerated, which depend on minimum surface temperatures, are significantly overestimated when based on calculated average values.

The ratio of the difference in temperature between the warm surface and outside air to the overall air-to-air temperature difference is a convenient index for describing thermal characteristics of windows. It is essentially independent of overall temperature difference over a range of outside temperatures, provided that convection conditions are constant. The values of this index for the basic window of this study provide a useful yardstick for rating the performance of practical window arrangements. For optimum performance the value of the index for sash, frame, and glass would be at least as high as the minimum values for the basic window of similar dimensions.

Practical window arrangements can be constructed for which the minimum values of the index are not less than those for the basic window. With metal windows this requires the incorporation of a substantial insulating separation in frames or sash. This becomes difficult with narrow air spaces due to space limitations. In practice there are a number of factors tending to lower the value of the index, for example, recessing of windows into the wall, stools, and blinds or drapes. Correspondingly, the introduction of heat and forced convection will tend to increase the temperature index at the expense of heat loss.

Air space conductances measured in this study show variations with thickness similar to those measured by other investigators. The conductances appear, however, to decrease more rapidly with the wider thicknesses.

The work reported in this paper is part of a research program on the thermal performance of windows, intended to provide information for the improvement of design and for the development of standards. Results for vented window arrangements are being prepared for publication.

NOTATION

C = Thermal conductance, Btu per (hr) (sq ft) (F temp diff)

C_p = Specific heat of air at constant pressure, Btu per (lb) (F)

γ = Local acceleration of gravity, ft per (sec)^2

H = Height of air space, ft

h_c = Local convection heat transfer coefficient, Btu per (hr) (sq ft) (F temp diff)

K = Thermal conductivity of air, Btu per (hr) (sq ft) (F per ft of thickness)

L = Thickness of air space, ft or in.

T, t = Temperature of surface or air, F
- T_i, average air temperature in guarded hot box
- T_o, average air temperature in cold room
- t_i, average temperature of warm pane
- t_o, average temperature of cold pane

X = Distance from bottom edge of window, ft

X' = distance from top edge of window, ft

Dimensionless Parameters

G_{rL} = Grashof number based on $L, \beta \gamma \rho^2 L^3 (t_i - t_o)/\mu^2$, dimensionless

G_{rx}' = Grashof number based on $X', \beta \gamma \rho^2 X'^3 (T_i - t_i)/\mu^2$, dimensionless

N_{ux}' = Nusselt number based on X', $h_c X'/K$, dimensionless

P_r = Prandtl number, $\mu C_p/K$, dimensionless

Greek Letters

β = coefficient of thermal expansion of air, (R deg)$^{-1}$

ρ = mass density of air, slugs per (ft)^3

μ = dynamic viscosity of air, lb per (hr) (ft)

REFERENCES

1. The Thermal Insulating Value of Air Spaces. Housing and Home Finance Agency, Housing Research Paper No. 32, April 1954.
2. ASHRAE Guide and Data Book, Fundamentals and Equipment, 1963.
3. A Unique Hot-Box Cold-Room Facility. W. P. Brown, K. R. Solvason and A. G. Wilson, ASHRAE Transactions, Vol. 67, 1961, pp 561-577.
4. Natural Convection in an Air Layer Enclosed Between Two Vertical Plates with Different Temperatures. E. R. G. Eckert and W. O. Carlson, International Journal of Heat Mass Transfer, Vol. 2, 1961 pp 106-120.
5. Heat and Mass Transfer. E. R. G. Eckert and R. M. Drake, Jr. McGraw-Hill, New York, 1959.
6. Unpublished Report to ASTM Committee E-6 by H. S. Wille.
7. Evaluation of Factory-Sealed Double Glazed Window Units. A. G. Wilson, K. R. Solvason and E. S. Nowak. ASTM Symposium on Testing Window Assemblies, Special Technical Publication No. 251, 1959, pp 3-16.