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An Experimental and Theoretical Approach to Heat Recovery in Air Conditioning Systems

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Abstract: In many buildings, heat recovery from ventilation air is the most important single means of energy conservation. The same level of indoor air quality is achieved with lower energy consumption if the heat recovery units are properly designed. (Seppanen, 2000). In this study, using a plate heat exchanger in an air handling unit, the relation between the plate echanger effectiveness and air flow rate has be en found. Values are obtained both theoretically and experimentally and compared by graph. Fresh air requirement of a room and the difference between indoor - outdoor temperature are known, saved energy from a unit exchanger surface area can be obtained from this generalized graph.

Key Words: Heat Recovery; Plate Heat Exchanger; Air Conditioning

Introduction

The cost of generating thermal energy has continuously increased in the last decades, so the methods for recovery of vaste energy have gained more importance. The plate exchangers supply cost savings by recovery heat when they are used in an air conditioning system. The first aim of an energy recovery system is reduction of eneryg consumption and costs in building process by transferring energ from outlet to inlet air flow.

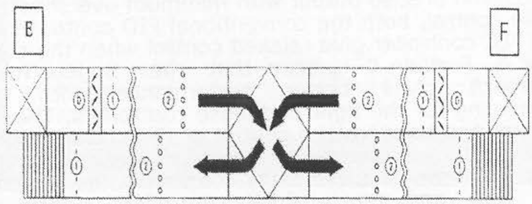

Fig. 1: Airhandling Unit with A Plate Exchanger System

E Exhaust air; F Fresh air; D Dampers; I Flowmeters; 2 Thermocouples

The air handling unit with a plate exchanger system which is used experimentally is shown basically in Fig. 1 and 2.

This study has been done in winter climatic conditions heating time and when the supply air flow rate is equal to the amount of exhaust air flow rate. Three different flow rates ($2000 m^3/h$, $3100 m^3/h$ and $4000 m^3/h$) have been used in experimental study. The heat exchahger has $46 m^2$ heat transfer surface area and plates are made of 0,3 mm aluminum sheet Inlet and outlet temperatures of supply and exhaust air the air flow rates and the pressure drops for each current have been measured. Before tests, both inlet and outlet air flow rates were measured and checked for air leak. After waiting for 20 miniutes for each input, the values were read.

In analytical study, the overall heat transfer coefficient of exchanger has been calculated. The effectiveness has

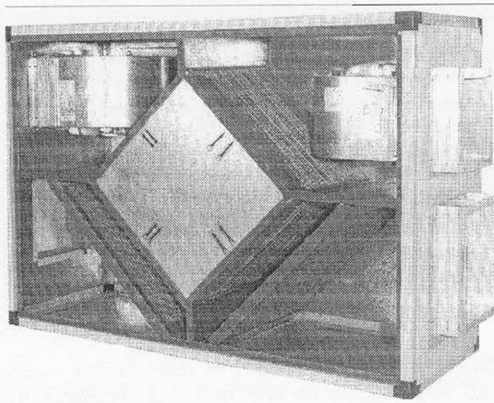

Fig. 2: Plate Heat Exchanger

been obtained by using NTU method (Incropera and De Witt, 1996) and compared with experimental results.

Experimental Set up and Experimental Results: For measurements air handling unit in Fig.1 has been used. Heat recovery system has been constructed and installed as shown also in Fig.1. The location of flow meters (Hesco, Pilgestec AG type), thermocouples (type K) and differential manometers(Magnehelic) are shown.Outdoor air flow rate was adjusted with dampers and measured with flowmeters. When the desired value was obtained, damper was fixed. Outdoor air dry bulb temperature was obtained by using 16 thermocouples.

Experimental results are summerized in Table 1. Flow rates versus heat exchanger effectiveness have been obtained by the help of the values in Table 1 and shown graphically in Fig.3 (Güngören, 1999).

Analytical Calculation Method: The heat transferred in the heat recovery system from exhaust gases is the saved energy. To find this energy, the heat balance between the exhaust air and fresh air:

$$Q = m_1 c_{p1} (T_{1,i} - T_{1,o}) = m_2 c_{p2} (T_{2,o} - T_{2,i}) \quad (1)$$

Also,

$$Q = K.A.\Delta T_m = \varepsilon C_{min} (T_{1,i} - T_{2,i}) \quad (2)$$

Table 1: Experimental Results: V_1: Volumetric Flow Rate of Exhaust Air (m^3/h); V_2: Volumetric Flow Rate of Outdoor Air (m^3/h); $T_{1,i}$: Inlet Temperature of Exhaust Air to the Plate Exchanger (^0c); $T_{1,0}$: Outlet Temperature of Exhaust Air from the Plate Exchanger (^0c); $T_{2,i}$: Temperature of Outdoor Air Entering the Heat Exchanger (^0c); $T_{2,0}$: Temperature of Outdoor Air Leaving the Heat. Exchanger(^0c) ; ε: Heat Exchanger Effectiveness (%); Δp: pressure Drop (Pa)

$V_1(m^3/h)$	$V_2(m^3/h)$	$T_{1,i}(°C)$	$T_{1,0}(°C)$	$T_{2,i}(°C)$	$T_{2,0}(°C)$	ΔP(Pa)	ε(%)
2000	2000	24	13.7	5.9	17.1	80	56.9
3100	3050	23.8	13	5.7	17.4	145	59.6
4400	4410	23.9	13.9	6	16.8	305	55.8

Where A is the total heat transfer area of the plate exchanger (46 m^2) , ΔT_m is the logaritmic main temperature difference in ^0C.

$$v = m / n.\rho.a.b.z \qquad (3)$$

Reynolds number is calculated as:

$$Re = v.\, d_h / \gamma \qquad (4)$$

For Laminar flow (Re<500)

$$(h.\, d_h /K) = 1.68 \,[\, Re.\, Pr.d_h /L \,]\, [\mu/\mu_{\bar{o}}]^{0.1} \qquad (5)$$

For Turbilant flow (Re > 500)

$$(h.d_h /K) = 0.2.\, Re^{0.67} .\, Pr^{0.4} [\mu / \mu_{\bar{o}}]^{0.1} \qquad (6)$$

Since the changes in the physical characteristics of inlet and outlet air at temperatures worked on can be ignored, the characteristic of average temperatures changes are used in the above given equations.
Convective heat transfer coefficient h is obtained from Equation 5 and 6 (Genceli, 1999). Furthermore overall heat transfer coefficient K has been calculated for steady state conditions. The effectiveness of plate exchanger ε , has been found by using NTU (Number of Transfer Units) method for various air flow rates.Effectiveness versus air flow rates plotted as a diagram in Fig.3. The inlet temperatures of air to exchanger is kept constant in two directions as in the experimental study.
For various outdoor- indoor temperature differences ($\Delta T = T_{1,i} - T_{2,i}$) and various constant air velocities (v= 1,2,3,.....,8 m/s), transferred heat energies per meter-square heat exchanger area has been calculated. Results have been presented as a graph in Fig. 4.

Results and Conclusion

The theoretical and experimental results of flow rates-effectiveness are plotted together in Fig.3. Comparison of the results are reasonably good although there are some differences in the slopes of analytical and experimental curves. This differences is believed to be the result of neglected heat losses from channel surfaces.
Since this analytical method has been found appropriate to the experimental results, the same method has been used to obtain velocity- temperature difference - heat recovery relationship (Fig. 4).
Either fresh air requirement of the room or the cross sectional area of the air handling unit is known, the air velocits in the exchanger can be found. Indoor-outdoor temperature difference is also known (the first one is the indoor design temperature, the second one is related to the local climate). One can use the diagram in Fig.4 to get the amount of saved energy per meter square of plate exchanger area.
In optimization work which will be done considering investment and running costs, depending on system capacity, whether the recovery system is economical or not can be decided by means of that diagram.

Nomenclature:

A	Total heat transfer area in the device , m^2
a , b	Distance between the plates , a= 10 mm, b= 4 mm
C_{min}	Heat capacity rate, W/°C
c	Specific heat, J/kg K
d_h	Hydraulic diameter, m
h	Convective heat transfer coefficient, W/m^2K
K	Overall heat transfer coefficient, W/m^2K
k_c	Conduction heat transfer coefficient of plate material, W/mK
L	Lenght of plate, m
m	Mass flow rate, kg/s
n	Number of plates, n= 147
z	Number of vertical tiers
Nu	Nusselt number
P	Pressure, Pa
Pr	Prandtl number
Q	Heat flux, W
q	Heat fluks per unit area, W/m^2
V	Air flow rate(vol), m^3/h
t	Time, seconds
T	Dry bulb temperature, °C
v	Velocity, m/s

Subcripts:

1	Exhaust air
2	Outdoor air
ō	Channel center
i	Inlet air
o	Outlet air

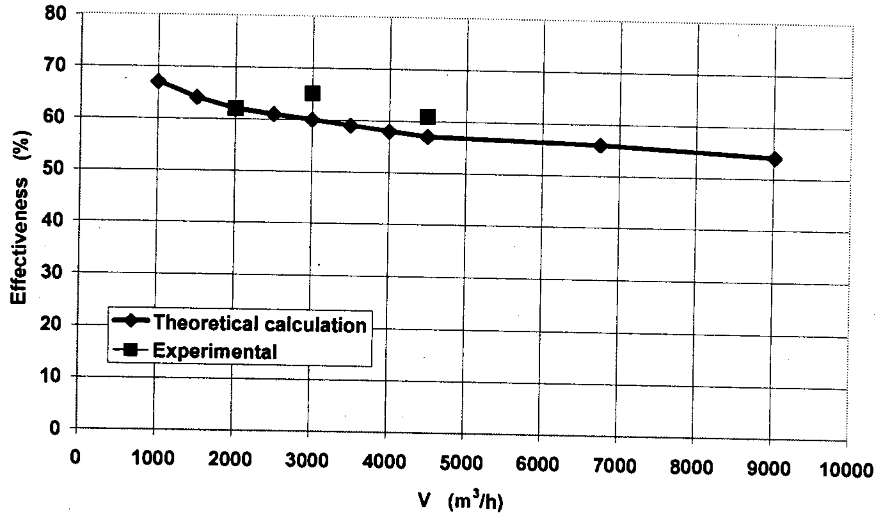

Figure 3-Heat exchanger effectiveness versus air flow rates

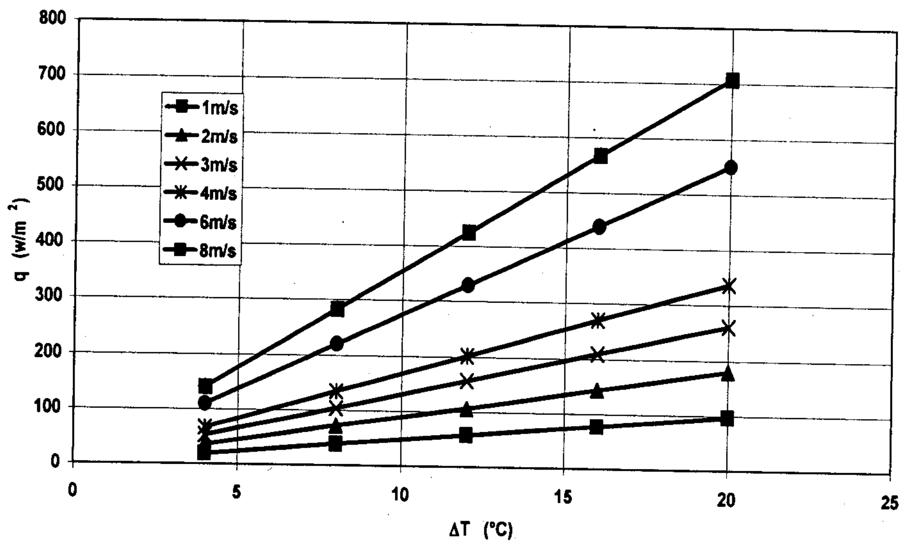

Figure 4-Heat flux recovered per unit area versus indoor-outdoor air temperature difference for verious air velocities

Greek Letters:

δ Thickness of plates, mm

Δ Difference

ε Effectiveness of plate heat exchanger,%

μ Dynamic viscosity,kg m/s

γ Kinematic viscosity,m^2/s

ρ Density,kg/m^3

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