#4629

METHOD OF RATING THE AVERAGE ANNUAL PERFORMANCE OF DOMESTIC AIR-TO-AIR HEAT RECOVERY UNITS

Hendriksen, L.J.A.M.
Netherlands Organization for Applied Scientific Research
TNO Division of Technology for Society
Apeldoorn, The Netherlands

SUMMARY

This paper describes a method of rating the average annual performance of small domestic air-to-air heat recovery units.
The main question here is how the average efficiency can be predicted for one year under average indoor and outdoor conditions. A calculation method is developed and the test data required for the calculations are specified. For this purpose the indoor and outdoor conditions over one year are described with aid of basic assumptions. Finally the method proposed is verified by testing a heat recovery unit in a laboratory. The method is generally applicable but this study refers to The Netherlands.

INTRODUCTION

Energy conservation in dwellings has become a more important factor and as a result houses are being built more airtight in order to reduce heat losses. The use of mechanical ventilation in these newer structures is a must to ensure that humidity, odours and other contaminants do not build up to harmful levels. A solution to indoor pollution, without sacrificing all the energy saved, is to install a mechanical ventilation system which incorporates an air-to-air heat recovery unit.
In continuation of ISSO Publication 11: "Heat Recovery Systems"[1], TNO is carrying out a project to develop a method for rating the average annual performance of small domestic air-to-air heat recovery units.
This paper deals only with heat recovery from exhaust air. This study is motivated by one of the results of an earlier project [2] which showed the great interest in a project of this nature expressed by heat recovery unit manufacturers and dealers. It is generally held that comparing the performance of heat recovery units is of great relevance to all parties and will stimulate the development and application of such apparatus.
The purpose of this project is to:
- describe a method of calculating the average annual efficiency;
- specify the data required for the calculations;
- specify the test requirements and procedures for obtaining performance data to be used in the calculations.

To verify the method proposed, laboratory tests are carried out with one heat recovery unit with a crossflow heat exchanger. This study has not yet been completed, so it is possible that some details of the method will be optimised in the near future.

METHOD OF CALCULATING THE AVERAGE ANNUAL EFFICIENCY

Definitions

The temperature efficiency is defined as the ratio of the temperature change achieved and the theoretical maximum temperature change:

$$\eta T = \frac{T_{c,o} - T_{c,i}}{T_{w,i} - T_{c,i}} \qquad (1)$$

where T represents the dry-bulb temperature of the entering and leaving air streams as shown in figure 1.

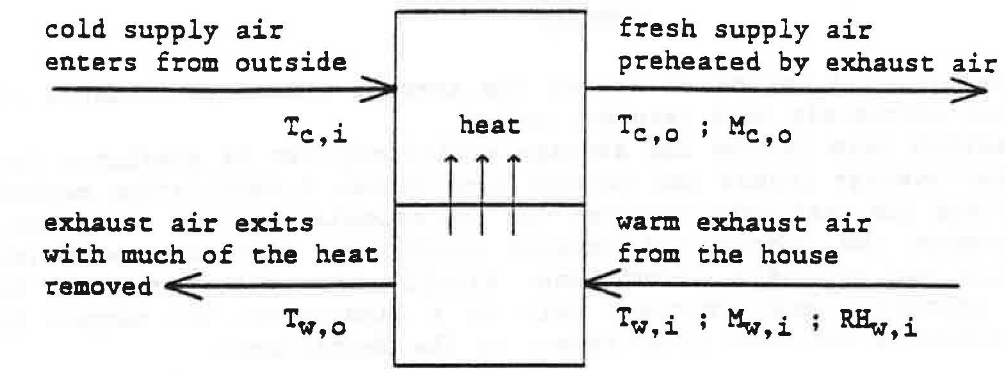

Figure 1 Schematic diagram and flow stream nomenclature for an air-to-air heat recovery unit

However, when the mass flow rates in the exhaust and supply air sides are not equal, the efficiency measure can lead to erroneous impressions. A more adequate description of the heat recovery unit performance is the effectiveness which is defined as:

$$\epsilon T = \frac{M_{c,o}\ (T_{c,o} - T_{c,i})}{M_{w,i}\ (T_{w,i} - T_{c,i})} \qquad (2)$$

The mass flow rates $M_{c,o}$ and $M_{w,i}$ are used as reference values, because those are the mass flows that enter and leave on the application side (house) [3].
The effectiveness definition reduces to equation (1) when the mass flow rates are equal on both sides of the heat recovery unit. In this study a balanced mass flow is regarded and therefore the effectiveness and efficiency are identical.
From equation (1) we can conclude that ηT depends upon $T_{c,i}$ and $T_{w,i}$. From literature and earlier measurements [2] it is known that ηT also depends upon the mass flow rate M and the humidity of the exhaust air $RH_{w,i}$.
By testing a heat recovery unit in a laboratory the temperature efficiency is determined as a function of the four parameters $T_{c,i}$, $T_{w,i}$, M and $RH_{w,i}$.

The results are momentary values of the temperature efficiency, belonging to one combination of these four parameters.

The average year temperature efficiency $[\eta T]y$ is the average of the momentary efficiencies:

$$[\eta T]y = \frac{\sum_{j=1}^{j=k} (\eta T)j}{k} \tag{3}$$

where k is the number of time intervals into which a year is divided. Every time interval represents one $(\eta T)j$.
Because the supply air is preheated in the heat exchanger less energy is needed to heat the supply air to the wanted value T_i.
The higher ηT, the less the needed energy. This effect is expressed in the average annual energy efficiency $[\eta E]y$:

$$[\eta E]y = \frac{\text{saved energy}}{\text{max. saveble energy}} = \frac{\sum_{j=1}^{j=k} (\eta T)j.M.(T_i-T_o)_j}{\sum_{j=1}^{j=k} M.(T_i-T_o)_j} \tag{4}$$

The main question is under which conditions the heat recovery unit should be tested to get sufficient data to determine for every time interval of the year the momentary efficiency.
The four parameters are related to the indoor and outdoor conditions under which a heat recovery unit in practice operates. Therefore it is necessary to describe the indoor and outdoor conditions during a year. For this purpose some basic assumptions are made.

<u>Assumptions</u>

a. We suppose conditions average to The Netherlands.

b. The temperatures of the air streams entering the heat recovery unit are equal to the temperatures inside or outside the house: $T_{w,i} = T_i$ and $T_{c,i} = T_o$.

c. Dutch outdoor conditions are obtained from:
   - NEN 5060: "Short reference year for weather conditions" [4] or
   - The representative year 1964/1965.
   The benefit of NEN 5060 is that the short reference year is composed of 56 days of 24 hours. The representative year 1964/1965 is a normal year of 365 days. Using NEN 5060 means less computer calculation time.
   For each hour of these two years the weather conditions are known. Normally NEN 5060 is only used in calculations concerning energy savings. This study will show if NEN 5060 is applicable in the calculations of the average year performance.

d. Because of assumption c. a day is split up in 24 time intervals of an hour and the calculations are based on hourly values.

e. A day is divided in:
   - a day-period : 08.00 h - 22.00 h;
   - a night-period: 22.00 h - 08.00 h [1].

f. Indoortemperature T_i during: * day-period : T_i = 20 °C.
   * night-period: T_i = 17.5 °C.

g. Houses in The Netherlands, which are mechanically ventilated, usually have two different values of air exchange:
   - low (basis ventilation during most time of the day);
   - high (when extra ventilation is required e.g. during cooking and bathing).

   Airstream flow rate:

low	high
09.00 h - 12.00 h	08.00 h - 09.00 h
13.00 h - 17.00 h	12.00 h - 13.00 h
19.00 h - 08.00 h	17.00 h - 19.00 h

h. If T_o < -8 °C, the freeze protection system is activated and the hourly average temperature efficiency is reduced by 15% [5].

i. The heat production inside the house is leaving aside. Heat recovery is taken into account as long as $T_{c,o} \leq T_i$.
   In practice $T_{c,o} > T_i$ if $T_o > T_i$ or by influence of the fan heat.

j. The indoor humidity X_i is the sum of the outdoor humidity X_o and the moisture production X_p in the house:

   $X_i = X_o + X_p/M$ [6].

   In this study X_p varies per hour according to table 1, but is chosen identical for every day.
   Moisture-infiltration is not taken into account.

   Table 1 Average moisture-production in a house (4 persons) [6]

time	X_p (g/h)
00.00 - 08.00 h	160
08.00 - 09.00 h	1630
09.00 - 12.00 h	200
12.00 - 13.00 h	640
13.00 - 16.00 h	120
16.00 - 17.00 h	370
17.00 - 18.00 h	1320
18.00 - 19.00 h	2620
19.00 - 20.00 h	930
20.00 - 21.00 h	620
21.00 - 23.00 h	180
23.00 - 24.00 h	160

k. The method applies only to the heat recovery unit itself. Energy losses or leakage in any ducting before or after the unit are not taken into consideration.

Testprocedure

Measurements have shown that in practice occurring temperature, ηT depends only upon the temperature difference between the warm air entering and the cold airstream and not upon the absolute temperatures. So test data must be recorded for one value of $T_{w,i}$. In this study $T_{w,i}$ = 20 °C is chosen.

In totally for 24 different measuring points data are recorded, shown in a matrix in table 2.

Table 2 Matrix of measuring points

		$T_{c,i}$ (°C)			
$RH_{w,i}$ (%)	M (%)	-5	2	9	16
40	high				
	low				
65	high				
	low				
90	high				
	low				

The test results are presented in a plot of thermal efficiency versus temperature difference, as in figure 2. For M = high a similar figure can be presented.

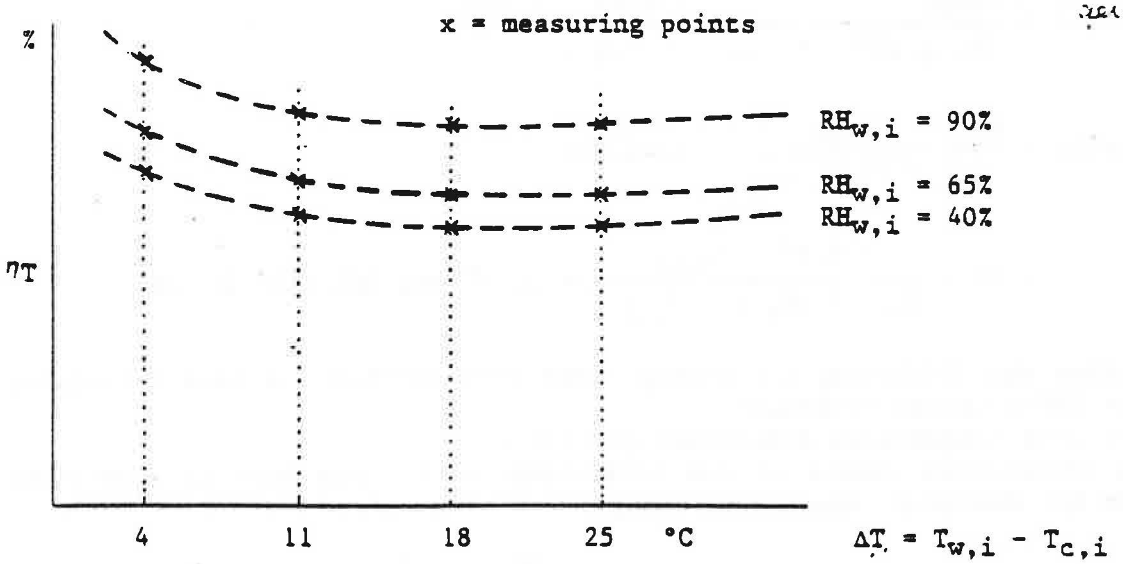

Figure 2 Temperature efficiency versus temperature difference for Tw,i=20 °C and M=low (67%).

Calculations

To determine ηT as an algabraic function of ΔT for one value of M and $RH_{w,i}$ the 4 measuring points of equal $RH_{w,i}$ are curve fitted. In totally we get six algabraic functions. For every hour ηT can be calculated, while for every hour of the regarded year ΔT (= $T_{w,i} - T_{c,i}$), $RH_{w,i}$ and M are known. Regarding $RH_{w,i}$ the exact value of ηT is calculated by interpolation between the two surrounding values of $RH_{w,i}$.
For every hour of the regarded year the efficiency is calculated. The average year efficiency is the average of these calculated hourly values. TNO has developed a computer programme to calculate the average year efficiencies.

Fan corrections

Because the small fans and fan motors used in heat recovery units typically have a low efficiency, most of the electrical energy consumed by the fans is immediately released as heat.
The fan heat will cause an increase in the temperature change of the cold airstream. Therefore it can happen that $T_{c,o} > T_{w,i}$ and so $\eta T > 100\%$. If $T_{c,i}$ approaches to $T_{w,i}$, ηT gets infinite. The hours in which $\eta T > 100\%$ are discarded in the calculation of the average year efficiency.
The corrected temperature efficiency $(\eta T)c$ is calculated by assuming that the fan power is totally converted into heat.
This calculated temperature change can be substracted from the actual temperature change of the airstreams. For the heat recovery unit tested in this study, the fans and fan motors were located in the airstreams downstream of the heat exchanger core. Therefore, all of the heat from the cold airstream fan and none of the heat from the hot airstream fan will be added to the cold airstream.
The effect of the fan heat on the temperature change of the cold airstream is:

$$(\Delta T)f = \frac{P_{vent}}{(M_{c,o}/3600) * Cpl} = \frac{3.5785 * P_{vent}}{M_{c,o}} \qquad (5)$$

$$(\eta T)c = \frac{M_{c,o} * (T_{c,o} - (\Delta T)f - T_{c,i})}{M_{w,i} * (T_{w,i} - T_{c,i})}$$

$$= \eta T - \frac{357.85 * P_{vent}}{M_{w,i} * (T_{w,i} - T_{c,i})} \quad \text{(with } \eta T \text{ and } (\eta T)c \text{ in \%)} \qquad (6)$$

Therefore the following two average year efficiencies are also calculated in the TNO-computer programme:

* Corrected temperature efficiency $[(\eta T)c]y$.
  The temperature change of the airstreams by the fan heat is subtracted from the measured temperature change.

* Corrected energy efficiency $$[(E)c]y = \frac{\sum_{j=1}^{j=k} [(\eta T)c]j.M.(T_i-T_o)}{\sum_{j=1}^{j=k} M.(T_i-T_o)} \qquad (7)$$

In principle $[(\eta E)c]y$ is the most important efficiency, because this calculated number shows the real profit of the heat recovery unit.

RESULTS OF LABORATORY TESTS

TNO has a facility for testing various performance aspects of a residential air heating system. One can produce simultaneously warm and cold air with temperatures and humidities at choice, representative to the indoor and outdoor environments. The attainable minimum air temperature is -10 °C.
It is possible to test a whole system or to test one or more components. One component can be a heat recovery unit. Tests have been performed with a heat recovery unit with a crossflow aluminium core and with fans located in the airstreams downstream of the heat exchanger core. The aim of these tests was to verify whether the 24 measuring points are sufficient for calculating the average annual efficiency in the way proposed. The tests were carried out with one heat recovery unit.

After actually carrying out the tests and calculations, the following conclusions can be drawn:

1. It appears that 4 measuring points are required to determine ηT as an algebraic function of $T_{w,i} - T_{c,i}$, for one value of M and $RH_{w,i}$.
2. The average year performance can be calculated with data of totally 24 measuring points, according to table 2.
3. The difference between calculated efficiencies with NEN 5060 and the representative year 1964/1965 is smaller than 0.5%. So calculating with NEN 5060 is permitted.
4. The influence of the freeze protection on the calculated average year efficiency is smaller than 0.3% and therefore negligible.
5. In this case the influence of the fan heat is approx 5% on the temperature efficiency and 7% on the energy efficiency. Therefore it is necessary to mention the location of the fans, their power consumption and their influence on temperature and energy efficiency in the test report.

Acknowledgement

The study reported above has been made possible by grants of ISSO and Novem.

NOMENCLATURE

Cpl	specific heat of air at constant pressure	kJ/kg.K
E	energy	W
j,k	integer number	
M	mass flow rate	kg/h
P_{vent}	fan power	W
ϵ	effectiveness	%
η	efficiency	%
$(\eta)c$	corrected efficiency	%
$(\eta)k$	efficiency, belonging to time interval k	%
$[(\eta)]y$	average year efficiency	%
RH	relative humidity	%
T	temperature	°C
$(\Delta T)f$	temperature change by fan heat	°C
X	absolute humidity	g/kg

indices

w,i	warm exhaust air stream entering heat exchanger
w,o	warm exhaust air stream exiting heat exchanger
c,i	cold supply air stream entering heat exchanger
c,o	cold supply air stream exiting heat exchanger
i	indoor
o	outdoor
p	production

REFERENCES

[1] ISSO-publikation 11: Heat recovery systems.
ISSO Rotterdam 1982 (in Dutch).

[2] Hendriksen, L.J.A.M. Study optimalisation heat recovery units, installed in houses (fase 2).
TNO-MT/WKT report, Apeldoorn 1988 (in Dutch).

[3] Eurovent 10/2. Heat recovery devices. Methods of testing heat recovery devices for HVAC systems.
Maschinenbau-Verlag GmbH, Frankfurt, 1983.

[4] NEN 5060: "Short reference year for weather conditions.
UDC 697.132:551.506:628.85/86, 1983.

[5] Fisk, W.J. et. al. Performance of residential air-to-air heat exchangers during operation with freezing and periodic defrosts.
Lawrence Berkely Laboratory, University of California, 1985.

[6] Wolfs, B.G. Humidity in dwellings.
I²-Architecture and Civil technics, no 4, 1986 (in Dutch).