ENERGY SAVING THROUGH THE USE OF A SUPPLY AIR WINDOW

Pertti Sarkomaa[1], Petri Kuisma[2] and Kai Ryynänen[2]

[1]*Department of Energy and Environmental Technology, Lappeenranta University of Technology, Lappeenranta, P.O. Box 20, FIN-53851, FINLAND*
[2]*Rovaniemi Polytechnic, School of Technology, Rovaniemi, FIN-96300, FINLAND*

ABSTRACT

In Arctic and sub-Arctic climates, such as those in Scandinavia, multiple-glazing windows that consist of at least three panes are widely used. Typically, the replacement air for the extracted air, especially in low-cost accommodation with forced extraction, enters the interior space in the form of leakage flow through the window jambs and the walls or through the supply air vents. The temperature for the air entering the room is close to that of the outdoor air, which may cause a sensation of draft. Moreover, the whole supply airflow must be heated up to the indoor temperature by the heating system.

In a supply air window, air enters from the outside into the window cavity through holes or vents at the lower edge of the window frame. The air flows upwards in the cavity, while being heated by convection from the surrounding panes. The heated air is then drawn into the room through a vent or an adjustable air valve in the upper edge of the window frame. When compared to the conventional case, the following results can be identified: 1) the supply airflow is pre-heated, which reduces the feeling of draft; 2) less heat flow is required to heat up the supply air, which reduces the energy consumption of the room. Furthermore, the size of the radiators can be reduced. All of these factors contribute to a better indoor climate and savings in the required investments and in energy as well.

In this paper, the window is considered to be a heat exchanger in which heat is transferred by radiation and convection between the panes and the supply airflow. Denoting the indoor and outdoor temperature with T_i and T_o, correspondingly, equations are presented for the determination of the recuperation ratio, ε, of the windows. The recuperation ratio can be used for calculating the temperature of the supply air entering the room, $T_{sa} = T_o + \varepsilon (T_i - T_o)$. The results are compared to measured values for recuperation ratio. The equation for the overall heat transfer coefficient, U, is also presented for the supply air window.

KEYWORDS

Supply air, window, heat recovery

INTRODUCTION

In the Arctic and sub-Arctic regions of the World, the walls, roof and floors of new buildings are very well thermally insulated, and their overall heat transfer coefficient, U, is in the order of 0,15 – 0,3 W/Km^2. The largest part of the consumption of heat energy by a building in a cold climate is in the form of heat loss through windows and the heating of supply air for the ventilation system. The inlet air of residential building is usually 10 – 25 g/s per room and in cold weather ($T \leq -10 ... -15^\circ C$), it is half of this. Thus, the variation of the inlet air is usually between 5-25 g/s per normal room of a residential building.

It is possible to avoid some of the disadvantages of the cold supply air for ventilation by using supply air windows. The airflow enters the outer cavity of the windows, where the distance between the panes is about 80 – 100 mm, through holes or vents of the lower edge of the window frame. The air flows upwards in the cavity and is heated by convection from the walls of the cavity. The heated air then flows into the room through holes, a vent or an adjustable air valve in the upper edge of the window frame.

RECUPERATION RATIO AND THE OVERALL HEAT TRANSFER COEFFICIENT OF A SUPPLY AIR WINDOW

A supply air window is like a heat recovery heat exchanger. The recuperation ratio of an air supply window is defined by Eqn. 1.

$$\varepsilon = \frac{T_{sa} - T_o}{T_i - T_o} \tag{1}$$

The recuperation ratio, ε, is also the proportional measure of the energy saving of the supply air for ventilation. The Reynolds number of the flow in the cavity is defined by the following equation:

$$\mathrm{Re}_t = \frac{2wt}{\nu} \tag{2}$$

The variation of the velocity of the measurements was w = 0,1 m/s … 0,2 m/s and Re_t = 1200 … 2800, which means that the outflow was laminar or turbulent. The critical velocity is about w = 0,16 … 0,19 m/s. The measured surface temperature of both glass panes around the airflow was rather constant. Also, because the temperatures outside the window are constant, the recuperation ratio of the window can be calculated from Eqn. 3.

$$\varepsilon = \frac{T_{sa} - T_o}{T_i - T_o} = \frac{\overline{T} - T_o}{T_i - T_o}\left(1 - e^{-z}\right) \tag{3}$$

The ratio of the mean overall heat transfer coefficient between the air flow and the outdoor temperature, U_o, and the mean overall heat transfer coefficient between the indoor air and the flow, U_i, is defined by Eqn. 4.

$$x = \frac{U_o}{U_i} \tag{4}$$

In this research, the variations of the overall heat transfer coefficients ranged between the following values: U_o = 4 …6 W/m^{2}K and U_i = 2,5 … 3,5 W/m^{2}K. The mean temperature, $\overline{T}$, in Eqn. 3 using ratio x is the following:

$$\overline{T} = \frac{T_i + xT_o}{1 + x} \tag{5}$$

The corresponding number of heat transfer units, z, is

$$z = (1 + x)\frac{G_i}{q_m c_p}, \tag{6}$$

where q_m is the mass flow through the window, c_p the specific heat of the air at constant pressure and G_i the conductance [W/K] from the indoor air to the air flow. Thus,

$$\varepsilon = \frac{T_{sa} - T_o}{T_i - T_o} \approx \frac{1}{1+x}\left(1 - e^{-\frac{G_i}{q_m c_p}(1+x)}\right) \tag{7}$$

When the change in the mass flow of the air is small, ratio x remains approximately constant and the ratio of the recuperation ratios can be calculated from Eqn. 8.

$$\frac{\varepsilon_2}{\varepsilon_1} \approx \frac{1 - e^{\frac{q_{m1}}{q_{m2}}\ln(1-(1+x)\varepsilon_1)}}{1 - e^{\ln(1-(1+x)\varepsilon_1)}} \tag{8}$$

The mean overall heat transfer coefficient of the glass surfaces of the cavity to the outdoor temperature is U_{o0}. The mean overall heat transfer coefficient of the indoor air to the glass surfaces of the cavity is U_{i0}. The mean convective heat transfer coefficient from the glass to the airflow is h and the mean heat transfer coefficient of radiation between the glass panes of the cavity is h_{s12}. Using these notations, we get mean temperature $\overline{T}$ and the number of heat transfer units, z, from Eqns. 9 and 10.

$$\overline{T} = \frac{(U_{i0} + h + 2h_{s12})U_{o0}T_o + (U_{o0} + h + 2h_{s12})U_{i0}T_i}{(U_{i0} + h + h_{s12})(U_{o0} + h + h_{s12}) - h_{s12}^2 - [0{,}5(U_{i0} + U_{o0}) + h + 2h_{s12}]h} \tag{9}$$

$$z = \frac{2Ah}{q_m c_p}\left[1 - \frac{[0{,}5(U_{i0} + U_{o0}) + h + 2h_{s12}]h}{(U_{i0} + h + h_{s12})(U_{o0} + h + h_{s12}) - h_{s12}^2}\right] = \frac{H}{L_0} \tag{10}$$

A is the area and H the height of the glass panels. Usually, the frame of the window is made of timber and the heat flow through the frame is very small. If we assume that the frame is a perfect insulator, then the mean radiation heat transfer coefficient between the glass panes of the cavity is calculated from Eqn. 11.

$$h_{s12} \approx \frac{\sigma\left(T_1^4 - T_2^4\right)}{2\left(\frac{1}{\varepsilon} - 1 + \frac{1}{1 + F_{12}}\right)} \tag{11}$$

σ is the Stefan-Bolzmann constant. ε is the emissivity of the glass. T_1 and T_2 are the mean temperatures of the glass surfaces and F_{12} is the radiation shape factor between the glass panes. If the glass panes are large, then $F_{12} \approx 1$ and

$$h_{s12} \approx \frac{\sigma\left(T_1^4 - T_2^4\right)}{\frac{2}{\varepsilon} - 1} \tag{12}$$

The mean surface temperatures of the panes are the following:

$$\begin{aligned} T_1 = {} & \frac{\left(U_{o0} + h + h_{s12}\right)U_{i0}T_i + h_{s12}U_{o0}T_o}{\left(U_{i0} + h + h_{s12}\right)\left(U_{o0} + h + h_{s12}\right) - h_{s12}^2} + \\ & + \frac{\left(U_{o0} + h + 2h_{s12}\right)h}{\left(U_{i0} + h + h_{s12}\right)\left(U_{o0} + h + h_{s12}\right) - h_{s12}^2}\left[\overline{T} + \frac{\left(\overline{T} - T_o\right)L_0}{H}\left(e^{-H/L_0} - 1\right)\right] \end{aligned} \tag{13}$$

$$\begin{aligned} T_2 = {} & \frac{\left(U_{i0} + h + h_{s12}\right)U_{o0}T_o + h_{s12}U_{i0}T_i}{\left(U_{i0} + h + h_{s12}\right)\left(U_{o0} + h + h_{s12}\right) - h_{s12}^2} + \\ & + \frac{\left(U_{i0} + h + 2h_{s12}\right)h}{\left(U_{i0} + h + h_{s12}\right)\left(U_{o0} + h + h_{s12}\right) - h_{s12}^2}\left[\overline{T} + \frac{\left(\overline{T} - T_o\right)L_0}{H}\left(e^{-H/L_0} - 1\right)\right], \end{aligned} \tag{14}$$

where

$$L_0 = \frac{q_m c_p}{2Bh}\left[1 - \frac{\left[0{,}5(U_{i0} + U_{o0}) + h + 2h_{s12}\right]h}{\left(U_{i0} + h + h_{s12}\right)\left(U_{o0} + h + h_{s12}\right) - h_{s12}^2}\right]^{-1} \tag{15}$$

The Nusselt number of the convective heat transfer coefficient, h, from the glass panes to the airflow is usually presented proportional to the power of 0,8 of the Reynolds number. The change in the known heat transfer coefficient, h_{ref}, can be predicted using Eqn. 16.

$$\frac{h}{h_{ref}} = \left(\frac{q_m}{q_{m,ref}}\right)^{0{,}8}\left(\frac{T_{ref}}{T}\right)^{-0{,}208} \tag{16}$$

The overall heat transfer coefficient of the glass panes of the whole window is U.

$$U = \frac{T_2 - T_o}{T_i - T_o}\left[\frac{s}{k} + h_o\right]^{-1} \tag{17}$$

MEASUREMENTS

The cold outdoor air was simulated using a refrigerated chamber. The window was installed on the wall of the chamber at a height of 1100 mm from the floor of laboratory. The velocity of the air in the chamber and in the laboratory was about 0,2 m/s. The window was a normal one, except for the fact that eleven holes (Φ = 10 mm) were drilled in the lower edge at uniform distances from each other. The same number of holes were also drilled in the upper edge of the window. A thermally insulated collection chamber surrounded the holes in the upper edge, and an insulated connecting channel joined the chamber with the mass flow measurement device, which was followed by a fan. The glass pane of the window was 1225 mm x 820 mm, and the thickness of the panes 4 mm, 4 mm and 4 mm. The gaps between the window panes were, from the inside to the outside, 12 mm and 85 mm. The flow area of the window was 915 mm x 85 mm.

The conditions of the measurements were:

- the temperature of the cold chamber varied from 0 ^{o}C to - 20 ^{o}C
- the temperature of the laboratory was about + 22 ^{o}C
- the mass rate of the airflow through the window was about 10 and 20 g/s.

At the beginning, the temperature of the chamber and the mass flow through the window were regulated to the appropriate level. Some time after the transient period (all together 4 h), when condition of the system was a steady state, we made the following measurements:

- the temperature of the cold camber, T_o
- the temperature of the laboratory, T_i
- the temperature of supply air after the window, T_{sa}
- the temperature of the air before the flange of the mass measurement
- the atmospheric pressure in the laboratory
- the pressure difference between the laboratory and the flow before the flange of the mass measurement

RESULTS OF THE MEASUREMENTS

The measured data and calculated recuperation ratio of the test runs are presented in Table 1.

TABLE 1
The measured and calculated data of the test runs (TR).

	TR 1	TR 2	TR 3	TR 4	TR 5	TR 6	TR 7
T_i /K	295	295	295	293	293	293	294
T_o /K	273	273	253	253	273	273	273
T_{sp} / K	280	278	267	262	279	277	268
q_m / g/s	9,02	18,24	10	20	8,44	20,8	28,9
ε	0,305	0,240	0,338	0,233	0,310	0,230	0,245

From the measurements in Table 1, we can calculate the best value of x = 1,9 for the ratio of Eqn. 8.

The agreement of the calculated ε with one measured value of ε and Eqn. 8, where x = 1,9, is very good. The difference between the calculated and measured data is of the same order as the variation in the measured data.

DISCUSSION

The variation of the recuperation ratio of the supply air window ranged between 0,23 and 0,34. The larger values are connected with a small intake airflow.

It is possible to develop a simple function for the recuperation ratio of the supply air window using the equations developed here and a limited number of measurements.

LIST OF SYMBOLS

g	gravity, 9,81 m/s^2
H	height of the glass plane of the window, m
h	heat transfer coefficient, W/(m^{2}K)
k	conductivity, W/(mK)
Pr	Prandtl number
q_m	mass flow, kg/s
Ra	Rayleigh number
Re	Reynolds number
s	thickness of the pane, m
T	temperature, K
$\overline{T}$	mean temperature defined by Eqns. 5 and 9
t	distance between the glass panels of airflow, m
U	overall heat transfer coefficient, W/(m^{2}K)
U_i	mean overall heat transfer coefficient between the indoor air and the flow, W/(m^{2}K)
U_o	mean overall heat transfer coefficient between the flow and the outdoor, W/(m^{2}K)
w	velocity, m/s
z	number of heat transfer units, NTU, defined by Eqns. 6 and 10
x	ratio U_o / U_i
ε	recuperation ratio
ν	kinematic viscosity, m^2/s

Subscripts

0	quantity between surface of the cavity and indoor or outdoor
1, 2	surface of the gap
i	indoor
o	outdoor
ref	reference value
s	heat radiation
sa	supply air
t	based on distance between panes