AIR-FLOW WINDOWS - AN EVALUATION OF THEIR POTENTIAL FOR USE

IN ARCTIC AND SUB-ARCTIC ENVIRONMENTS

by

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June 1986

Prepared for:

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ABSTRACT

Air-flow windows, developed in Scandinavia, are being considered for application in arctic and sub-arctic environments.

Air-flow windows consist of a double or triple-glazed outer sash and a single glazed inner sash. Room air is returned to the building heating, ventilating and air-conditioning system through the window cavity existing between the inner and outer sashes, thus warming the inner pane of glass.

Air-flow windows have the potential of improving room comfort and reducing building heat losses, particularly if the outdoor air requirement is greater than or at least can be matched to the air extracted through the windows.

A sample air-flow window was tested in a guarded hot box at various air flow rates at cold side temperatures ranging from -50°F to +10°F. Based on the test results, U-values were calculated for winter night time conditions. The economics of this window system are discussed. The energy balance of an air-flow window is established.
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ACKNOWLEDGEMENTS

I am indebted to Dr. John Zarling and Dr. Terry McFadden of the University of Alaska, and John Rezek and Lee Leonard of the State of Alaska, Department of Transportation and Public Facilities for their advice and suggestions in this study; to technicians Rick Briggs and Jeff Evans for their help in setting up the test apparatus; to Alan Braley for his help in computer programming; to Jackie Lemon for her hours spent in typing and proofreading; and to Jim McDaniels and Harri Rapatti of the Ekono Windo Company, Bellevue, Washington, for providing a sample air-flow window and valuable background material.

In addition, gratitude is extended to Dr. Robert F. Boehm of the University of Utah for first introducing the author to the air-flow window concept.
1.0 INTRODUCTION

In recent years greatly improved performance has been demanded of the building envelope in general and of the window in particular. The window has been a source of particular concern to energy researchers because of its potential for large heat gains and losses. As a result, many designers have explored ways of minimizing the detrimental energy related characteristics of windows, while preserving their thermal, lighting, and psychological benefits. Well-designed, a window system may actually provide a net energy benefit; poorly designed, it can be an enormous burden.

1.1 Description of Air-Flow Windows

In the air-flow window concept, room air is exhausted through the cavity between the inner and outer sashes. The inner sash is single glazed and operable. The outer sash may be double or triple glazed, and is generally fixed. Some manufacturers design the outer sash to be operable as well, so the window can be cleaned from the inside. The two sashes may be constructed of aluminum and separated by a thermal break, or they may be constructed of solid wood with a wood frame.

Either venetian or vertical louver blinds can be hung in the air cavity, their position controlled by a draw string or turning mechanism, or by an automated, motorized control that responds to solar radiation. Although expensive, automated blinds conserve more energy.

Air-flow windows come in two types—return-air, and exhaust-air (Figure 1.1).

In return-air systems, room air enters the window cavity through an opening between the bottom frame and operable inner sash, moving up the window, across the blinds, and exiting through an exhaust slit in the top of the window frame. This air returns to the central heating, ventilating and air conditioning (HVAC) equipment via ducts where it is either exhausted to the outside or mixed with makeup air and returned in part to the occupied space. A return-air window cannot be considered an independent component of the building. Rather, it is an integral component of the HVAC system.

In the exhaust-air system, air is forced into the window cavity at the
Figure 1.1 AIR-FLOW WINDOW TYPES
top by slightly pressurizing the conditioned space. The air flows downward through the cavity and exhausts to the outside at the bottom frame. A neoprene air valve located just inside the opening prevents outside air from blowing back into the cavity.

A third type of air-flow system, the supply-air window, has been proposed. In this configuration, outside air would be supplied to the building through the window cavity.

1.2 Brief History of Air-Flow Windows

Air-flow windows originated in Sweden in 1956. The first large-scale implementation took place in 1967 when the city of Helsinki incorporated them into its Building Department offices (Rapatti, 1984). Today, nearly 100 large buildings in Scandinavia and central Europe—mostly offices and hospitals—use them.

The concept has been slow to receive serious attention in America, partly because of misleading advertising. The Ekono Windo Company of Bellevue, Washington, claims its window can achieve an "effective R-17." Carda, a Swedish firm represented in America by A. O. Stilwell Co. of Buffalo, New York, states their window, "Makes conventional heating of a room unnecessary even when temperatures are below zero outside."

Such advertising disregards the physics and energy balance of an air-flow window while failing to define the term "effective R-value." Ekono has, to date, installed windows in two schools, one large office building, and a residence in the Seattle area.

Air-flow windows appear to have the best application in regions where air-conditioning or heating needs are great, in buildings that require large quantities of conditioned make-up air.

Air-flow windows are especially useful in regions with strong seasonal climate fluctuations and high solar radiation impact on buildings. Peak loads are reduced, particularly for cooling, as the air flow combined with the adjustable shading devices allows direct transport of radiation energy to the outside (Brandle and Boehm, 1981).

Brandle and Boehm (1981) evaluated return- and exhaust-air windows, comparing performance to a standard double pane insulating window. Their investigations included laboratory testing, numerous site visits to buildings
in Europe with air-flow windows, and interviews with architects and engineers who have designed and operated such buildings.

Their results showed considerable energy savings when using air-flow windows versus the conventional insulating glass type.

1.3 Problem Statement

The lack of detailed evaluation of air-flows prompted this study, since architects and engineers require much more than comparison of manufacturers' published effective U-values of conventional and air-flow systems.

The energy loss associated with air-flows consists of the transmission loss through the inner pane to the exhaust air, and of the loss through the external pane to the outside of the building. Because the outside-facing surface of the inner glazing is warmed by the air flowing through the window cavity, the temperature difference across this pane is small. It is the reduced heat transfer from the room through this first glazing that is the basis of the manufacturers' terms "effective R-value" and "effective U-value."

The air-flow's ability to capture solar heat adds another complication to its energy evaluation, along with the fact it increases mean radiant temperatures more than conventional windows, allowing the ambient temperature to be lowered\(^1\). Obviously, the architect or engineer has much more to consider than merely the R-value of the inner windowpane.

1.4 Scope of Investigations

A sample return-air window was tested in a guarded hot box to evaluate its performance under winter night time conditions with 100% of the air being returned to the occupied space. Predictions were then made of the window's performance under the same conditions when a fraction of the air was exhausted to the outside and the remainder mixed with make-up air for the habitable space.

The window was tested at flow rates ranging from 0 to 42 cfm (the

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\(^1\) The relation between mean radiant temperature and room ambient temperature is discussed in Section 4.0
The manufacturer's recommended flow rate is 1 - 4 cfm per square foot of window area and the test window area is 12 square feet) at cold side temperatures ranging from -50 to +10°F. The scope of this work did not include tests of emissivity, shading coefficient, or how well the window performed as an active solar collector. How these points affect the performance of an air-flow window system are discussed by Brandle and Boehm (1981).
2.0 TEST APPARATUS AND PROCEDURE

The Ekono Windo Company of Bellevue, Washington, provided a sample air-flow window for testing. It had frame and sashes of solid wood. The outer sash was fixed, sealed, and triple glazed, while the inner sash was single glazed and operable. The window was 3 feet wide, 4 feet high, and 4 inches deep. Venetian blinds were suspended in the cavity between sashes. Air entered the cavity through a 1/2-inch by 2-foot 9-1/2-inch opening at the bottom, which ran the width of the operable sash. It exited through a 1-inch by 10-inch slit cut into the top of the window frame. Sections of this window are shown in Figure 2.1.

The window was tested in a guarded hot box co-owned and operated by the Research Section of the Alaska Department of Transportation and Public Facilities and the Mechanical Engineering Department, University of Alaska.

The guarded hot box is an apparatus used to measure heat transfer through a test wall. It consists of a cold box, mask wall (test wall), meter box (hot box), and the guard box. A diagram is shown in Figure 2.2.

The meter box was heated by light bulbs whose electrical power input was carefully monitored. The surrounding guard box was electrically heated very close to the temperature of the meter box. Thirteen thermocouples connected in series formed a thermopile to measure the temperature difference between the guard and meter boxes, which generally averaged about 0.5°F. A positive temperature difference indicated heat transfer from the meter box to the guard box; a negative temperature difference indicated the reverse. An insignificant temperature differential between the two indicated that the principal heat loss from the meter box was through the mask wall and window assembly, as desired.

Knowing the temperature difference between the meter and guard boxes, the construction and surface area of the meter box, and the total electrical power input into the meter box, the heat transfer rate through the mask wall and window assembly could be calculated.

The mask wall holding the sample window was constructed of 1-inch pine framing with a 1/4-inch plywood skin on both sides. The wall was insulated with 3-1/2 inches of Dow extruded polystyrene insulation (one 1-1/2-inch layer and one 2-inch layer). The first layer of insulation was placed in the
Figure 2.1 Sample Window Section (courtesy of the Ekono Windo Company, Bellevue, WA)
\[ T_{mb} = T_{gb} \]

Therefore, heat transfer from meter box is through mask wall.

Figure 2.2 GUARDED HOT BOX
wall with its long edge running vertically, the second layer with its long edge running horizontally, so the layers overlapped, making the insulated portion of the wall as homogeneous as possible.

By knowing the construction of the insulated mask wall and the temperature difference across the wall, the heat transfer through the window alone could be found by calculating the heat transfer through the insulated wall and the small amount of heat transfer through the meter box, and then subtracting these quantities from the total heat input.

The manufacturer's recommendation for air flow through the inner cavity is 1 to 4 cfm per square foot of window area. To achieve this, fans of the size used to cool electronic equipment were mounted in a sheet metal duct section that was slipped into the discharge opening of the window. They were controlled by a variable-speed switch, the kind used in residential ceiling fans. Together they produced air flows up to 30 cfm. A third fan was added to achieve a flow of 42 cfm.

A photograph of the guarded hot box and data logger is shown in Figure 2.3. Figure 2.4.1 shows the mask wall containing the test window installed in the cold box. Figure 2.4.2 shows the guard box with the meter box suspended inside.

Air velocity through the duct section (exiting the window) was measured by a hot wire anemometer inserted into the duct approximately 18 inches downstream. The duct dimensions at this point were 6 inches wide by 2 inches high. Velocities were measured at five points, at one inch intervals, along the centerline of the duct cross-section. The average value of these five measurements was taken as the duct air velocity, and the air flow rate was computed using the known cross-sectional area.

The window was tested at flow rates of 0, 9, 18, 30, and 42 cfm at cold side temperatures of -50, -30, -10, and +10°F. Additionally, air flow rates of 12, 24, and 36 cfm were run at -30°F. All tests were conducted with the blinds fully retracted.

Calibrated Type T copper constantan thermocouples were attached with cellophane tape at four points on each accessible glass surface (i.e. on each side of the operable inner glazing and on the inner and outer surfaces of the sealed, triple-glazed portion). Thermocouple locations are shown in Figure 2.5. Thermocouples were also attached to the wood frame of the window at four locations on both the inside and outside surfaces, and at two different
FIGURE 2.3 Test Apparatus – Guarded Hot Box and Data Logger
FIGURE 2.4.1 Mask Wall Installed in Cold Box

FIGURE 2.4.2 Guard Box with Meter Box Suspended Inside
Figure 2.5 Thermocouple Locations

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<td>0</td>
<td>-</td>
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<td>Warm Side of Triple Glazed Unit</td>
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</table>
points on the warm and cold surfaces of the insulated mask wall. Two thermocouples measured the warm side air temperature and two measured the cold side temperature. One thermocouple was inserted in the duct beside the anemometer probe, to measure the window air discharge temperature. One thermocouple was taped to the bottom of the operable sash so that it hung down into the air stream at the window air inlet, thus measuring the inlet air temperature. Two more thermocouples measured the temperature difference across that portion of the wall where the insulation had to be cut away to allow the duct to be inserted into the air discharge slit cut into the top of the window frame.

During the course of each test, the temperatures at the thermocouple locations, the guard box/meter box temperature difference, and the meter box relative humidity were recorded every 10 minutes on both paper and magnetic tapes using a Fluke data logger and a Tektronix data cassette recorder. The watt-hour input to the meter box was also recorded at each interval.

To insure that the mask wall had reached a steady state condition prior to testing, the cold box temperature was adjusted in the evening at the conclusion of each test. The guarded hot box was then allowed to stabilize thermally through the night at this new adjusted cold box temperature before the next test was begun the following morning.

Test duration at each air flow rate and cold side temperature was approximately 8 hours. This resulted in the accumulation of a large amount of raw data—approximately 2000 temperature, relative humidity, and watt-hour input measurements for each test. This data was transferred from the Tektronix tape recorder to an IBM-PC with an IBM communications program. A computer program was then composed to reduce the data to a useful form. The program calculated the 8-hour averages of the meter box relative humidity, the guard box/meter box temperature difference, and the temperatures at each thermocouple location. The program also calculated the standard deviations of these averages.

Because parasitic power can be a consideration, it was deemed important to appreciate the costs of fan energy to achieve the desired air flows through the window. A Dwyer micromanometer capable of measuring pressure differences as small as 0.01 mm of water was installed in the exhaust duct several inches downstream from the cavity outlet to monitor pressure drops through the window. The pressure at the window air inlet was assumed to be
atmospheric. Pressures were measured at air velocities through the duct ranging from 550 to 100 feet per minute, stepping down at 50-foot-per-minute intervals. These correspond to air flows of 3.82 to 0.69 cfm per square foot of window area.
3.0 ENERGY BALANCE OF AN AIR-FLOW WINDOW UNDER WINTER NIGHT TIME CONDITIONS

The energy balance of an air-flow window was established by control volume analysis. The control surface is shown as the dashed line in Figure 3.1.

Heat is transferred from the room to the inside surface of the window by convection and radiation acting in parallel.

From Newton's Law of Cooling:

\[ Q_{r-a} = \overline{h}_{t1} * A_w * (T_r - T_a) \]  \hspace{1cm} \text{(3.1)}

where \[ \overline{h}_{t1} = \overline{h}_{r1} + \overline{h}_{cv1}. \] \hspace{1cm} \text{(3.2)}

Likewise, heat is transferred from the outside surface of the window to the outside environment, again by the parallel mechanisms of convection and radiation.

\[ Q_{d-o} = \overline{h}_{t2} * A_w * (T_d - T_o) \]  \hspace{1cm} \text{(3.3)}

where \[ \overline{h}_{t2} = \overline{h}_{r2} + \overline{h}_{cv2}. \] \hspace{1cm} \text{(3.4)}

The First Law of Thermodynamics states:

\[ Q_{\text{control volume}} + \dot{m}(h_i + \frac{V_i^2}{2} + gZ_i) \]

\[ \sum = \dot{m}(h_e + \frac{V_e^2}{2} + gZ_e) + W_{cv} \]  \hspace{1cm} \text{(3.5)}

There is no work input into the control volume and the velocity and elevation terms are assumed to be insignificant, therefore:

\[ Q_{r-a} - Q_{d-o} + \dot{m}(h_i - h_e) = 0 \]  \hspace{1cm} \text{(3.6)}

The third term in Equation 3.6 represents the sensible heat lost or gained by the air stream in the window cavity and is given the symbol \( Q_{i-e} \).
Figure 3.1 ENERGY BALANCE OF AN AIR-FLOW WINDOW UNDER WINTER NIGHT TIME CONDITIONS

\[ Q_{r-a} = \bar{h}_{t1} A [T_r - T_a] \]
\[ \bar{h}_{t1} = \bar{h}_{r1} + \bar{h}_{cv1} \]
\[ Q_{i-e} = \dot{m} C_p [T_i - T_e] \]
\[ Q_{d-o} = \bar{h}_{t2} A [T_d - T_o] \]
\[ \bar{h}_{t2} = \bar{h}_{r2} + \bar{h}_{cv2} \]
\[ Q_{r-a} + Q_{i-e} = Q_{d-o} \]
\[ (h_i - h_e) = C_p (T_i - T_e) \]  
Therefore:
\[ Q_{i-e} = m C_p (T_i - T_e) \]  

The steady state winter night time (i.e. without solar heat gain) energy balance of an air-flow window, with the control surface as shown in Figure 3.1 is therefore given by:
\[ Q_{r-a} + Q_{i-e} - Q_{d-o} = 0 \]  

Placing the term representing heat transfer out of the control volume on the right side of the equation, Equation 3.9 may be stated as:
\[ Q_{r-a} + Q_{i-e} = Q_{d-o} \]  

3.1 Effective U-Value

In establishing the performance evaluation parameter, U-effective, of an air-flow window, one more control volume will be considered. The control volume surface surrounding the entire building is shown as the dashed line in Figure 3.2. The mass flow rate of air through the windows is represented by the symbol \( \dot{m} \). That fraction of the air flow through the windows that is returned directly to the building is represented by the symbol FR. Therefore:

\[ \dot{m} = (FR) (\dot{m}) + (1 - FR) (\dot{m}) \]  

where \((FR) (\dot{m}) = \) that portion of \( \dot{m} \) returned directly to the building

\[(1 - FR) (\dot{m}) = \) that portion of \( \dot{m} \) that is exhausted to the outside.

Neglecting infiltration, assuming there is no heat storage within the control volume, and lumping the heat lost through the walls, doors, roof, and floor slab or basement into one term, the heat balance for the control volume of Figure 3.2 is given by:
\[ Q_{\text{wall}} = U_{\text{wall}} A (T_r - T_o) \]

\[ \dot{m} = (\text{FR}) \dot{m} + (1 - \text{FR}) \dot{m} \]

**Q_{\text{TOTAL}}** = HEAT SUPPLIED BY HEATING SYSTEM AND GENERATED BY LIGHTS, PEOPLE AND EQUIPMENT

\[ = Q_{\text{WALLS}} + Q_{\text{WINDOWS}} + Q_{\text{FRESH}} \]

\[ Q_{\text{FRESH}} = (1 - \text{FR}) \dot{m} c_p (T_r - T_o) \]

\[ Q_{\text{WINDOW}} = (\text{FR}) Q_{r-e} + Q_{r-a} \]

**FR** = FRACTION OF AIR EXTRACTED THROUGH WINDOWS WHICH IS RECIRCULATED TO THE BUILDING

**Figure 3.2** WINTER NIGHT TIME ENERGY BALANCE WITH CONTROL SURFACE SURROUNDING ENTIRE BUILDING
\( Q_{\text{TOTAL}} - Q_{\text{WALLS}} - Q_{\text{WINDOWS}} = (1 - FR) \dot{m} (h_r' - h_o) \ldots (3.12) \)

\( Q_{\text{TOTAL}} \) represents the heat supplied to the building by the heating system and generated by lights, people and equipment. \( Q_{\text{WALLS}} \) represents the heat loss through the walls, roof, doors, and floor slab or basement. \( Q_{\text{WINDOWS}} \) represents the heat loss through the windows. The quantity \((1-FR) \dot{m} (h_r' - h_o)\) is the sensible heating load associated with conditioning the make-up air and is given the symbol \( Q_{\text{FRESH}} \). This quantity, dictated by the building fresh air requirement, is part of the total building heating load, but of course, it is not part of the window heating load. The window heating load, for the condition where all of the air flow through the window is returned to the building, is given by Equation 3.10. Using the symbol \( FR \), Equation 3.10 may be rewritten as:

\[ Q_{\text{WINDOW}} = (FR)Q_{i-e} + Q_{r-a} \ldots \ldots \ldots \ldots \ldots \ldots \ldots (3.13) \]

An effective \( U \)-value for the window may now be defined:

\[ U_{\text{EFF}} = \frac{Q_{\text{WINDOW}}}{A_w(T_r - T_o)} \ldots \ldots \ldots \ldots \ldots \ldots \ldots (3.14) \]

3.2 Energy Balance of an Air-Flow Considering Insolation

The scope of this investigation did not include studying the effects of insolation. However, to provide a better understanding of air-flow fenestration systems, an energy balance with insolation is developed here.

Energy transfer through conventional window glass can result from a temperature difference across the glass and from direct and indirect solar radiation. Direct solar radiation, diffuse sky radiation, and reflected solar radiation may be incident upon the outer surface of the window. Part of this radiation will be directly transmitted through the glass, part will be reflected, and part may be absorbed. As discussed earlier, energy exchange takes place by the parallel mechanisms of convection between the glass outer surface and the outside air, and by long wave radiation exchange with the sky and surrounding objects. Convection and radiation exchanges also occur between the glass inner surface and the room interior. With these heat transfer mechanisms in mind, the instantaneous energy balance for a single light of sunlit glazing material may be stated as follows:
\[ I_t + U(T_o - T_r) = q_R + q_S + q_T + q_{RCo} + q_{RCi} \ldots \ldots \ldots (3.15) \]


where

- \( I_t \) = Total incidence of solar radiation (direct, diffuse sky radiation, and radiation reflected from surrounding surfaces). (Btu/hr-sq. ft.)
- \( U \) = Overall heat transfer coefficients (includes both the glazing and the frame). (Btu/hr-sq. ft.-\(^\circ\)F)
- \( T_o \) = Outside air temperature (\(^\circ\)F)
- \( T_r \) = Inside air temperature (\(^\circ\)F)
- \( q_R \) = Solar energy reflected (Btu/hr-sq. ft.)
- \( q_S \) = Energy stored in the glass (Btu/hr-sq. ft.)
- \( q_T \) = Solar energy transmitted through glass (Btu/hr-sq. ft.)
- \( q_{RCo} \) = Heat flux outward by radiation and convection (Btu/hr-sq. ft.)
- \( q_{RCi} \) = Heat flux inward by radiation and convection (Btu/hr-sq. ft.)

This heat balance schematic is shown in Figure 3.3.

The total instantaneous rate of heat gain through the glass may be expressed as:

\[ \text{Total heat admission through glass} = \text{Solar heat gain} + \text{Conduction heat gain} \ldots \ldots \ldots (3.16) \]

This may be written as:

\[ q = F \times I_t + U(T_o - T_r) \ldots \ldots \ldots \ldots \ldots \ldots \ldots (3.17) \]

where

- \( q_A \) = The instantaneous rate of heat admission through the fenestration (Btu/hr - sq.ft.)
- \( F \) = The ratio of solar heat gains to the incident solar radiation (dimensionless)

The Solar Heat Gain Coefficient \( F \) is a function of fenestration type and incident solar angle. Because of the infinite number of possible
Figure 3.3 INSTANTANEOUS HEAT BALANCE FOR A SINGLE LIGHT OF SUNLIT GLAZING

(ILLUSTRATION FROM ASHRAE HANDBOOK 1977 FUNDAMENTALS, p. 26.15)
combinations of fenestrations, incident angles, and solar radiation intensities, it would be impractical to develop a procedure for estimating the quantities in Equation 3.15 for all possible circumstances.

Instead, ASHRAE has developed a method for estimating solar heat gains through fenestrations, based on a reference standard of double strength (3.2 mm or 0.125 in.) sheet glass. The solar heat gain through this reference material, called the Solar Heat Gain Factor (SHGF) is calculated and tabulated for daylight hours of the 21st day of each month, for 17 principal orientations, for latitudes between 0 and 64° North. The Solar Heat Gain Factors tabulated by ASHRAE represent the values expected on average cloudless days.

To adjust for the different types and combinations of glazing material and shading devices used, the Shading Coefficient (SC) is defined as the ratio of the solar heat gain through a fenestration system under a specific set of conditions to the solar heat gain through the reference glazing material under the same conditions. This ratio is specific and assumed to be constant for each fenestration. Stated in equation form:

\[
SC = \frac{\text{Solar Heat Gain of Fenestration}}{\text{Solar Heat Gain of Double Strength Glass}} \quad (3.18)
\]

For any fenestration the solar heat gain will be:

\[
\text{Solar Heat Gain} = \text{Shading Coefficient, SC,} \times \text{Solar Heat Gain Factor, SHGF, for Given Orientation and Existing Conditions} \quad (3.19)
\]

The total instantaneous heat gain in Btu/hr-sq.ft. will be:

\[
q_A = (SC) \times (\text{SHGF}) + U(T_r - T_o) \quad (3.20)
\]

Based on the preceding discussion and definitions, the energy balance of an air-flow window, with insolation considered, can now be developed. The control volume approach is again employed, with the control surface surrounding the window as shown in Figure 3.4. The rates of heat transfer (in or out) between the inside glass surface and the room interior, and between the outside glass surface and the outside environment, by radiation and convection, are given by Equations 3.1 and 3.3. The sensible heat lost or gained in the window cavity is again given by Equation 3.8.
Figure 3.4 ENERGY BALANCE OF AN AIR-FLOW WINDOW WITH INSOLATION CONSIDERED
The incident solar radiation on the control surface is given the symbol \( Q_I \) and the reflected solar radiation is given the symbol \( Q_{\text{REF}} \). The solar heat gain through the window is given by Equation 3.19.

The instantaneous energy balance of an air-flow window accounting for the energy carried with the air flow and solar heat gain, is now given by:

\[
Q_{\text{r-a}} - Q_{\text{d-o}} - Q_{\text{sol}} + Q_I - Q_{\text{REF}} = \\
\dot{m}C_p(T_e - T_i) .................................................. (3.21)
\]

Using the symbol, \( FR \), and considering the entire building as the control volume, the effective U-value for an air-flow window with insolation may now be developed:

\[
(Q_{\text{WINDOW}})_{\text{insol}} = Q_{\text{d-o}} = Q_{\text{r-a}} + Q_I - Q_{\text{sol}} - \\
Q_{\text{REF}} + \dot{m}C_p(T_i - T_e) .......... (3.22)
\]

\[
(U-\text{EFF})_{\text{insol}} = \frac{(Q_{\text{WINDOW}})_{\text{insol}}}{A_w(T_r - T_o)} .......... (3.23)
\]
4.0 MEAN RADIANT TEMPERATURE

Mean radiant temperature refers to the equivalent radiant temperature required for the heat exchange between the human body and the surfaces that surround it (Brandle and Boehm, 1981). The mean radiant temperature for a person in a room is calculated as the value of all surface temperatures surrounding the person, weighted according to the magnitude of the respective radiation angle factors. Mathematically this may be stated:

\[ \text{MRT} = T_1 F_{p-1} + T_2 F_{p-2} + \ldots + T_n F_{p-n} \ldots (4.1) \]

For example, if one of the room surfaces in question is a window, its contribution to the mean radiant temperature will be through the surface temperature of the glass and its view factor with the person. The view factor will vary with the geometry and position of the window, and the distance between the person and the window. The magnitude of the view factors, for a person seated or standing within an enclosure, varies between 0 and about 0.13. The mean radiant temperature in any space will be effected by the room side surface temperature of the window.

Fanger (1970) established that thermal comfort is a function of activity level (metabolic rate), thermal resistance of clothing, ambient air temperature, mean radiant temperature, relative air velocity, and water vapor pressure in ambient air.

The purpose of the body's thermoregulatory system is to maintain a constant internal temperature. For long periods of exposure to a constant (moderate) thermal environment with a constant metabolic rate, it can be assumed that a steady state heat balance will exist for the human body—i.e., the heat production will equal the heat loss, with no significant heat storage within the body. The heat balance for this condition is:

\[ H - ED - ESW - ERW - L = K = R + C \ldots (4.2) \]

where

- **H** = the internal production of heat
- **ED** = the heat loss by water vapor diffusion through the skin
- **ESW** = the heat loss by evaporation of sweat from the
surface of the skin.

ERW = the latent respiration heat loss

L = the dry respiration heat loss

K = the conduction heat transfer from the skin to the outer surface of the clothed body

R = the radiation heat loss from the outer surface of the clothed body.

C = the convection heat loss from the outer surface of the clothed body.

Applying experimental results to the general body heat balance equation, Fanger (1970) developed a Thermal Comfort Equation, which can predict any combination of air temperature, mean radiant temperature, relative air velocity, and water vapor pressure that will achieve a comfortable environment for a person in a variety of clothing ensembles, and engaged in a variety of activities.

Applying Fanger's comfort equation, a designer can compare the required ambient air temperatures for rooms with conventional window systems and air-flow systems by calculating expected mean radiant temperatures for the two systems.
5.0 TEST RESULTS AND DISCUSSION

During this experiment measurements of temperature, relative humidity, and watt-hour input were taken every 10 minutes throughout each 8 hour test run. The averages and standard deviations were computed for each test. Using this data, effective U-values were calculated and condensation potential evaluated. The range of variation in the test results for the different cold side temperatures at each flow rate was very narrow. This range is denoted by the error bars for each flow rate in Figures 5.1 through 5.5 and in Figures 5.7 through 5.10. Error bars have been omitted on all other graphs. Pressure drop through the window as a function of air-flow rate was also measured and plotted.

The unit, "cfm/sf of window area," is the unit used by the Ekono Windo Company in describing the air flow requirements and performance of their window system. At least one other manufacturer (Protecta-Sol) uses the unit "cfm/ft. of window width." The unit used to report the results of this investigation is "cfm/sf of window area."

5.1 U-Values

Effective U-values for the various flow rates and return air fractions were calculated by first separating out the value for the heat transfer through the window from the measured total heat transfer through the mask wall and window assembly by the method outlined in Section 2.0, and then applying the data to Equation 3.12.

These effective U-values are plotted as a function of air flow rate per square foot of window area for return air fractions of 0, 0.25, 0.5, 0.75, and 1.0 in Figures 5.1 through 5.6. The return air fraction, FR, is air drawn through the windows and returned to the central HVAC system, where it is recycled to the occupied space. The energy to condition this air must be considered part of the window energy load.

For FR's of zero (i.e. all of the air extracted through the window was exhausted to the outside), the effective U-value decreased from 0.27 Btu/hr-sq ft-°F at no flow to 0.06 Btu/hr-sq ft-°F 2.5 cfm/sf. At an FR of 0.25, the effective U-value continued to decrease with increasing air flows, reaching 0.14 Btu/hr-sq ft-°F at 2.5 cfm/sf. At FR of 0.5 the effective
Figure 5.1 Effective U-Value vs Air Flow Rate/Window Area When Return Air Fraction FR = 0

Figure 5.2 Effective U-Value vs Air Flow Rate/Window Area When Return Air Fraction FR = 0.25
Figure 5.3 Effective U-Value vs Air Flow Rate/Window Area
When Return Air Fraction FR = 0.5

Figure 5.4 Effective U-Value vs Air Flow Rate/Window Area
When Return Air Fraction FR = 0.75
Figure 5.5 Effective U-Value vs Air Flow Rate/Window Area
When Return Air Fraction FR = 1.0

Figure 5.6 Effective U-Value vs Air Flow Rate/Window Area
U-value remained constant through the complete range of flow rates—approximately 0.20 Btu/hr-sq ft-°F. For return air fractions greater than 0.5, the effective U-value began to increase with increasing air flow rates. At FR of 1.0 the average effective U-value, at a flow rate of 3.5 cfm/sf, was 0.37 Btu/hr-sq ft-°F. At no flow through the window cavity, the average effective U-value was 0.27 Btu/hr-sq ft-°F, ranging from 0.26 to 0.28.

Effective U-values are plotted as a function of return air fraction for air-flow rates of 0.75, 1.5, 2.5, and 3.5 cfm/sf in Figures 5.7 through 5.11. In each case, U-values increased with increasing return air fraction in a linear fashion, the rate of increase being greatest at the highest air flow rate. Figure 5.11 shows these linear relationships plotted for each flow rate on the same set of axes. The lines cross at a return air fraction of about 0.5, corresponding to an effective U-value of approximately 0.2 Btu/hr-sq ft-°F.

5.2 Air Stream Temperature Drop

Air stream temperature drop as a function of outside air temperature is plotted for flow rates of 0.75, 1.5, 2.5, and 3.5 cfm/sf in Figure 5.12. Figure 5.13 shows air stream temperature drop as a function of air flow rate for cold side temperatures of -50, -30, -10, and +10°F. This data will allow the designer to estimate the window exit air temperature. Also using this data, the sensible heat lost from the air stream flowing through the window cavity can be easily inferred from the relationship given in Equation 3.8.

5.3 Condensation Potential

Relative humidity limits of room air at 70°F, above which condensation will occur on the inside surface of the triple-pane insulating glass, are plotted as a function of outside air temperature for the various flow rates in Figure 5.14, and in Figure 5.15 as a function of air flow rate for the various cold side temperatures at which the tests were conducted. These limits were arrived at by using the psychrometric chart to determine the relative humidity of air at 70°F that corresponds to a dew point equal to the average temperature of the inside facing surface of the outer insulating
Figure 5.7 Effective U-Value vs Return Air Fraction When Air Flow Rate = 0.75 cfm/sq.ft.

Figure 5.8 Effective U-Value vs Return Air Fraction When Air Flow Rate = 1.5 cfm/sq.ft.
Figure 5.9 Effective U-Value vs Return Air Fraction When Air Flow Rate = 2.5 cfm/sq.ft.

Figure 5.10 Effective U-Value vs Return Air Fraction When Air Flow Rate = 3.5 cfm/sq.ft.
Figure 5.11 Effective U-Value vs Return Air Fraction
Figure 5.12 Air Stream Temperature Drop vs Outside Air Temperature

Figure 5.13 Air Stream Temperature Drop vs Air Flow Rate/Window Area
Figure 5.14 Room Air Relative Humidity Limit to Avoid Condensation on the Inner Surface of the Triple Pane Insulating Glass vs Outside Air Temperature

Figure 5.15 Room Air Relative Humidity Limit to Avoid Condensation on the Inner Surface of the Triple Pane Insulating Glass vs Air Flow Rate/Window Area
glass. A linear interpolation of the raw data for a room side air temperature (i.e. meter box temperature) of 70°F was made for each flow rate so that the results for each flow rate could be compared by plotting this data on the same set of axes. Four thermocouples attached to the glass surface approximately 6 inches in from each corner were used to determine the glass surface temperature. The surface temperature was taken as the average of these four readings. This was in error, because the glass adjacent to the frame was obviously much colder than the glass where the thermocouples were attached. Even at the highest flow rate at which tests were conducted—3.5 cfm/sf—frost formed around the perimeter of the window on the inside surface of the triple pane insulating glass adjacent to the frame at cold side temperatures of -50 and -30°F. Although frost formed adjacent to the frame, the remainder of the glass remained free of frost or condensation at the higher flow rates. The relative humidity levels of the tests ranged from about 19% to 25%.

The relative humidity limits shown in Figures 5.14 and 5.15 are lower than for conventional triple-pane insulating glass. This is because the window cavity air is cooler than room air. At higher flow rates the inside surface temperature of the insulating glass will rise, alleviating for the most part the problem of condensation, but at very cold outside temperatures of -30°F or colder, frost formation adjacent to the frame is still likely to occur. It may be possible to solve this problem by designing a thermal break in the outside sash.

In climates where many consecutive days of temperatures colder than -30°F can be experienced, air flow rates of 3.5 cfm/sf or higher should be used.

5.4 Inside Glass Surface Temperatures

Inside glass surface temperature as a function of outside air temperature is plotted for the various flow rates in Figure 5.16. As in Figures 5.14 and 5.15, the data were normalized to a warm side air temperature of 70°F for each flow rate so that the results for each flow rate could be compared and plotted on the same graph. Even at a cold box temperature of -50°F the inside surface of the inner glass pane remained remarkably warm at 62.1°F at a flow rate of 3.5 cfm/sf. At 0.75 cfm/sf and a
Figure 5.16 Inside Glass Surface Temperature vs Outside Air Temperature

Figure 5.17 Temperature Difference of Room Air and Inside Glass Surface vs Air Flow Rate/Window Area
cold box temperature of -50°F the temperature on the inside surface dropped to only 53.5°F.

The temperature difference of the meter box air and the inside facing glass surface temperature is plotted as a function of air flow rate for cold side temperatures of -50, -30, -10 and 10°F in Figure 5.17. The data from these two graphs can be used by the designer wishing to calculate mean radiant temperatures for a room utilizing air flow windows.

5.5 Pressure Drop Through The Window

Pressure drop through the window at flow rates ranging from 0.69 to 3.82 cfm/sf were measured using a Dwyer Micromanometer. These data are plotted in Figure 5.17.1. The drop proved to be very low. At 3.5 cfm/sf it was 0.037 inches of water.

5.6 Temperature Profile Across the Test Window

Glass surface temperatures and cold and warm side air temperatures for each test conducted at 0, 0.75, 1.5, 2.5, and 3.5 cfm/sf are presented graphically in Figure 5.18 through 5.37 at the end of this section.

5.7 Possible Experimental Error

Situations or conditions that could have contributed to error during the investigation are discussed below.

Step input of heat into the meter box and guard box.

Heat input into the meter box and guard box was accomplished by energizing light bulbs located in both boxes. The light bulbs are on an on-off controller.

In starting and ending each test run it was necessary to pay close attention to whether the power input to the meter box was on or off. The procedure was to start and stop each test immediately after the first count of the watt-hour counter, after the power had cycled off.

Air flow measurement technique.

As described in Section 2.0, air flow was measured using a hot wire anemometer to take air velocity readings at five points across the duct.
Figure 5.17.1  Pressure Drop Through the Window as a Function of Air Flow Rate/Window Area.
cross-section. The area weighted average of these five readings was taken as the discharge velocity and the flow rate was then computed knowing the duct cross-sectional area. The instrument used was a Kurz Model No. 424-4 hot probe anemometer. From the manufacturer's literature, a calibrated instrument will have an accuracy of ±2% with ±0.25% repeatability.

One difficulty with this instrument was its extreme sensitivity. The measurements were taken by inserting the probe into a hole drilled into the side of the duct and bracing the probe against the side of the hole. Any tremor of the hand used to brace the probe would cause the reading to jump 5 to 10 fpm. In spite of these difficulties, flow measurements taken at the beginning and end of the test runs varied less than 1%. The data gathered in this fashion produced very consistent curves.

Using the approximation \[ \Delta h = \frac{C_p}{p} \Delta T. \]

This approximation was used to solve for the sensible heat loss from the air stream passing through the window cavity. The specific heat of air, \( C_p \), was assumed to have the constant value of 0.24 Btu/lbm·°F. If the relative humidity of the air stream could have been accurately measured at both the window inlet and discharge, then the change in enthalpy could have been read directly from the psychrometric chart or tables.

Air leak between meter box and guard box.

It was impossible to achieve a tight seal of the meter box to the mask wall. This allowed air leakage between the meter box and guard box. Because the air temperatures of the boxes were maintained very close to each other, any leak would have created a very slight temperature change.

Thermocouple error.

All thermocouples were calibrated in an ice bath. Using this technique, an accuracy of ±0.2°F can be expected.

Mask wall and window surface temperature measurements.

The temperature drop across the insulated mask wall was taken at two points and the average used in calculating the heat transfer through the mask wall. Readings at the two points were always within 2°F of each other and more frequently within 1°F.

Glass surface temperatures were taken as the average temperature of four points on each accessible glass surface (see points J, K, L and M of Figure 2.5 - Thermocouple Locations). Temperatures measured at these four
points varied more widely than did the temperatures measured at the two points on each surface of the insulated wall. The variation decreased with increasing air flow rates and with warmer cold side temperatures.

Except for the test conducted at no air flow, the coldest temperatures occurred on the right side of the window rather than in the upper half as might be expected, because the air was extracted upward through the cavity. It was hypothesized that the window was unevenly cooled because the cold box air circulation fan was causing a slight impingement of cold air on the right side of the window. This would not have produced an error in calculating heat transfer through the window because the value was derived by subtracting the heat transfer through the insulated wall and the slight amount of heat transfer through the meter box from the total heat input. It was felt the heat transfer through the insulated mask wall was calculated accurately because care was taken to construct the wall so that it was homogeneous. Further, the temperature drops across the wall at the two different locations were very close to each other. The slight uneven cooling of the window may have shifted the relative humidity limit curves downward. It is recommended that the baffle placed in front of the cold box air circulation fan be enlarged to eliminate this problem in future tests that might be conducted using this guarded hot box.

**Frequency of Measurement.**

Measurements were recorded at 10 minute intervals for each 8-hour test. These readings were averaged for each test and standard deviations computed. The deviations were consistently low, indicating the wall and window assembly were in thermal equilibrium.

**Pressure Drop Measurements.**

There was no smooth straight section of duct close to the window at which to measure pressure drop, but the low flow rates eliminated it from being a problem. The data plotted a smooth, consistent curve through the complete range of flow rates.
6.0 ECONOMICS OF AIR FLOW WINDOWS

The capital cost of air-flow windows is significantly higher than conventional double or triple glazed insulating windows—as much as 60% higher. Against the higher first cost of an air-flow window system the following benefits must be considered:

1. Reduced heating and cooling loads that reduce the capital costs of the HVAC system.
2. Reduction of HVAC space requirements.
3. Decrease in annual energy costs for heating, cooling, and electrical lighting.
4. Reduction in peak load conditions and related peak energy demand ratings (Brandle and Boehm, 1981).
5. Increase in thermal comfort and/or reduced costs to achieve thermal comfort because of the increase in mean radiant temperature.

The heating energy saved by air-flow windows is:

$$Q_{SAVED} = \frac{(U-value \ difference) \times (A) \times (DD) \times (24)}{(Boiler \ Efficiency)} \ldots \ldots \ldots \ldots \ldots (6.1)$$

where $Q_{SAVED} = \text{Btu/yr}$

$A = \text{window area in square feet}$

$DD = \text{heating degree F \text{ - days}}$

Estimating the savings from the improved shading coefficient of air-flow windows is much more difficult. The only reliable methods for estimating cooling energy demands require hour-by-hour calculation of the cooling load, utilizing representative weather data.

In addition to the annual heating energy savings many other factors must be considered:

1. The size/capacity of the central heating plant may be reduced (usually between 10%-30%) as well as the size/capacity of the air handling equipment (usually between 10%-25%) (Rapatti, 1984).
2. Because the mean radiant temperature is increased, people can be
positioned closer to the windows without experiencing discomfort, resulting in more available useful space. Because the mean radiant temperature is increased, the ambient air temperature may be lowered. (See Section 4.0)

3. Because air-flow windows have a warm inside surface, the under-window perimeter heating units, which serve the purpose of offsetting the down draft off of the cold surface of a conventional window, can be eliminated. The central heating plant is capable of meeting all heating requirements (Rapatti, 1984).

4. If heating and cooling loads occur simultaneously, heat recovery may be achieved by "load shifting" from one part of the building to another. For example, under favorable solar radiation conditions air-flow windows can serve as solar collectors, warming the air in the window cavity and transporting this energy for heating in another zone of the building, with or without intermediate energy storage (Brandle and Boehm, 1981).

Once the above considerations have been made in terms of capital costs for a proposed facility, an economic analysis can be performed to compare investment in air-flow windows versus a conventional fenestration system. There are several approaches to this, using principles of engineering economics, including the present worth of annual savings in energy costs, and the capital recovery factor (CRF) method to examine total annual costs of the facility.

The equation for predicting the present worth of annual savings is:

\[ P = \frac{A \cdot (1 + i)^N - 1}{i \cdot (1 + i)^N} \]

\[ \text{where}\ P = \text{present worth} \]
\[ A = \text{annual savings} \]
\[ i = \text{fuel escalation rate} \]
\[ N = \text{useful life of the facility} \]

The capital recovery approach, on the other hand, examines total annual costs of the facility under each option. The equation is:
CRF = \frac{i}{1 - (1 + i)^{-n}}

where \( i \) = market interest rate (or required rate of return)

\( n \) = useful life of the facility

The CRF is multiplied by the total cost ($/sq. ft.) of construction to arrive at the annualized cost. The annual energy cost is calculated by multiplying the present cost of energy ($/Btu) times the annual heat loss for the facility (Btu/sq. ft.-year) divided by the heating system seasonal efficiency. The annualized capital cost and annual energy cost are added together to arrive at the total annual cost ($/sq. ft.).

Economically, the most appropriate use for return-air windows appears to be in large office buildings having high outside air requirements and large window-to-opaque-wall ratios, located in extreme climates.
7.0 SUMMARY AND CONCLUSIONS

In evaluating and comparing the energy performance of a return-air window system to a conventional fenestration system it is necessary to consider the fate of the air after it is extracted through the window cavity. This was accomplished by defining the Return Air Fraction, FR, which is the air that flows through the windows and returned to the central HVAC system to be conditioned and recycled to the occupied space. The energy required to condition this air must be considered as part of the window energy load.

At an FR of zero (i.e. all of the air extracted through the window is exhausted to the outside) the return-air window achieved an effective U-value of 0.06 Btu/hr-sq ft-°F at an air flow rate of 2.5 cfm/sf. This corresponds to an effective R-value of 16.7, substantiating Ekono Windo Company's claim.

On the other hand, at an FR of 1.0 (i.e. all of the air flow through the windows is recirculated to the space), the window achieved effective U-values of no better than 0.34 at 2.5 cfm/sf and 0.37 at 3.5 cfm/sf. These correspond to R-values of 3 and 2.7, respectively, about the same as a standard triple-pane window.

The mean radiant temperature of rooms with air-flow windows will be improved, as the inside glass surface temperature will be much closer to room air temperature in comparison to a conventional double or triple-glazed window. This is especially significant in rooms with large window-to- opaque-wall ratios.

Relative humidity limits to avoid condensation on the inside facing surface of the outer insulating pane are slightly lower than for standard triple-pane insulating windows because the cavity air is cooler than the room air. To avoid condensation in climates where many consecutive days of temperatures of -30°F or colder can be experienced, air flow rates of 3.5 cfm/sf or higher should be applied. To prevent frost or condensation from forming around the perimeter of the glass next to the frame, the three panes of outer insulating glass should be mounted in a casing with a thermal break.

Pressure drop through the window proved to be very slight. At 3.5 cfm/sf the pressure drop was 0.037 inches of water.

The capital cost of air-flow windows is estimated to be 40% to 60%
higher than that of conventional double or triple-glazed insulating windows (Brandle and Boehm, 1981). The requirement for two separate sash assemblies at each window, custom engineering and manufacturing, and the cost of duct connections to the return air system all contribute to the higher cost (Rapatti, 1984).

Against the higher capital cost, the following benefits must be considered:

1. Reduced heating and cooling loads of the building, thus reducing the capital cost of the HVAC equipment and distribution system.
2. Reduction of space because of a smaller HVAC system.
3. Decrease in yearly energy consumption for heating, cooling, and electrical lighting.
4. Reduction in peak load conditions and related peak energy demand ratings.
5. Increase in thermal comfort and/or reduced costs to achieve thermal comfort because of the increase in mean radiant temperature.
6. Tax credits that may be available to investors in energy-saving devices.

Although air-flow windows have been installed in a variety of building types, the best candidates appear to be large office buildings with high ventilation requirements, large window to opaque wall ratios and located in areas with a large fluctuation in seasonal climate. In fact, their major application has been in such buildings. Hospitals are another good candidate because of the high ventilation requirement and because of the improved comfort of patient rooms resulting from the increase in mean radiant temperature.
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