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Heat Recovery in Ventilation Systems.

F.Steimle and S. Schädlich

Universität Essen, Angewandte Thermodynamik und Klimatechnik, Universitätsstr.15, W-4300 Essen 1, Germany

Heat recovery in ventilation systems

Synopsis

In well insulated buildings the ventilation heat is sometimes higher than the heat losses by transmission. For a air change rate of 0,8 per hour the specific heat flux must be calculated with 25 W/m^2, so heat recovery can save some energy. In all considerations the saving in the heating system must be compared with the additional energy for the fans, because this energy is of a higher quality.

To optimize the heat recovery system, the different designs of the heat exchanger, the annual running hours and the annual hours for heat recovery must be taken into account. Heat recovery heat exchangers can be optimized with an efficiency of about 60%. To reach a higher overall efficiency a heat pump included in the system is a good possibility in special cases. To compare the different systems and combinations an overall COP can be derived not only for the heat pump but also for the heat exchanger. This is very important for the right decisions, because the aim can not be to save heating energy by spending more electricity.

List of symbols

A	$[m^2]$	-	heat exchanger area
c	$[J/kg\,K]$	-	specific heat capacity
$\dot{C}$	$[W/K]$	-	heat capacity flux
F_z	$[-]$	-	form coefficient
$\dot{H}$	$[W]$	-	enthalpy flux
k	$[W/m^2K]$	-	overall heat transfer coefficient
K_{St}	$[-]$	-	constant of material
l	$[m]$	-	length
$\dot{M}$	$[kg/s]$	-	mass flow
P	$[W]$	-	electric power
Δp	$[Pa]$	-	pressure drop
$\dot{Q}$	$[W]$	-	heat flux
t	$[K]$	-	temperature
v	$[m^3/kg]$	-	specific volume
$\dot{V}$	$[m^3/s]$	-	air flow rate

α	$[W/m^2K]$	-	heat transfer coefficient
ϵ^*	[-]	-	heat recovery efficiency
η	[kg/m s]	-	dynamic viscosity
η_f	[-]	-	efficiency of power (fan)
κ	[-]	-	number of thermal units, ntu
λ	[W/m K]	-	thermal conductivity
ϕ	[-]	-	thermal characteristic of heat exchange
ω	[-]	-	ratio of heat capacity flux
ϑ	[K]	-	temperature difference
ϑ_o	[K]	-	maximum temperature difference

1. Introduction

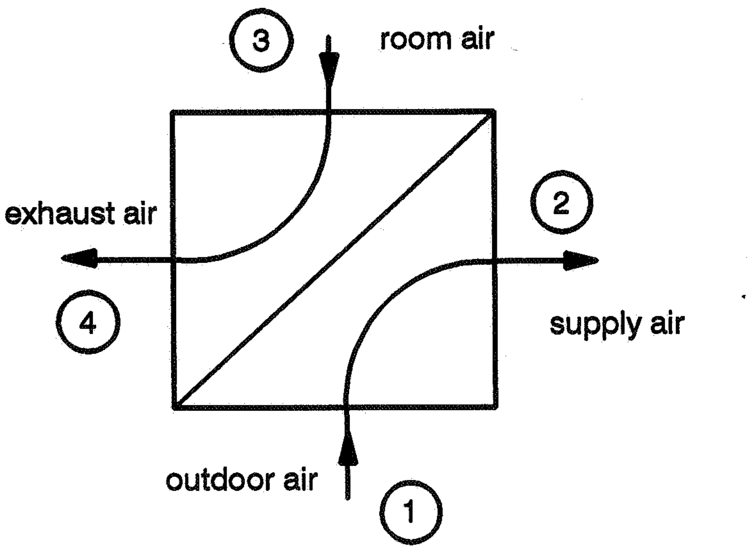
Commonly heat exchanger systems are suited for heat recovery in big buildings, but nowadays energy consumption becomes also important in the domain of dwellings. Especially thermal high insulated buildings have only a transmission heat loss between 10 to 15 W/m^2, but a constant ventilation heat loss of 25 W/m^2. For decreasing this first part, different types of air-to-air heat recovery units are used, exhausting the air from the most polluted rooms (kitchen, bathroom, toilet, etc.) and taking it through a heat exchanger. Here the supply air is warmed up and flows into the bedrooms and living rooms.

The heat recovery units can achieve an efficiency of 60% depending on their design. A recouperative system allows only the heat recovery of the sensible heat whereas regenerative systems recover supplementary moisture. Conventional recouperative heat exchangers consist of solid walls like tubes or plates which divide the exhaust and the supply air. A more sophisticated system is the heat pipe system with finned tubes filled with a layer of wick containing a working fluid. One end of the pipes is exposed to the warm exhaust air stream, so that the liquid evaporates and the vapour flows to the other end which is in contact with the cold supply air stream. Here the vapour condenses and the heat is transferred to the cold air with an efficiency of nearly 70%.

One of the regenerative systems is the rotary air-to-air heat exchanger, a large slowly rotating wheel containing a metallic or non-metallic media for heat and moisture recovery which allows the transfer of sensible and latent heat. This wheel isn't used in dwellings because of the danger of odour transfer.

2. Characteristic numbers of heat exchangers

For a description of the heat exchange process between two fluids some characteristic numbers are useful.



heating relation	release from room air	intake of outdoor air
sensible heat	$\Phi_{t,ra} = \frac{t_3 - t_4}{t_3 - t_1}$	$\Phi_{t,oa} = \frac{t_2 - t_1}{t_3 - t_1}$
water content	$\Phi_{x,ra} = \frac{x_3 - x_4}{x_3 - x_1}$	$\Phi_{x,oa} = \frac{x_2 - x_1}{x_3 - x_1}$
enthalpy	$\Phi_{h,ra} = \frac{h_3 - h_4}{h_3 - h_1}$	$\Phi_{h,oa} = \frac{h_2 - h_1}{h_3 - h_1}$

FIG. 2.1: Air flows through a heat exchanger and different thermal characteristic of heat exchanges.

Fig. 2.1 shows the different air flows through a heat exchanger and the different definitions of the thermal characteristic of heat exchange Φ relating to the temperature, the water content and the enthalpy. Considering the maximum temperature difference ϑ_0 of the heat exchanging streams and the temperature difference Δt of one of the fluids between inlet and outlet of the heat exchanger, the thermal characteristic of heat exchange is:

$$\Phi_t = \frac{\Delta t}{\vartheta_0}$$

For recouperative heat exchangers it is also useful to consider the transfer of moisture or of enthalpy. So there are additional thermal characteristics of heat exchange:

$$\Phi_x = \frac{\Delta x}{x_0} \qquad \Phi_h = \frac{\Delta h}{h_0}$$

The heat flux of a mass flow $\dot{M}$ with the spezific heat capacity c and temperature decrease Δt is determined by the following relation:

$$\dot{Q} = \dot{M} \cdot c \cdot \Delta t = \dot{M} \cdot c \cdot \Phi \cdot \vartheta_0$$

Assumed there is no heat exchange with the environment, the given heat from the one air stream (1) is completely taken from the other air stream (2). Then the quotient of the temperature difference is the following:

$$\frac{\Delta t_1}{\Delta t_2} = \frac{\Phi_1}{\Phi_2} = \frac{\dot{M}_2 \cdot c_2}{\dot{M}_1 \cdot c_1}$$

Considering the equation for the heat transfer through the wall

$$\dot{Q} = k \cdot A \cdot \vartheta,$$

the number of thermal units κ is defined as:

$$\kappa = \frac{k \cdot A}{\dot{M} \cdot c}$$

Another characteristic number is the relation of the heat capacity flux ω:

$$\omega = \frac{\dot{C}_1}{\dot{C}_2} = \frac{\dot{M}_1 \cdot c_1}{\dot{M}_2 \cdot c_2} = \frac{\Delta t_2}{\Delta t_1} = \frac{\Phi_2}{\Phi_1} = \frac{\kappa_2}{\kappa_1}$$

Here the bigger heat capacity has to be in the denominator of ω, so ω is a number between 0 and 1.

The amount of Φ, κ and ω depends on the type of heat exchanger (parallel, counter or cross flow).

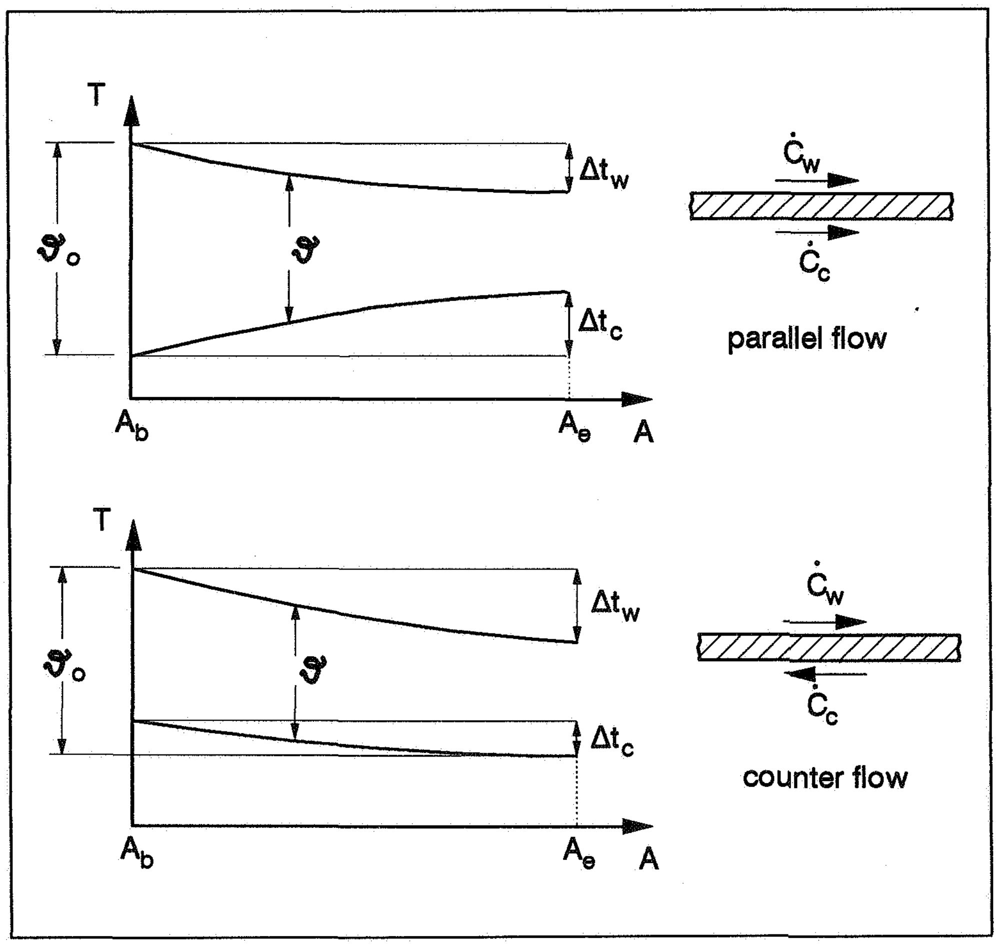


FIG. 2.2: Temperature relations in different types of heat exchangers.

Fig. 2.2. shows the temperature relations between a warm (w) and a cold (c) air flow in parallel and counter flow heat exchangers.

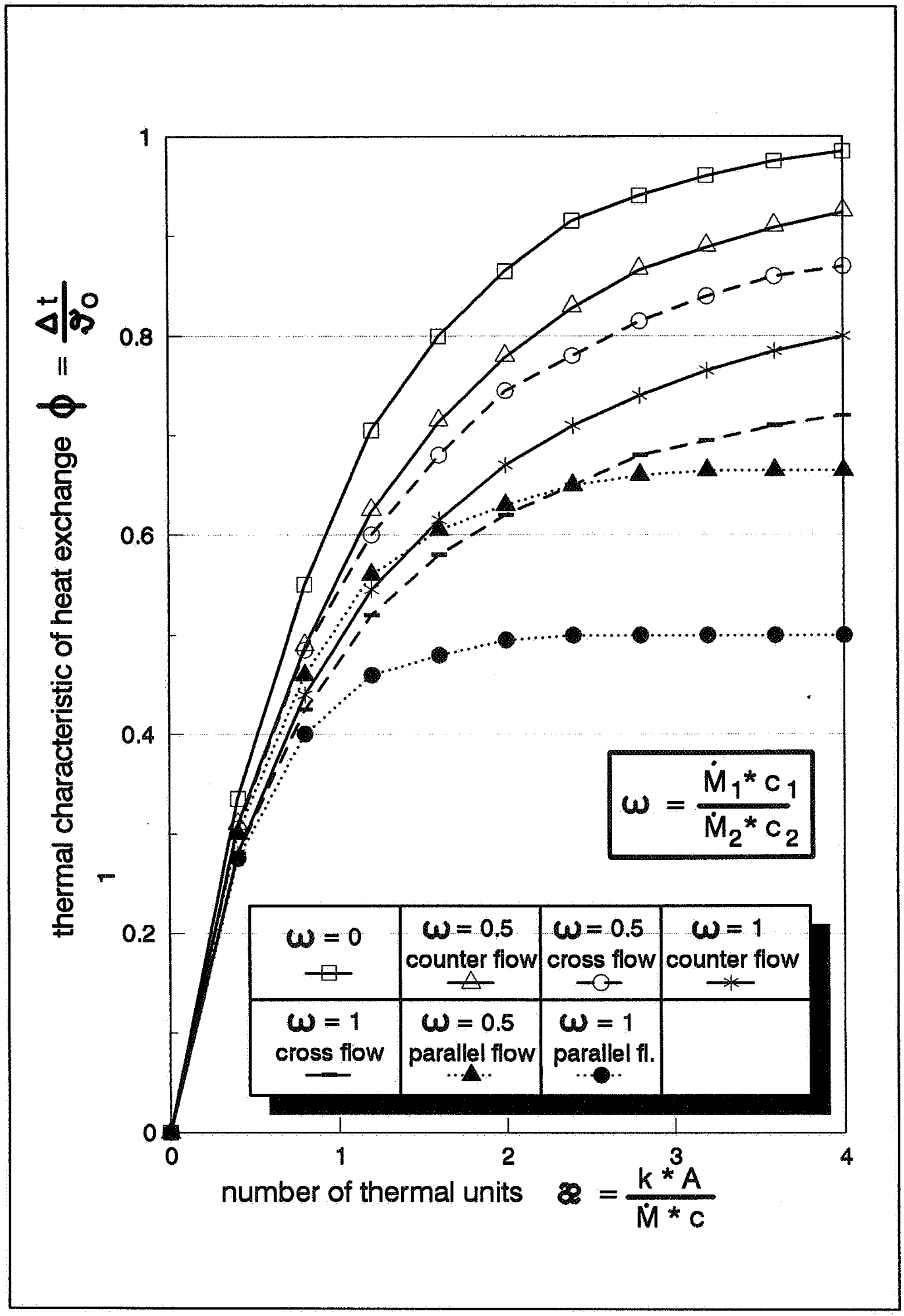


FIG. 2.3: Operating characteristic for different types of heat exchangers.

In FIG. 2.3 the thermal characteristic of heat exchange Φ is plotted against the number of thermal units κ with the parameter ω for different types of heat exchangers. The relations between Φ, κ and ω show the highest thermal characteristic of heat exchange for counter flow heat exchangers. Remembering Fig. 2.2 it is evident that parallel flow heat exchangers have the lowest Φ of only 0.5 (ω = 1) because of the temperature relations between the cold and warm air stream.

But Fig. 2.3 is also useful to optimize heat exchangers. Assuming a counter flow system with an efficiency of 60% and ω = 1, then the thermal characteristic of heat exchange is 1.5. To improve the efficiency factor up to 80%, we get a Φ of 4, so a 2.7 times larger heat exchanger surface is needed. This provides beside higher investments an increasing pressure drop and a higher power consumption of the fans. So there is an optimal operating point for a fan depending on several influences.

3. Material of heat exchangers

For heat exchangers different materials, f. ex. aluminium, glass, plastic, etc. with various thermal conductivities λ are used. The overall heat transfer coefficient k is dependent on the heat transfer coefficient α and the thermal conductivity λ.

$$\frac{1}{k} = \frac{1}{\alpha_W} + \frac{s}{\lambda} + \frac{1}{\alpha_C}$$

Assuming the same α - values for the cold and the warm side of the heat exchanger of for example $\alpha_w = \alpha_c = 17\ W/m^2K$ and a thickness of s = 1 mm it follows:

aluminium	(λ = 200 W/mK)	k_{al} = 8,5 W/m^2K
glass	(λ = 0,76 W/mK)	k_{gl} = 8,4 W/m^2K
plastic	(λ = 0,35 W/mK)	k_{pl} = 8,3 W/m^2K

So it is evident that the impact of material is of secondary order and the most important value for the overall heat transfer coefficient is the α - value which increases with the mass flow. But at the same time the number of thermal units κ and also the thermal characteristic of heat exchange Φ decreases, because M is in the denominator of κ.

Beyond that, the time for passing the heat exchanger is too short to produce a real improvement of heat flux. In addition an increasing velocity provokes a higher pressure drop, so that a higher power consumption is required. To estimate the impact of all influences, a theoretical treatment of heat transfer and fan power is useful.

4. Theoretical treatment

Based on theoretical knowledge of the heat transfer mechanism and the influences on the fan power consumption, various equations are derived to determine the relation between the different characteristic numbers. The precise mathematical formulation of the heat transfer problem is the Nusselt number:

$$Nu = \frac{\alpha \cdot l}{\lambda} = \frac{\alpha \cdot \dot{M}}{\lambda \cdot \eta} = F_z \cdot \left(\frac{(\Delta p / l) \cdot \dot{M}^3}{a \cdot \eta^4} \right)^{0.37}$$

Under consideration of the fan power

$$P = \frac{\Delta p \cdot \dot{V}}{\eta_f} = \frac{\Delta p \cdot \dot{M} \cdot v}{\eta_f}$$

we get the following equation:

$$P = \frac{1}{\eta_f} \cdot \dot{M}^{3.4} \cdot \left(\frac{2 \cdot \kappa}{K_{St} \cdot A \cdot F_z} \right)^{2.7}$$

with K_{St} depending only from the material and the fluid:

$$K_{St} = \frac{\lambda \cdot \eta}{c} \cdot \left(\frac{1}{a \cdot \eta^4 \cdot v} \right)^{0.37}$$

5. Boundary conditions and optimization

Under consideration of the efficiency of electrical generation in power stations the boundary condition for useful heat recovery is:

$$3 \cdot P = Q$$

$$\frac{3 \cdot \Delta p \cdot \dot{M} \cdot v}{\eta_f} = \dot{M} \cdot c \cdot \Phi \cdot \vartheta_0$$

So the pressure drop additionally provoked by the heat exchanger is limited by the following equation:

$$\Delta p = \frac{c \cdot \Phi \cdot \vartheta_0 \cdot \eta_f}{3 \cdot v}$$

To optimize the heat exchanging process, the required electrical power and the non-transferred heat flux has to be a minimum value:

$$3 \cdot P + \Delta\dot{Q} = \text{minimum}$$

The non-transferred heat flux is:

$$\Delta\dot{Q} = \dot{M} \cdot c \cdot \vartheta_0 \cdot (1 - \Phi) = \dot{M} \cdot c \cdot \vartheta_0 \cdot \frac{1}{1 + \kappa}$$

Then it follows:

$$\frac{3 \cdot 1}{\eta_f} \cdot \dot{M}^{3,4} \cdot \left(\frac{2 \cdot \kappa}{K_{St} \cdot A \cdot F_z}\right)^{2,7} +$$

$$\dot{M} \cdot c \cdot \vartheta_0 \cdot \frac{1}{1 + \kappa} = \text{minimum}$$

This equation allows only a qualitative judgement of the impact of the different influences like the area and the length of the heat exchanger, the mass flow, the kind of fluids, etc..

6. Investigations

In order to determine the efficiency of a special heat pipe system of a ventilation window, a ϵ^*-value is defined as the relation of the enthalpy flux difference between the room air and the exhaust air to the required fan power.

$$\epsilon^* = \frac{\dot{H}_{\text{room air}} - \dot{H}_{\text{exhaust air}}}{P_{\text{fan}}}$$

Fig. 6.1 shows the fan power increasing with the mass flow and its impact on the heat recovery efficiency ϵ^*. Under consideration of the factor 3 for the generation of electrical power for this system only a fan power up to 40 W is economical for heat recovery.

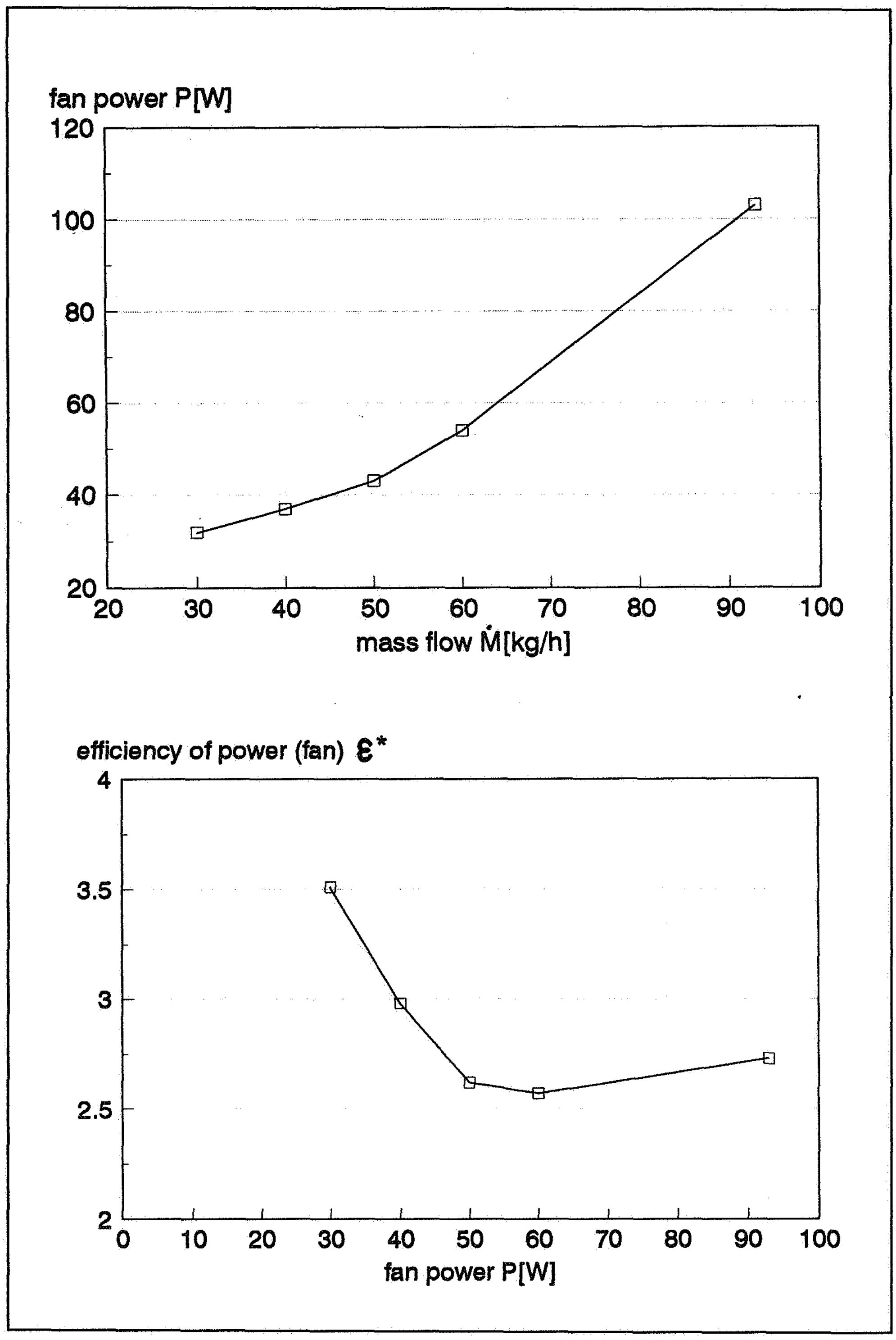


FIG. 6.1: Mass flow and efficiency of a heat pipe system as a function of the fan power.

7. Annual running hours

A heat exchanger in a ventilation system represents an additional pressure drop, so the fan power increases. In order to calculate a realistic efficiency for a heat recovery system, the annual hours for the heat recovery in relation to the total annual running hours of the system with the higher pressure drop must be taken into account.

Here some ventilation systems offer the possibility to take off the heat recovery unit in times where it isn't needed, so the required fan power decreases.

8. Summary

The impact of ventilation heat losses are increasing because of the improved insulation of buildings, so heat recovery systems are used to save more energy. But the higher pressure drop of the heat exchangers causes additional power consumption of supply and exhaust fans. This must be offset against the energy reclaimed from the air streams under consideration that electricity is of a higher quality than heating energy.

Under consideration of all influences on heat exchanger efficiency taken by theoretical and practical estimations as well as calculations of the annual running hours and the required additional fan power, it is possible to design a economical heat recovery system for different applications.

8. References

/1/ Steimle, F.
"A general analogy between heat transfer and pressure drop in turbulent flows"
Commissions II & III, London 1970, Annex 1970-1, Supplement of the Bulletin of the International Institute of Refrigeration (Extract)

/2/ Steimle, F.
"The effect of superimposed free and forced convection on heat transfer and pressure drop correlation"

Commissions B-1, B-2 and E-1, Freudenstadt 1972, Annex 1972-1 Supplement of the Bulletin of the International Institute of Refrigeration (Extract)

/3/ Steimle, F., Gräff, B.
"Investigation about heat recovery of ventilation windows"
Institut für Angewandte Thermodynamik und Klimatechnik Universität Essen, 1979