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AIC 1703

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DESIGN OF LOW COST VENTILATION AIR HEAT EXCHANGERS

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PROCEEDINGS, HEAT TRANSFER IN BUILDINGS CONFERENCE, BELGRADE, YUGOSLAVIA, SEPT, 1977

SPONSORED BY INTERNATIONAL CENTRE FOR HEAT AND MASS TRANSFER

ABSTRACT

The design of a low cost ventilation air heat exchanger ~~proposed~~ which utilizes plastic sheets as the heat transfer surface is presented. Laboratory tests on such a counter flow heat exchanger have demonstrated very high values of overall heat transfer coefficients, heat exchanger effectiveness, and temperature recovery when the flow is laminar and buoyancy effects assist the heat transfer. The costs of such heat exchangers would make them attractive for many building applications.

NOMENCLATURE

A	heat exchanger area
C_p	specific heat at constant pressure
C	cost
D	distance between heat exchanger plates
g	gravitational acceleration
Gr_D	Grashof number
	$= g\rho^2\beta\Delta T\ D^3/\mu^2$
h_{ig}	heat of sublimation or vaporization for water
K	present value factor
	$= \dfrac{\left(\frac{1+x}{1+r}\right)^N - 1}{\left(\frac{x-r}{1+r}\right)}$
m	mass flow rate
N	number of years

$NTU = \frac{UA}{\dot{m}_c C_{Pc}}$ number of transfer units

q heat rate

Q total heat required for ventilation air per year

r discount rate for money

T temperature

U overall heat transfer coefficient

V average velocity

x annual inflation increase in fuel prices

β expansion coefficient

ΔT change in temperature from mean surface to mean bulk

ϵ heat exchanger effectiveness

η_f fuel burner efficiency

μ viscosity

ρ density

ω humidity ratio

Subscripts

c cold air

f fuel

h hot air

HE heat exchanger

TE thermal energy

VA ventilation air

1 warm air inlet

2 warm air exhaust

3 cold air exhaust

4 cold air inlet

INTRODUCTION

In the past, heat recovery from ventilation air exhaust has not usually been considered for small buildings due to its large capital costs relative to the small savings incurred from reduced heating loads. Now with more expensive fuel costs, heat recovery from ventilation air may be one of the most cost effective ways to cut fuel consumption. This paper describes the design, cost analysis and testing of a ventilation air, counter flow, heat exchanger which would be suitable for use in houses and other small buildings.

The efficacious application of a heat recovery system for ventilation air demands that a building be "tight" so that uncontrolled air flow leakage is reduced to a small fraction of the ventilation air flow rate. This may be done by incorporating an effective vapor and air flow plastic seal or barrier on all the inside wall, ceiling, and floor surfaces and by eliminating such things as open chimneys and cracks in doors and windows. While new buildings can and should be built to have such features at a modest extra cost, older buildings might require extensive retrofitting to be suitable for ventilation air heat recovery. In addition, heating systems that are directly compatible with ventilation air heat recovery are restricted to electrical heating, central steam or hot water heating, and solar heating. By isolating the combustion supply and chimney for fuel burning in buildings, one is able to make efficient use of a ventilation air heat exchanger.

HEAT EXCHANGER DESIGN

For the purpose of laboratory testing, a ventilation air heat exchanger has been designed and constructed as shown in Figure 1. The heat exchanger's 11.36 m^2 of heat transfer surface was constructed of eleven 0.15 mm thick polyethylene plastic sheets such that each 50x203 cm sheet was placed 1.27 cm apart. The heat exchanger was designed to operate in the vertical position such that cold air entered at the bottom and flowed 2 m up through alternate sheets and the warm air which entered at the top and flowed down. The exterior frame of this counter flow heat exchanger consisted of 1.27x5 cm plywood stripes covered with 5 cm of polystyrene insulating board which gave an effective thermal resistance in excess of 1°C m^2 W^{-1} on all sides of the heat exchanger.

HEAT EXCHANGER TESTS AND RESULTS

Tests were done on the heat exchanger over a range of flow rates from 0 to 3 m^3 per minute and the temperature recovery fraction, heat exchanger effectiveness and the overall heat transfer coefficient were calculated using the following equations for the heat transfer rate:

$$q = m_c C_{Pc}(T_3-T_4) \tag{1}$$

$$= m_h[C_{Ph}(T_1-T_2) + (\omega_1-\omega_2)h_{ig}] \tag{2}$$

$$= \epsilon\, m_c C_{Pc}(T_1-T_4) \tag{3}$$

$$= UA\,\overline{\Delta T} \tag{4}$$

where

$$\overline{T} = \text{LMTD} = \frac{(T_1-T_3) - (T_2-T_4)}{\ln\left(\frac{T_1-T_3}{T_2-T_4}\right)}$$

For thermal energy recovery in ventilation air the most important thermal parameter is the temperature recovery factor, $(T_3-T_4)/(T_1-T_4)$, or heat exchanger effectiveness, ϵ. The results from tests presented in Figure 2 suggest that very high values of heat exchanger effectiveness exist at low flow rates. In addition, the heat exchanger effectiveness as a function of number of transfer units, NTU, may be compared to the theoretical results for the case $m_c C_{Pc}/m_h C_{Ph}$ = 1 as shown in Figure 3. That is,

$$\epsilon = \frac{NTU}{1 + NTU} \tag{5}$$

when $m_c C_{Pc}/m_h C_{Ph}$ = 1 for a counterflow heat exchanger where NTU is defined by

$$NTU = \frac{UA}{m_c C_{Pc}} \tag{6}$$

The overall heat transfer coefficient, U, is presented as a dimensionless Nusselt number Nu = 2 UD/k as a function of Gr/Re^2 in Figure 4 where Gr/Re^2 has been shown to be the appropriate dimensionless parameter in forced/natural convection problems. It can be seen from these results that the Nusselt number increases substantially beyond that for a fully developed laminar flow in long ducts with constant wall heat flux. An approximate correlation of the experimental results is given by the equation

$$Nu = 3.6 + 3.4x10^4 \ (Gr/Re^2)^2 \tag{7}$$

It would appear that buoyancy effects, as given by the Grashof number, Gr, increase as forced convection effects, as given by the Reynolds number, Re, decrease. Attempts to operate the heat exchanger against the gravitation forces which indicated heat exchanger effectiveness of much less than .5 would add further evidence to this theory. Flows against gravity not only produced low heat exchanger effectiveness values, but they are inherently unstable and the results are not necessarily reproduceable with slightly different experiments. Hence, no further investigations were made of the inverted mode of operation.

In cold climates heat exchangers may condense water and freeze up if warm moist inside air is cooled down below the local dew point or freezing point of water. When freezing occurs, the layer of frost forms on the inside air flow channels which restricts the flow passages and reduces the heat transfer rate. Long term tests showing the reduction in temperature recovery and flow rate as a function of time are presented in Figures 5 and 6. Again, if the flow rate is small, the efficiency is seen to maintain a high value prior to gradually dropping to very low values over a period of days. Freeze-up of the heat exchanger causes the deterioration in performance, but periodic defrosting of the heat exchanger rapidly restores the heat exchanger ot its original performance.

COST CONSIDERATIONS

The total present value cost of introducing a ventilation air heat exchanger which is financed over a number of years of operation, N, consists of the initial cost of installing the heat exchanger, C_I, plus the cost of thermal energy over the same number of years required to preheat the required ventilation air up to room temperature, C_A. That is

$$C_T = C_I + C_A \tag{8}$$

where the heat exchanger cost may be assumed to vary directly with its heat transfer area A such that

$$C_I = C_{I1} + C_{I2} A$$

and (9)

$$C_A = K \frac{C_f}{\eta_f} (1-\varepsilon) \int_{year} m_c C_{Pc} (T_3 - T_4) \, dt$$

or

$$C_{TE} = \frac{K \, C_f \, Q}{\eta_f} (1-\varepsilon)$$

It can be seen from (5) that

$$1-\epsilon = \frac{1}{1 + NTU} \tag{11}$$

For a constant rate of air flow required for ventilation NTU will vary directly with the UA product in (6) where

$$U = \frac{Nu\ K}{2D}$$

The Nusselt number was shown to vary in Figure 4 with the parameter Gr/Re^2 in a quadratic manner as given by (7). The parameter Gr/Re^2 is given by $Dg\beta\Delta T/V^2$ where both the temperature difference ΔT and the flow velocity tend to vary inversely with the heat exchanger area, A, over finite changes in 1/A and for a specified flow rate. Combining these equations for NTU in (11) finally gives approximately

$$1-\epsilon = \frac{1}{1 + .20\ A + 1.0\ A^3} \tag{12}$$

where A is in m^2. The condition for minimum cost is given by

$$\frac{d\ C_T}{dA} = 0 \tag{13}$$

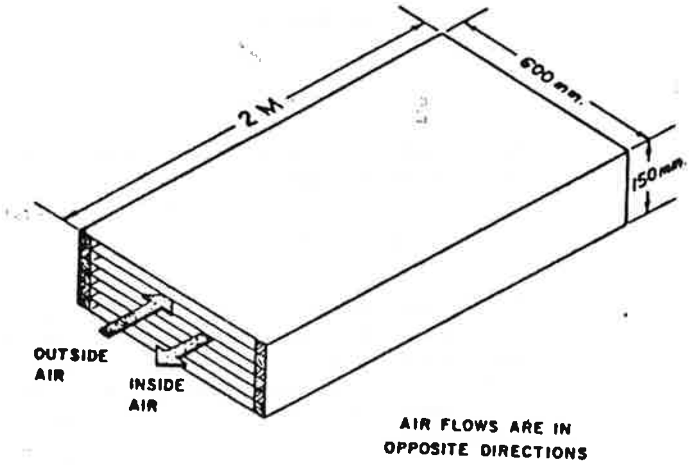
or

$$\frac{C_{I2}}{K\ C_f\ Q} \left[1 + .20\ A + 1.0\ A^3\right] = .20 + 3.0\ A^2 \tag{14}$$

which gives an optimum heat exchanger area of approximately 50 m^2 for a typical house. The corresponding effectiveness as given by equation (12) is nearly 100% under normal flow conditions. Dust, condensation, frosting, and variable flow would likely given lower values. Furthermore, natural convection will cause a small flow rate under most conditions so that Nu and ϵ will be limited. Different load cost and climatic conditions would yield other sizes.

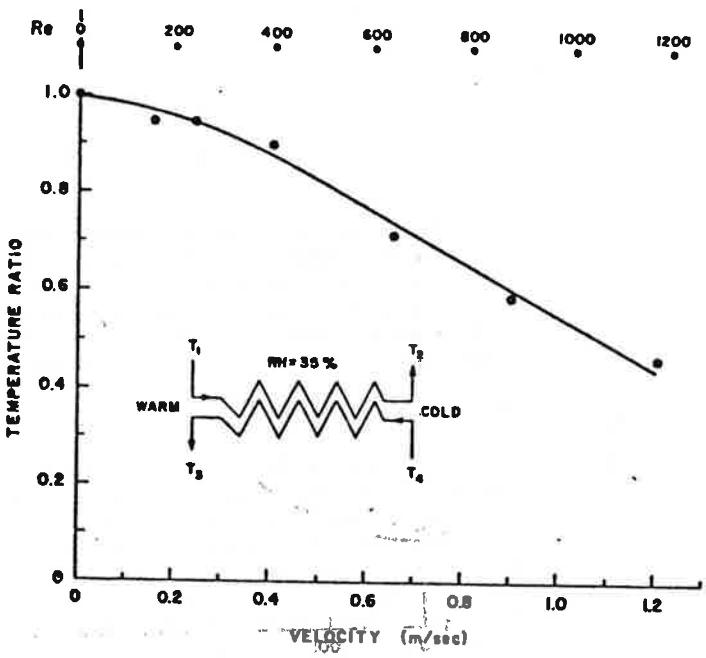
DISCUSSION AND CONCLUSIONS

The design of a low cost ventilation air heat exchanger has been presented. Such a heat exchanger could be used to recover most of the thermal energy from ventilation air which must be brought into buildings by fans. The experimental results indicate that such a heat exchanger should be operated so that gravitational forces assist the flow and hence the heat transfer when the flow rates are low and the flow is laminar. The method of optimizing costs over a specified period of time presented indicates that the use of low cost materials such as plastics permits the designer to use very high heat exchanger effectiveness values. In very cold climatic conditions where frosting occurs periodic defrosting must be provided for in the design.



AIR-TO-AIR HEAT EXCHANGER SCHEMATIC

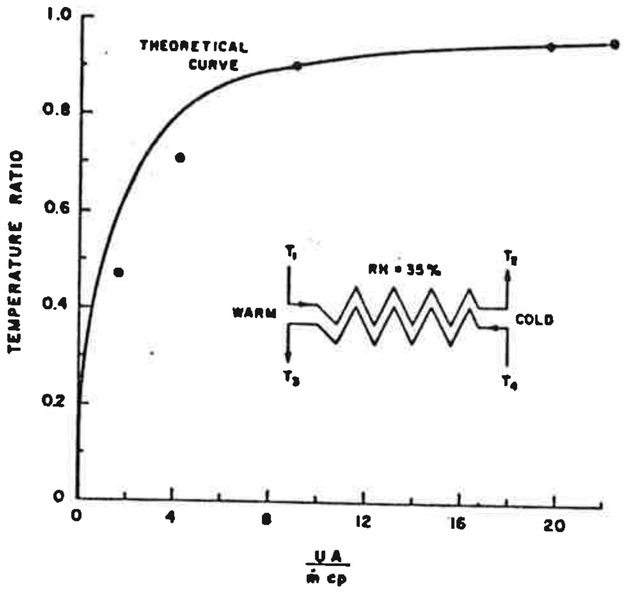
Figure 1



AIR TO AIR HEAT EXCHANGER PERFORMANCE

TEMPERATURE RATIO $\frac{T_3 - T_4}{T_1 - T_4}$ VS AIR VELOCITY

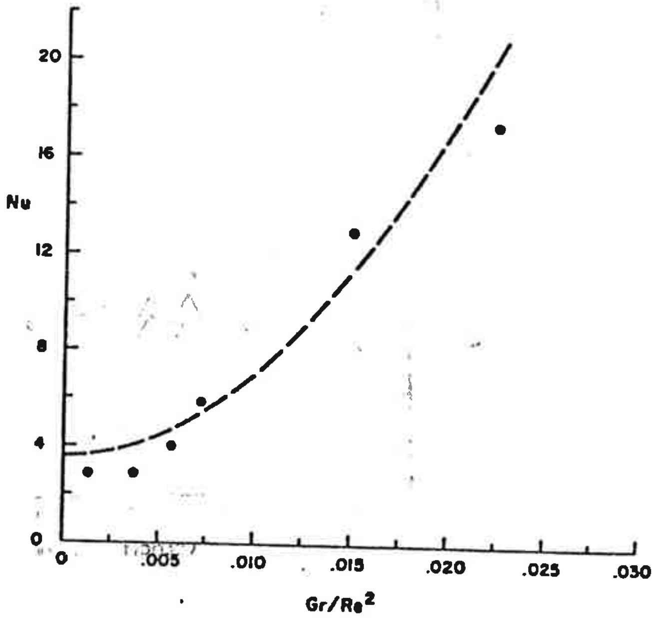
Figure 2



AIR TO AIR HEAT EXCHANGER PERFORMANCE

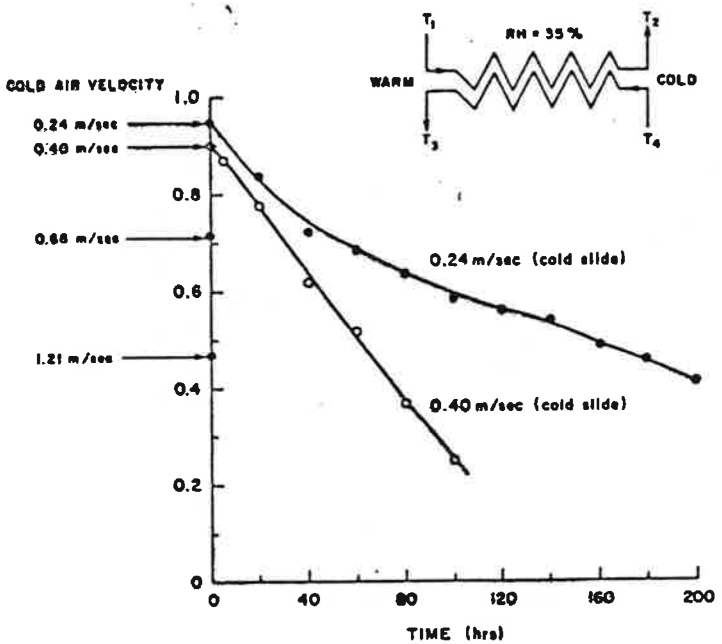
TEMPERATURE RATIO $\frac{T_3 - T_4}{T_1 - T_4}$ VS $\frac{UA}{\dot{m}\ cp}$

Figure 3



PLOT OF NUSSELT NUMBER VERSUS GRASHOF/(REYNOLDS)^2

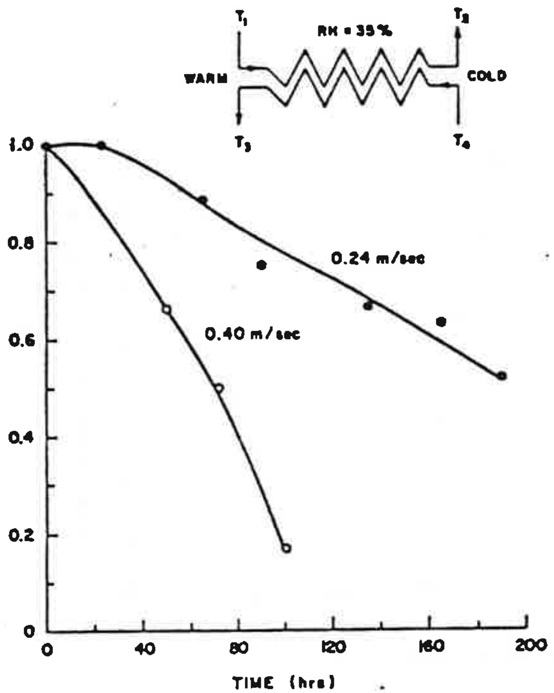
Figure 4



AIR TO AIR HEAT EXCHANGER PERFORMANCE

TEMPERATURE RATIO $\frac{T_3 - T_4}{T_1 - T_4}$ VS TIME (hrs)

Figure 5



AIR TO AIR HEAT EXCHANGER PERFORMANCE

FLOW EFFICIENCY Q_1/Q_2 $Q_{hot\ side}/Q_{cold\ side}$

Figure 6