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Longitudinal flow spiral recuperators in building ventilation systems

Mariusz Adamski *

Technical University of Białystok, Department of Heat Engineering, ul. Wiejska 45E, PL-15-351 Białystok, Poland

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ABSTRACT

Capital expenditure and exploitation costs of ventilation systems with longitudinal counterflow spiral recuperators are minimized taking into consideration the results of measurements and calculations together with the well-known effectiveness–NTU method.

It was estimated that ventilation system with longitudinal flow spiral recuperators refunds the capital expenditure within 2 or 3 years. Due to their advantages, spiral recuperators with the longitudinal counter current flow should be widely utilized for ventilation systems in winter and cool recuperation in summer.

1. Introduction

The increasing emphasis to use energy efficiently motives to analyze the performance and operational strategies of heating, ventilating and air conditioning (HVAC) systems in buildings. Improvements in efficiency of HVAC systems could be instrumental in maintaining existing power-plant generation capacity and avoiding further dependency on fuels. HVAC systems in commercial, industrial and residential buildings consume approximately one-half of the total energy in these buildings [1].

The European building sector is responsible for about 40% of the total primary energy consumption [2]. Heat recovery equipments find an extensive literature (for instance [3–7]).

A survey of the previous work related to the HVAC systems indicates that efforts have been made in computer simulations and experimental works [8–11].

The aim of heat recovery ventilation is to provide fresh air in the way in which thermal comfort as well as energy saving are maintained, using a recuperator with heat recovery from removed air. In particular, heat recovery should be used in buildings of public utility (banks, offices, cinemas, etc.), gastronomic institutions, swimming-pools and water parks, halls and sport objects, hospitals and clinics, industrial institutions and halls, shops, market-halls and supermarkets, in one-family and multifamily buildings.

The ventilation stations with heat recovery consist of:

- the longitudinal spiral recuperator;
- two ventilators;
- two filters;
- electric switch-board and controller;

and supplementary:

- the reheater of air;
- the cooler of air;
- bedewing cabins and humidifiers;
- the silencers of noise;
- the recirculation of air;
- the by-pass of heat exchanger.

Heat recovery from the exhaust air is a simple process.

The main cost items are investment and running costs. As far as the running costs are concerned, they are essentially two: the cost of heating energy (here assumed the district heating system), or alternatively electrical energy for cooling equipment and the cost of electrical energy for fans of heat recovery system.

Investment costs and year round savings can be evaluated during the useful life of the heat recovery system. Two methods were utilized: the present worth (PW) and the payback period. The former is a reliable evaluation procedure for alternative investments, whereas the latter is a rough one, but well understood by engineers.

Longitudinal flow spiral-tube heat exchanger is made of metal sheets, which are wound at constant intervals between subsequent windings [12]. In comparison with cross-flow ventilation heat exchangers, they obtain greater efficiency ε for the same value of the parameter NTU. Furthermore, longitudinal counterflow spiral recuperators have more uniform thermal field in each transverse

* Tel.: +48 85 746 96 33; fax: +48 85 746 95 59.
E-mail address: adamski@pb.bialystok.pl.

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Nomenclature

a_1, a_2, a_3, a_4, n coefficients
A heat transfer area (m^2)
A_C area of the duct cross-section (m^2)
c specific heat at constant pressure (J/(kg K))
c_h, c_c, c_e unit cost of heat energy, of cool energy, of electrical energy, respectively (zł/Wh)
$C = \dot{m}c = \rho A_C v c$ heat capacity flow rate (W/K)
D, L, L_o diameter, length, length of the tested exchanger, respectively (m)
D_H hydraulic diameter of channel (m)
DD_C degree-days for cooling (°C day)
DD_H degree-days for heating (°C day)
$\dot{E}$ wasted energy on pressure drops in channels of the recuperator (W)
j the month number
K cost (zł)
ΔK_W additional costs considering pressure drops in channels of the recuperator (zł)
$Ld(j)$ the number of heating days in the j-month of the year
N_D number of exploitation days
NTU number of transfer units
Δp pressure drops (Pa)
PP payback period (years)
$\dot{Q}$ heat flow rate (W)
t temperature (°C)
$t_e(j)$ mean exterior temperature in the j-month of the year (°C)
U overall heat transfer coefficient (W/(m^2 K))
v mean velocity in channels of the longitudinal flow spiral recuperator (m/s)
$V = A_C v$ volumetric flow rate (m^3/s)
$V_h = 3600 A_C v$ volumetric flow rate (m^3/h)

Greek symbols

ε effectiveness
η fan efficiency
θ wetted perimeter of the duct cross-section (m)
ρ mean density of air in ducts (kg/m^3)
φ dimensionless parameter

Subscripts

1, 2 cooled, heated air
i, o inlet, outlet
min, max, opt minimum, maximum, optimum
outflow wasted
R, W exchanger, ventilator

sections of the air stream. As a result, they are more resistant to outdropping moisture from air-cooled stream and the effect of frosting practically does not occur. In order to drain condensed water vapour effectively, they should be installed almost horizontally or vertically so that the condensate flows to the waste pipe.

Although there are many previous studies on optimum heat exchanger size [11,13–26], all of these are not directly related to the basic idea of the present study.

2. Formulation of the problem

The wasted energy rate in ventilation systems with heat recovery (Figs. 1 and 2) is as follows:

$$\dot{Q}_{outflow} + \frac{1}{\eta}(\dot{E}_1 + \dot{E}_2) \tag{1}$$

or in dimensionless form:

$$N_E = \frac{\dot{Q}_{outflow} + (1/\eta)(\dot{E}_1 + \dot{E}_2)}{\dot{Q}_{max}}. \tag{2}$$

For the ventilation systems without heat recovery $\dot{Q}_{outflow} = \dot{Q}_{max}$, $\dot{E}_1 = \dot{E}_2 = 0$ and we obtain $N_E = 1$. For the ventilation systems with an ideal heat recuperator $\dot{Q}_{outflow} = 0$, $\dot{E}_1 = \dot{E}_2 = 0$ and $N_E = 0$. Moreover, for the most frequently used cross-flow recuperators could be taken $\dot{Q}_{outflow} \le 0.5\dot{Q}_{max}$, $(1/\eta)(\dot{E}_1 + \dot{E}_2) \le 0.25\dot{Q}_{max}$ so $N_E \le 0.75$.

The energy losses rate (1) has got to complement to keep appropriate thermal conditions in the ventilated zone. The cost rate of the wasted energy rate for the heat recovery is as follows:

$$c_h\dot{Q}_{outflow} + \frac{c_e}{\eta}(\dot{E}_1 + \dot{E}_2) \tag{3}$$

and for the chill recovery:

$$c_c\dot{Q}_{outflow} + \frac{c_e}{\eta}(\dot{E}_1 + \dot{E}_2). \tag{4}$$

The above functions ((3) and (4)) could be normalized to the form:

$$N_h = \frac{c_h\dot{Q}_{outflow} + (c_e/\eta)(\dot{E}_1 + \dot{E}_2)}{c_h\dot{Q}_{max}} \tag{5}$$

or

$$N_c = \frac{c_c\dot{Q}_{outflow} + (c_e/\eta)(\dot{E}_1 + \dot{E}_2)}{c_c\dot{Q}_{max}}. \tag{6}$$

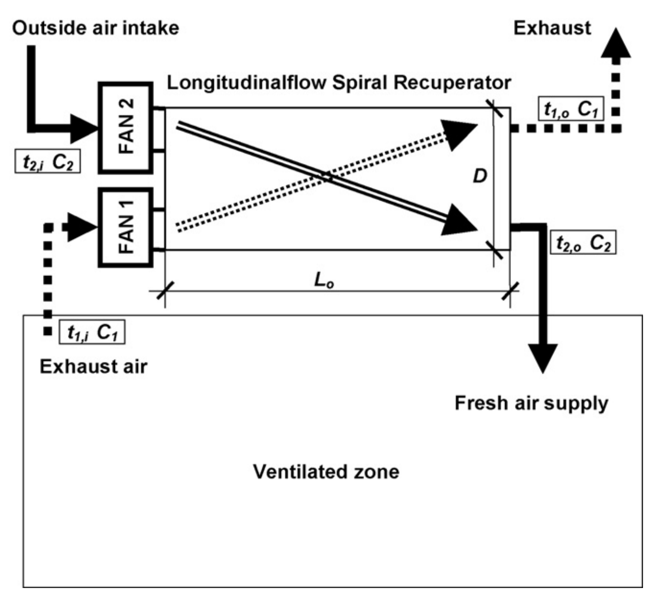

Fig. 1. Scheme of heat recovery system with the longitudinal flow spiral recuperator.