Is Heat Recovery in Air Handling Units Efficient?

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Résumé

Des récupérateurs de chaleur sont installées dans de plus en plus d'unités de traitement d'air pour diminuer la consommation d'énergie pour le chauffage et le refroidissement des bâtiments. L'efficacité nominale du récupérateur est souvent utilisée pour calculer l'énergie ainsi économisée. Toutefois, des fuites et des courts-circuits peuvent réduire considérablement l'efficacité de la récupération de chaleur. De plus, l'énergie électrique consommée par les ventilateurs peut s'avérer plus coûteuse que la chaleur récupérée.

Le rendement de récupération réel a été mesuré dans 13 unités de traitement d'air. Dans les trois meilleurs cas, le rendement réel est compris entre 60 et 70% alors que le récupérateur seul a un rendement de 80%. Dans trois cas graves, le rendement global de récupération est inférieur à 10%, et la chaleur récupérée ne compense pas la consommation supplémentaire d'énergie primaire!

Zusammenfassung

Sowohl zur Beheizung wie auch zur Kühlung der Raumluft in Gebäuden werden in zunehmendem Maße Lüftungsanlagen mit Wärmerückgewinnung eingesetzt. Zur Berechnung der Energie-einsparung durch solche Anlagen wird oft deren nominelle Rückwärmezahl verwendet. Durch parasitäre Luftwege im Lüftungsgerät sowie durch Leckagen in der Gebäudehülle wird jedoch die reale Rückwärmezahl eines Gerätes unter Umständen dramatisch reduziert. Hinzu kommt, daß der elektrische Aufwand zum Betrieb der Ventilatoren größer sein kann als die Einsparung von thermischer Energie.

Im vorliegenden Beitrag wurden 13 Anlagen mit Wärmerückgewinnung meßtechnisch untersucht. In den drei besten Fällen lag die reale Rückwärmezahl zwischen 60% und 70%, obwohl deren nominaler Wert 80% betrug. In den drei schlechtesten Fällen lag die Rückwärmezahl unterhalb von 10%. Hierbei benötigte das System mehr Primärenergie als es einsparte.

Abstract

More and more air handling units are equipped with heat recovery systems, with the aim of decreasing the energy use in buildings for heating and cooling. The design efficiency of the heat recovery system is often used to calculate the energy saving. However, parasitic shortcuts in air-handling units and leakage in the building envelope decrease dramatically the efficiency of heat recovery. In addition, the electrical energy used for fans may be more precious than saved heat.

Real energy recovery was measured in 13 air handling units. In the best three cases, the real, global heat recovery efficiency was between 60 and 70% for units having a 80% nominal efficiency. In the three worst cases, the global efficiency was less than 10%. For these cases, the heat recovery system uses more primary energy than it saves.

1. Introduction

Ventilation in buildings - especially in large buildings and advanced low-energy and passive-solar houses - is becoming increasingly important for many reasons. One of them is the excellent standard of thermal insulation, which easily raises the contribution of ventilation losses - depending on the building's compactness and air change rate - to more than 50% of total thermal loss. Another reason for the importance of ventilation is air-tightness of buildings' envelopes, which avoids air infiltration heat loss but does not anymore provide sufficient ventilation. To cope with ventilation requirements with regard to hygiene and building physics, mechanical ventilation systems are of increasing use. In order to reduce energy consumption, ventilation systems with energy-efficient heat recovery systems are almost mandatory.

However, air-handling units may have parasitic shortcuts and leakage [1-5], which can decrease dramatically the efficiency of ventilation and heat recovery. Moreover, leakage in a building's envelope allows warm air to escape outdoors without passing through the heat recovery system. In addition, these units use electrical energy for fans, which may, in some cases, overpass the saved heat. The influence of these phenomena on the real energy saving is addressed in this paper.

2. Effect of leakages and shortcuts on heat recovery

Airflow rates, heat loss and heat recovery efficiency

Let us consider the air- and heat flows in the unit schematically represented in Figure 1. Outdoor air, *o*, enters the inlet grille, *i,* and is blown through the heat recovery system *HR*, where it is either heated or cooled. Then, after subsequent heating or cooling, *rs*, it enters the supply duct, *s*, to be distributed into the ventilated space. As the envelope is not perfectly airtight, the supply air may be mixed with infiltration air, *inf*. A part of the indoor air may also be lost by exfiltration (*exf*). The extract air, *x*, passes through the other part of the heat recovery system, *re*, where it is either cooled or heated. The air is then blown out to the atmosphere, *a*, through the exhaust duct, *e,*.

If the exhaust and inlet grilles are not well situated, a part of the exhaust air may re-enter the inlet grille, resulting in an external recirculation rate R_e. Leakage through the heat recovery system may also result in internal recirculation, from inlet to exhaust R_{ie}, or from extract to supply, R_{xs}.

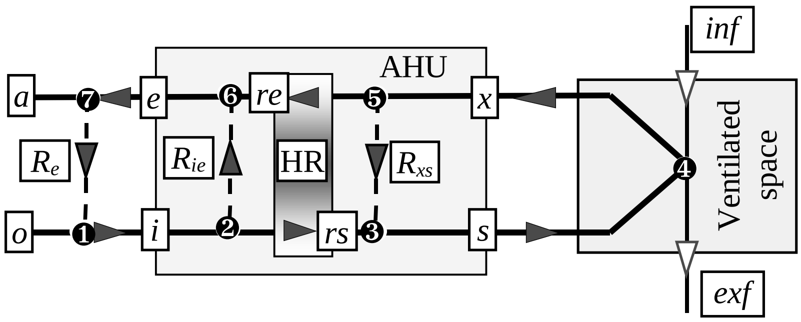

Figure 1: The simplified network representing the air handling unit and ducts. Arrows represent considered airflow rates.

In simplified methods to calculate heating (or cooling) demand of buildings, ventilation heat loss, Φ_V, is calculated by [6]:

$$\Phi_V = c\,\dot{m}\left(\theta_x - \theta_o\right)\left(1 - \eta_G\right) \quad (1)$$

where:

- c is the heat capacity of air, i.e. 1000 J/(kg·K)
- $\dot{m}$ is the mass flow rate of outdoor air in kg/s
- θ_x the temperature of extract air, which is considered as representative of the indoor air.
- θ_o the temperature of outdoor air, and
- η_G is the global efficiency of the heat recovery system.

This global efficiency, η_G, should consider the whole system, consisting of the ventilated building and its ventilation equipment. But, instead, often the nominal temperature efficiency of the heat recovery unit itself, ε_{HR}, is used. This efficiency is:

$$\varepsilon_{HR} \cong \frac{\theta_x - \theta_{re}}{\theta_x - \theta_o} \qquad (2)$$

where the signification of subscripts can be seen in Figure 1. This replacement leads to optimistic results when the air-handling unit has recirculation or when the building is leaky.

Global heat recovery efficiency

If there were no heat recovery, the heat loss of the building, Φ_L, resulting from ventilation is the sum of extract heat flow and exfiltration heat loss, equal to the heat necessary to bring outdoor air to indoor climate conditions:

$$\Phi_L = c(\dot{m}_x + \dot{m}_{exf})(\theta_x - \theta_o) = (\dot{m}_s + \dot{m}_{inf})(\theta_x - \theta_o) \qquad (3)$$

Neglecting latent heat, the recovered heat is:

$$\Phi_R = \dot{m}_{re}(\theta_x - \theta_{re}) = \dot{m}_{rs}(\theta_{rs} - \theta_i) \qquad (4)$$

since, in first approximation, all the heat taken from extract air is given to supply air. The global heat recovery efficiency of the system, η_G , was calculated as a function of the fresh airflow, exfiltration, and recirculation rates, by taking account of mass conservation at the nodes of the system [7]. The full relation is rather complex but, when there is no external recirculation, the global efficiency can be expressed as a function of exfiltration ratio and internal recirculation rates only:

$$\eta_G = \frac{\Phi_R}{\Phi_L} \cong \frac{(1-\gamma_{exf})(1-R_{xs})}{1-R_{xs}\,\gamma_{exf}}\varepsilon_{HR} \qquad (5)$$

where:

$$R_{xs} = \frac{\dot{m}_s - \dot{m}_{rs}}{\dot{m}_x} = \frac{\dot{m}_x - \dot{m}_{re}}{\dot{m}_x} \quad \text{and} \quad \gamma_{exf} = \frac{\dot{m}_{exf}}{\dot{m}_o + \dot{m}_{inf}} \qquad (6)$$

are respectively the internal recirculation rate and exfiltration ratios. Equation (5), illustrated in Figure 2 is a good approximation when external recirculation rate does not exceed 20%.

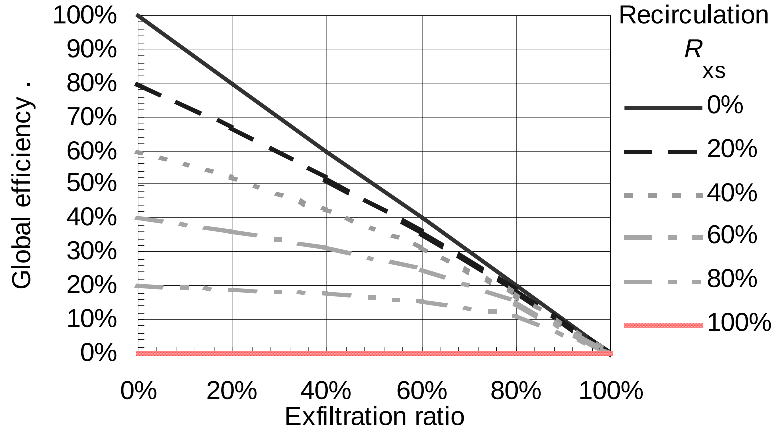

Figure 2: Global heat recovery efficiency in function of exfiltration ratio γ_{exf} and internal recirculation rate R_{xs}. In this figure, ε_{HR} = 100 %, R_{ie}=R_e= 0.

Global efficiency η_G equals the effectiveness ε_{HR} only if there is no exfiltration, and there is neither external- nor extract-to-supply recirculation. Otherwise, η_G is smaller than ε_{HR}.

The inlet to exhaust recirculation, as well as the infiltration ratio have only a small effect on heat recovery efficiency, but reduces the amount of fresh air supplied by the unit to the ventilated space. This recirculation obviously results in an increased consumption of electric energy for the fans, which is approximately proportional to the cube of the airflow rate, without delivering more fresh air. However, such parasitic recirculation is often not noticed, and hence can lead to an undiscovered reduction of indoor air quality.

3. Specific Net Energy Saving

A crucial issue is that HR-systems recover thermal energy but use electrical energy for the fans. As a useful figure to deal with this fact we introduce the **s**pecific **n**et **e**nergy **s**aving per cubic meter of supplied outdoor air (SNES in Wh/m^3) averaged over a heating period, for which the mean outdoor temperature is $\overline{\theta}_o$. This figure is calculated by

$$SNES = \rho_o \frac{\eta_G \Phi_L + \Phi_{fan}\left(f_r - f_p\right)}{\dot{m}} \tag{7}$$

where:

$\Phi_L = \dot{m}\, c\left(\overline{\theta}_x - \overline{\theta}_o\right)$ is the ventilation heat loss, based on average internal and external temperature during the heating season;

f_r is the part of the fan power recovered as heat in the supply air. This factor f_r is close to one for supply fans and zero for exhaust fans;

f_p is a production factor, accounting for the fact that the production of 1 kWh of electrical energy requires much more primary energy;

Only if SNES is positive, a net gain in thermal or primary energy is achieved by the HR-system. Otherwise the system even wastes energy.

Another interesting figure is the Coefficient of Performance (COP), defined by the ratio of recovered heating power and final, used electrical power:

$$COP = \frac{\eta_G \Phi_L + f_r \Phi_{fan}}{\Phi_{fan}} \tag{8}$$

4. Measurement methods

The tracer gas dilution method is used since several years for diagnosis of air handling units [8, 9]. The technique is described in more detail elsewhere [9-11]. Tracer gases are injected at carefully chosen locations in the air-handling unit. Experience has shown that most efficient injection locations are in inlet, supply and extract ducts, that is locations *i, s* and *x* in Figure 1. Tracer gas concentrations are measured at convenient locations, in order to obtain enough equations from conservation of airflow and tracer gas flows to determine all required airflow rates.

The mechanical power delivered by a fan is the product of the volume airflow rate $\dot{V}$ delivered by the fan, times the pressure difference Δp across the fan. Airflow rate is measured as said above, and pressure difference is easily measured with a differential manometer. The electrical power consumed by the fan motor, Φ_{fan}, is measured with a wattmeter, and the fan efficiency is:

$$\eta_{fan} = \frac{\dot{V}\,\Delta p}{\Phi_{\mathrm{fan}}} \tag{9}$$

The temperature efficiency the heat recovery system itself is simply calculated from temperature measurements upwind and downwind the heat exchanger in both supply and exhaust channels, using equation (2).

5. Results

Airflow rates and heat exchanger efficiencies were measured in 10 large units located at the EPFL, and three small, wall-mounted room ventilation units, measured at the University of Siegen, in Germany. The main characteristics of these units are summarised in Table 1.

Table 1: Measured airflow rates with experimental uncertainty band (when available), total and specific fan power in audited units.

	Airflow rates (m³/h)				***Fan power***	
Unit	Outdoor air	Supply air	Extract air	Exhaust air	W	***Wh/m³***
B30	1'900 ±100	2'070 ±70	1'790 ±40	1'600 ±200	990	0.27
TP	2'530 ±80	2'900 ±200	1'860 ±50	1'500 ±200	850	0.19
BH	2'380 ±70	2'480 ±70	1'930 ±40	1'830 ±50	1800	0.42
CS	2'200 ±300	3'400 ±100	3'240 ±90	2'000 ±2000	1800	0.33
E1	5'000 ±200	5'400 ±100	6'000 ±700	5'500 ±700	3710	0.34
E2	15'000 ±2000	16'400 ±700	11'000 ±1000	10'000 ±3000	11800	0.45
E12	11'000 ±400	11'600 ±200	10'000 ±300	9'500 ±900	8180	0.39
E13	16'000 ±1000	17'400 ±700	13'400 ±600	12'000 ±2000	9760	0.33
E14	9'000 ±1000	10'000 ±2000	1'970 ±90	1'000 ±3000	3800	0.35
E15	14'300 ±600	16'200 ±400	3'420 ±70	1'000 ±1000	7970	0.45
HA	25	36	34	24	13	0.22
HB	42	75	74	41	27	0.24
HC	74	87	87	74	32	0.20

Recirculation ratios and efficiencies measured in these units are given in Table 2. In this table, the SNES and COP are calculated with 16 K indoor-outdoor average temperature difference during 210 days, a recovery factor for fans, f_r = 0,5 (taking account that here are two fans in these units, one of them in the supply duct) and a production factor f_p = 3,55, which is the average for low-voltage electricity in Europe according to Frichtknecht et al [12]. Note that a common value used in Germany for f_p is 2.8. French and Dutch regulations give smaller values, respectively 2,58 and 2,56.

Table 2: Outdoor air efficiency, η_o, exfiltration and infiltration ratios γ_{exf} and γ_{inf}, external and internal recirculation rates R_e, and R_{ie} R_{xs}, heat recovery effectiveness ε_{HR}, global heat recovery efficiency η_G, specific net energy saving, SNES in Wh/m^3, and coefficient of performance, COP, of audited air handling units.

Unit	η_o	γ_{exf}	γ_{inf}	R_e	R_{xs}	R_{ie}	η_x	ε_{HR}	η_G	*SNES*	**COP**
B30	97%	16%	0%	6%	7%	0%	86%	70%	56%	1.55	6.5
TP	92%	47%	9%	20%	5%	0%	59%	70%	39%	1.35	8.0
BH	100%	29%	7%	0%	5%	0%	72%	90%	62%	1.18	5.2
CS	68%	77%	76%	55%	1%	0%	31%	30%	9%	-0.05	3.3
E1	98%	8%	17%	0%	7%	0%	92%	80%	69%	1.92	6.7
E2	97%	43%	8%	0%	6%	0%	61%	90%	52%	0.69	4.5
E12	100%	14%	0%	4%	2%	0%	87%	80%	68%	1.45	5.5
E13	97%	25%	0%	0%	0%	0%	77%	70%	54%	1.17	5.5
E14	95%	97%	49%	0%	0%	0%	10%	50%	5%	-0.37	1.8
E15	93%	91%	18%	100%	6%	0%	18%	50%	8%	-0.92	1.5
HA	74%	8%	0%	0%	33%	0%	94%	63%	40%	1.37	6.2
HB	57%	2%	0%	0%	44%	4%	99%	80%	44%	2.21	6.8
HC	68%	0%	0%	0%	39%	25%	100%	90%	55%	2.69	8.2

Major leakage has been observed in several buildings. In three of them, infiltration represents a significant part of the outdoor air, and in four of them, most of the air leaves the building through the envelope instead of passing the heat recovery unit. Significant internal recirculation is observed in the three small units, and external recirculation above 20% is measured in three large units. Leakage and shortcuts significantly affect heat recovery efficiencies, which drop from nominal values between 50% and 90% down to actual values ranging between 5% and 69%. On the average, the heat recovery effectiveness ε_{HR} is 70%, but the global, real efficiency is only 43%. In the best case, an 80% heat recovery effectiveness is reduced by 15% down to a 69% real efficiency.

Specific net thermal energy savings (SNES) can be very small or even negative. In the best case, it reaches 2.7 Wh/m³, corresponding to 8 K average temperature increase of fresh air. It should be also noticed that the COP can overpass 8, but might also be much smaller than expected, as it is often the case for air-to-air heat pumps.

Best net energy saving in large units (E12 and E13 in tables 1 and 2) is 80'000 to 90'000 kWh per winter season, but another unit (E15) actually spills as much energy. Small units, (HA, HB and HC) save between 80 kWh (HB) and 350 kWh (HC) during an entire season. From energetic and economic aspects only, such ventilation units are disadvantageous and hard to recommend. Note that these results are obtained when the heat recovery is functioning.

6. Conclusions

Heat recovery from extract air is often installed in advanced low energy buildings in order to ensure efficient ventilation at low energy cost. However, global efficiency of heat recovery depends significantly on air infiltration and exfiltration, which should be minimized during the heating period. Internal and external recirculation also decrease the efficiency of the heat recovery units. Moreover, electrical energy for fans is used in order to supply fresh air and to recover thermal energy from exhaust air.

Characteristic figures for the evaluation of ventilation units with heat recovery have been defined and measured using the tracer gas dilution method. The most important of them are global efficiency of heat recovery and specific net energy savings. For several examined ventilation units energetic savings were small or even negative. Even if best technical performance is assumed (airtight building, ε_{HR} = 90%, specific fan power equal to 0.2 Wh/m^3) the economic viability of small ventilation units remains questionable. This, however, does not affect the other qualities of ventilation systems such as steady supply of fresh air with low concentrations of contaminants.

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