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The Study on Renewable Energy from Ventilation Recuperation

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Abstract. The paper goal is to give the short study of the possible energy saving by forced ventilation with integrated temperature recuperation in the living buildings, which must be permanently ventilated. The energy losses by ventilation with better constructions isolation grow relatively up. The basic thermal calculations explain the efficiency dependence on the outer temperature and the saved energy or power. The flow calculations give the principal dimensions of recuperators and the ventilators power consumption. The comparison of advertised savings of expensive products with typical temperatures in middle Europe is critical to the price of these devices produced in short series. The results from flow and thermal calculation can help not only in the recuperating unit choice in shopping, but also in DIY hobby realisation of simple counter flow recuperator for house. Only the mass production can reduce the price and bring wider use of this energy saving device.

Key words

Ventilation, recuperation, air flow, efficiency calculation, aerodynamic resistance,

1. Introduction

The saved energy from ventilation losses in living buildings is important especially in very hot or very cold countries, where the temperature difference between the room and outside can exceed 20K during longer period. It is the great truth that the saved energy is the cheapest one, and most ecological, but the physical limits for natural heath flow, without heat-pumping cannot be broken. Also the latent heat from the water condensation from internal humid air cannot be exploited and therefore instead heat recovery, there is spoken here about the temperature recovery.
The ventilation of living buildings is absolutely necessary, because every living body produces by its metabolism CO_2 and H_2O as a gaseous waste. If there is no fresh air available in the closed rooms, the carbon dioxide CO_2 concentration grows over dangerous value 3000 ppm when the body feels uncomfortable and it can die later from oxygen deficit. Because the CO_2 is produced from O_2, its concentration in the air decreases due to breath. The ventilation, which is the air exchange, brings not only fresh air from outside, but it keeps also the optimal water concentration in the air, which should be near to the value 50% of relative humidity. Higher concentration causes the condensation on cold walls with consequent growth of dangerous bacteria, moulds and fungi. The ventilation is very healthy, but the fresh air from outside must be heated and the warm air running from inside represents the energy losses. The modern buildings are more hermetical with seals in windows and instead of natural ventilation by open windows the artificial forced ventilation is introduced.
There are numerous ventilation units on the market from professional production and also a lot of calculations of energy savings to support the sale. There are also other calculations from HVAC construction specialists, which are more realistic. They are giving both sides of this air heat recovery, which is not only the heat energy savings, but also the additional expenses for more expensive electric energy consumed by the ventilators drive and the maintenance expenses including the filters, cleaning and disinfection. The published internal configuration of these recovery exchangers is rare and the calculations of exchanger design are not available.
Author of this paper has a long experience from heat and ventilation calculations of rotating electrical machines and he wants to present some basic calculation procedures and relations for successful realization of simple working model.

2. Free energy from recuperation

The thermal energy losses E [J] by ventilation can be easily evaluated by formula:

$$E = Q \gamma c \Delta\vartheta_{IE} t \qquad (1)$$

Where Q [m^3/s] is the air flow, γ [kg/m^3] is the air density, c [J/kg K] is the heat capacity, $\Delta\vartheta_{IE}$ [K] is the temperatures difference between the inside and outside, and t [s] is the time of ventilation. The material constants for the air are in Table I. The temperature difference depends on the geographical location, the season, the actual weather and daytime (sunshine, wind), generally said the climate. The airflow is dependent on the number of persons in the house, their physical activities and other activities connected with the water evaporation, like cooking, showering etc.

Table I. – The Air Constants at 0°C and 1,01325 bar

Thermal capacity	c	J/kg K	1003
Density	γ	kg/m^3	1,29
Oxygen		% vol	20,95
Nitrogen		% vol	78,09
CO2		ppm	334
Kinematical viscosity	ν	m^2/s	1,5 E – 5

A. The ventilation air flow

It can be determined empirically, that for keeping optimal CO_2 and H_2O concentration, the half of air volume in the living room should be exchanged each hour. Let us assume the typical apartment 60m^2 with the walls height 2,4m that has the total volume 144m^3, the air flow must be:

$$Q = 72\ m^3/h = 0{,}02\ m^3/s \qquad (2)$$

B. The temperature difference

The optimal interior temperature for wellbeing is about 20°C or high in dependence of outer walls isolation, because the sum of air and wall temperatures should be 36°C, which implies the worse is the walls isolation and their surface temperature consequently, the higher must be the air temperature.

The outer temperature changes from day to day randomly depending on the weather and season, but they are some typical average temperatures during the year. The lowest average temperatures in Czech Republic are in January, December and February can be taken equal, also November and March etc. as can be read from the datasheet in Table II. In fact the local data are different for various altitude and geographic position.

Table II. – Average temperatures in Czech Republic from [4] and the temperature differences for room temperature 20°C

Month	9.	10.	11.	12.	1.	2.	3.	4.	5.
[°C]	**12,5**	**7,4**	**2,4**	**-1**	**-7,1**	**-1,2**	**2,6**	**7,3**	**12,4**
$\Delta\vartheta_{IE}$	7,5	12,6	17,6	21	27,1	21,2	17,4	12,7	7,6

They are only three summer months when there is no need of heating and no ventilation losses can be created. Practically in September and May can be ventilation realised only during the day, when the temperature is higher than in the night and the temperature difference from Table II is too small to be effectively exploited.

C. The power losses by ventilation

The power P = E/t can be calculated from (1) and the values for various temperature differences are in Table III. The one day and one month energy losses by ventilation are in the next rows and the amount of energy, which is to recovery in the bottom part of Table III, evaluated in kWh and GJ is really impressive.

For realistic efficiencies 80 and 60 % respectively the amount of saved energy and money is also not negligible. Two energy prices are chosen for most expensive electrical energy (EL) and for the cheapest energy from centralised heat (CH) production.

Table III. - Type Sizes

$\Delta\vartheta_{IE}$ [K]	**12**	**17**	**21**	**27**
P[kW]	**0,288**	**0,408**	**0,504**	**0,648**
kWh/day	6,912	9,792	12,096	15,552
kWh/month	207,36	293,76	362,88	466,56
	Oc Ap	No Ma	De Fe	Jan
Recovery efficiency			80%	60%
YEAR	kWh	2 195	1756	1317
	GJ	7,90	6,32	4,74
EL (5CZK/kWh)	CZK	**10 973**	**8778**	**6584**
CH (500CZK/GJ)	CZK	**3 950**	**3160**	**2370**

The recovery of ventilation losses is not completely free, because the forced ventilation needs two ventilators (one for each direction) and their drives must be supplied from the most expensive electrical energy, which must be subtracted from the energy saving. The customer is more interested in the money saving than in the energy saving. The minimal ventilator power consumption and maximal energy recovery needs flow calculations.

3. Heat recovery exchanger

The counter flow heat exchange is the basic principle of effective energy recovery at low temperature difference between the outer and inner air $\Delta\vartheta_{IE}$. Along all the length of two counter flows the temperature difference $\Delta\vartheta_{CF}$ [K] is the same as is clear from Fig.1 where the both flows are separated by thin separator with low thermal resistance.

There are the two mirror heat transfers from warm air to separator and from separator to the cold air Fig.3. Both these transfers are due to convection and this is speed dependent. For convention is the formula:

$$\Delta\vartheta = P / (\alpha\, S_S) \qquad (3)$$

Where the power P [W] is known from Table III, α [W/K.m^2] is the heat transfer coefficient and S_S [m^2] is the surface of separator.

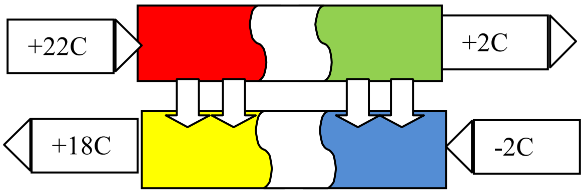

Fig.1. Schematic temperature recuperation with the counter flow and constant difference $\Delta\vartheta_{CW} = 4K$

For the precise calculation also the heat transfer by conduction trough separator must be calculated. The formula (3) can be used with substitution for α_S from the separator thickness b_S [m] and thermal conductivity λ_S [W/Km]

$$\alpha = \lambda_S / b_S \qquad (4)$$

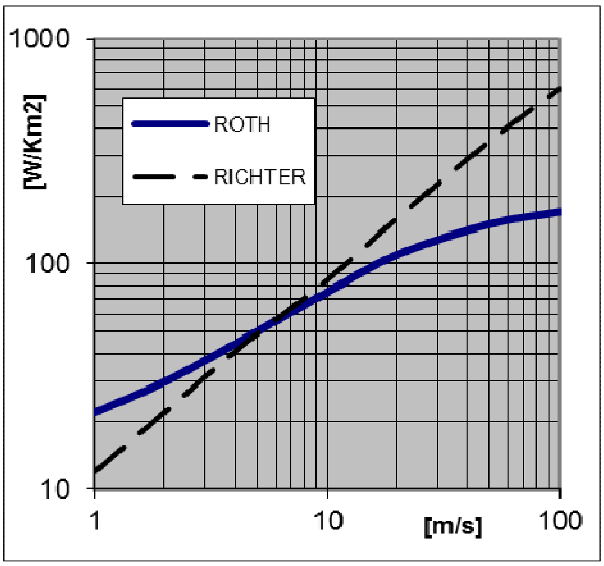

Fig.2 The heat transfer coefficient by Roth and Richter [2] for forced air cooling

The heat transfer coefficient speed dependence is in Fig.2 and in the speed range between 1 and 25m/s can be approximated by formula:

$$\alpha = 20\ v^{\,0,58} \qquad (5)$$

The sufficient speed range for recuperation is 3 – 15m/s with preference of lower speeds, due to low power, which necessary to drive the air flow.

A. Equivalent thermal circuit

The equivalent circuit of energy transfer from the warm to the cold air can is drawn in Fig.3 where the total $\Delta\vartheta_{CW}$ is divided between thee thermal resistors. The indexes are W – warm, S – separator, C – cold.

At the bottom of the Fig.3 is illustrated the temperature distribution along the coordinate perpendicular to the air flow.

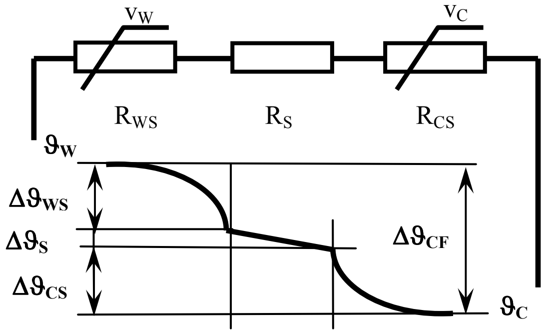

Fig.3 The counter flow thermal equivalent circuit and the principal temperature distribution

B. Separator surface

As is drawn in Fig.1 the warm and cold ducts must have sufficient area separator for the heat flux from the warmer air at low enough temperature difference to receive good efficiency. From Fig.3 is evident that the heat transfer between the air flow and the separator surface is speed dependent. The separator surface for chosen temperature rise (under 2K) can be calculated by (3) for various air speeds **v** in the channel which give the appropriate coefficient **α** using (4).

Table IV. – The separator surface area for various air speeds (the power flow P = 650W, one side temperature rise 2°C)

v	[m/s]	3	5	10	15	air speed
α	[W/K.m^2]	38	51	76	96	
S_S	**[m^2]**	**8,55**	**6,37**	**4,28**	**3,39**	

C. The separator temperature rise

The optimal material for the ducts separator is the thin metal sheet with good thermal conductivity $\lambda_S = 50 - 400$ but the plastic material is much cheaper and easy to elaboration, therefore the PVC or PE foil can be used with advantage. Using the (4) the equivalent heat transfer coefficient for $b_S = 0{,}1$mm and $\lambda_S = 0{,}2$W/K.m results in $\alpha_s = 2000$W/K.m^2, which is more than twenty times higher than are **α** in Table IV.

According to (3) the temperature rise on the separator is only few percent of values from Table IV., and the separator temperature rise can be neglected. The cheap material for separator can be also changed every year to keep the recuperation in hygienic state.

The opposite side of the separator can be realised by the same foil complemented by layer of expanded polystyrene or similar foam material with thermal conductivity $\lambda_i = 0{,}02$W/K.m, which is enough for good isolation. If the thickness of isolating layer is only $b_i = 5$mm, the equivalent thermal resistance, according to (4) is only $\alpha_i = 4$W/K.m^2 and the leakage of the heat is therefore minimal.

4. The air flow calculations

The total air flow Q [m^3/s] is known from (2) and the speed v [m/s] in the duct depends on its cross section area S_C [m^2], which can be calculated:

$$S_C = Q / v \tag{6}$$

The optimal channel configuration should have low height **h** €(6 – 10mm) and wide width **b**, which is also the width of separator. The length of each channel L [m] = S_S / b. From the channel dimensions the hydraulic diameter D_e [m] can be calculated by formula:

$$D_e = 4\, S_C / 2(b + h) \tag{7}$$

The Reynolds number **Re** [-] distinguishes between the laminar and turbulent flow, its value is:

$$Re = D_e\, v / \nu \tag{8}$$

where $\boldsymbol{\nu}$ [m^2/s] is kinematical viscosity from Table I. For **Re** ≥ 2300 (2000 – 3000) the flow is turbulent and the surface friction coefficient **λ** is calculated according to:

$$\lambda = 0{,}3164 / Re^{0{,}25} \tag{9}$$

The aero dynamical resistance Z [kg/m^7] of one channel is:

$$Z = \xi\, \gamma / 2\, S^2 \tag{10}$$

Where the density **γ** [kg/m^3] is from Table I. and the channel friction coefficient is:

$$\xi_F = \lambda\, L / D_e \tag{11}$$

Table V. – The flow calculations for various air speeds (the air flow Q = 0,02m^3/s)

v [m/s]	3	5	10	15
S_S [m^2]	**8,55**	**6,37**	4,28	3,39
S_C [cm^2]	**66,7**	**40,0**	20,0	13,3
b [cm]	**70**	**45**	33	22
h [cm]	**0,95**	**0,89**	0,61	0,61
2(b+h) [m]	**1,42**	**0,92**	0,67	0,45
D_e [m]	**0,0187**	**0,0174**	0,0119	0,0117
L [m]	**12,22**	**14,16**	12,96	15,39
Re	**3758**	**5811**	7935	11796
λ	**0,040**	**0,036**	0,034	0,030
ξ	**26**	**29**	36	40
Z [kg/m^7]	**354 691**	**1 103 877**	5 474 613	13 366 567
p [Pa]	**142**	**442**	2 190	5 347
p.Q [W]	**2,84**	**8,83**	43,80	106,93
[W] (η=0,3)	**9,46**	**29,44**	145,99	356,44

The basic calculations results, as they have been described above for various air speeds are briefly summarized in Table V.

5. Simple sample

There are also other influences increasing the aerodynamic resistance called local resistances, like the flow bends, the cross-section reductions or the shape of the duct input/output.
Especially for **spiral wounded** configurations, where both air ducts are in spiral configuration, each 90 degrees represents ξ = 1,1.

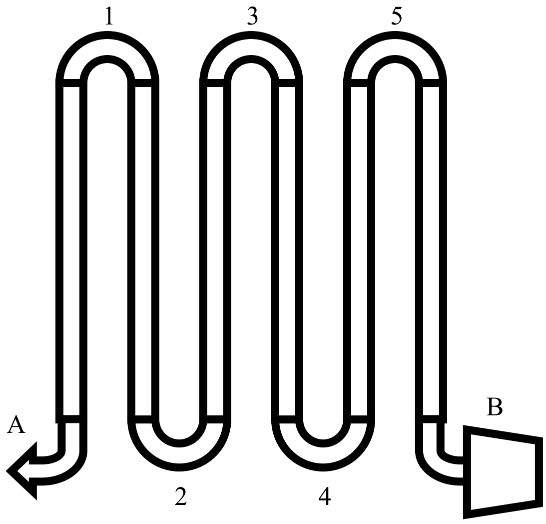

Fig.4 The schematic shape of one air duct with six linear sections with bends and input/output sections

The configuration with minimal additional resistance is in Fig.4, where the total length of each air duct is divided into the section 2m long. Such system can be installed in the thin box with the height 2,2m, which fits into the room 2,4m high.

Table VI. – The aerodynamic resistances calculations for configuration from Fig.4 (the air flow Q = 0,02m^3/s)

v [m/s]	**3**	**5**	NOTES:
b [cm]	70	45	
h [cm]	0,95	0,89	
L [m]	12,22	14,16	
ξ	26	29	friction
Z [kg/m^7]	354 691	1 103 877	
ξ	13,2	15,4	bending
ξ	2	2	in/out
ξ	**41,2**	**46,4**	**TOTAL**
	158	**160**	**+ %**
p [Pa]	**225**/(142)	**707**/(442)	All/friction
p.Q [W]	**4,5**/(2,84)	**14,13**/(8,83)	
[W] (η=0,3)	**15**/(9,46)	**47,10**/(29,44)	

The schematic shape of one 12m air duct consists from six linear sections with five 180° bends (1 – 5) and two 90° bends at input (B) and output (A).
The local resistance coefficient ξ can be found in Table VI separately for bending and for input. There is also the value of the increase ratio in this table and the increased values of pressure and ventilation power. The data are given for two air speed alternatives from Table V.
The structure can cause due to design some additional resistances and it can be said still now, that only the version with the air speed 3m/s can give good efficiency, the more precise analysis follows.

6. Ventilators

The blower (or ventilator) must create sufficient pressure to push adequate flux through the resistance that is described in above chapters. The "Ohm´s law" for the hydraulic circuits differs from the electric or thermal circuits, because here it is not linear and the pressure p [Pa] grows with quadrate of flow Q [m^3/s]:

$$p = Z\,Q^{2} \tag{12}$$

The mechanical power of ventilator output is calculated like in the electrical circuits:

$$P = p\,Q \tag{13}$$

The power input on the shaft from driving motor is, due typical efficiency about 30%, more than three times higher as is in the last row in Table V .and Table VI. The calculations do not include the influence of filters and other negative effects.
The recuperating unit must have two ventilators minimally, because each channel must be blown separately, and there is also the problem of keeping minimal overpressure inside of the building, to avoid thi input of cold air by parallel ways. This problem needs the control of speed for at least one ventilator, but this is not the topic of presented paper.
To reduce the price of the device, there is recommended to choose suitable ventilators in the wide offer of cheap ventilators for PC. The example can be the bulk ventilator SUNON with parameters: size 120x120x38mm, power input 19,20W, 12V/1600mA, speed 4200 RPM, which is for 15Euro only. DC ventilator can be easily controlled.

7. Efficiency

The ventilation recuperation works similarly as the heat-pump receiving free energy thanks to the energy which is necessary for mechanical drive of pump or ventilators in this case. The efficiency is here described by another formula comparing the received thermal energy with the energy consumed by device (typically electric energy from plug). This ratio is called COP (coefficient of performance) and can be written by formula:

$$COP = E_{HEATH} / E_{ELECTRIC} \tag{14}$$

The thermal energy production is from Table II and the consumption of the ventilators drive is from Table V and Table VI, the power source losses are estimated as is in Table VII., where the COP can be compared for various temperatures and both calculated models with low air speed.

Table VII. – The COP calculations for various temperatures and air speeds (the air flow Q = 0,02m^3/s)

	Oct, Apr	Nov, Mar	Dec, Feb	Jan
$\Delta\vartheta_{IE}$ [K]	**12**	**17**	**21**	**27**
P[kW]	**0,288**	**0,408**	**0,504**	**0,648**
	2 x 15W + 5W			
COP	**8,23**	**11,66**	**14,40**	**18,51**
	2 x 47W + 6W			
COP	**2,88**	**4,08**	**5,04**	**6,48**

8. Conclusions

The basic calculations give good survey in the internal processes in the heat exchanger and they are sufficient for practical realisation of working model.
Presented simplified analysis does not include the influence of latent heat in water vapours in warm air from inside, which can have only positive impact.
To save the ventilators drive consumption the air speed in counter flow channels should not exceed the maximal value v = 5m/s.
The spiral design increases the ventilators power.
The cheap material of separators and simple configuration, as well as cheap ventilators with low power consumption from mass production for PC power sources, can reduce the price of device important way.
The mass production of presented device can save much energy and moreover improve the climate in living rooms with positive health effects on the population.

Acknowledgement

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