INFRASTRUKTURA I EKOLOGIA TERENÓW WIEJSKICH
INFRASTRUCTURE AND EKOLOGY OF RURAL AREAS

Nr 3/2007, POLSKA AKADEMIA NAUK, Oddział w Krakowie, s. 149–159
Komisja Technicznej Infrastruktury Wsi

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HEAT RECUPERATION FROM EXHAUST AIR IN A SPORTS HALL WITH SWIMMING POOL

Summary

This paper deals with the determination of several efficiency types of a cross-current recovery exchanger which is a part of the air venting system in the swimming pool hall on the premises of the Czech University of Life Sciences (CULS) in Prague.

The product is a cross-current plate exchanger with a heat-exchanging surface of antirust aluminium. According to the manufacturer, the exchanger is fit for temperatures common in air ventilation systems. The air is forced in by fans at a flow quantity of 16,000 m^3, maximum speed 2000 rpm, electric motor output 7.5 kW, filters for the air coming from the outside – grade G4, pressure loss from 42 to 200 Pa, filters for the air coming from the inside – grade G3, pressure loss from 46 to 200 Pa.

The results presented are derived from in-process measurements taken on 31 January 2007 and 7 February 2007, from 9.15 to 11.30 on both days.

Air temperature and air humidity were measured with 9636-51B-type sensors by Ahlborn, connected to the AHLBORN ALMEMO 5990-2 centre. These sensors were placed into each of the four input/output channels, very close to the exchanger itself (Fig. 2). The data measured were saved in the centre memory every minute.

Figures 3, 4, 5 and 6 show the temperature and humidity curves at the exchanger inlets and outlets on measurement days as well as outside air temperature (t_e) and outside air relative humidity (φ_e) captured by the met station on the CULS premises.

Table 2 shows efficiency ranges calculated according to relations (1), (3), (4), (5), and (6) for air parameters ascertained at exchanger inlets and outlets on 31 January and 7 February 2007 and the calculated flow rates (Table 1).

The difference between the outside temperature t_e and outside humidity φ_e values taken by the met station and the temperature t_{e1} and humidity φ_{e1} values measured at the recuperator inlet can be explained as resulting from the air being drawn in from the premises affected by the building and 8-m air piping situated in the building's interior. Heat transmission to the surrounding air occurs despite the mineral wool heat insulation applied to the air piping.

The slight increase of thermal efficiency observed on 7 February 2007 resulted from throttling down the recuperator feed air inlet flaps. Reducing the heated air discharge volume (see Table 1) resulted in a greater temperature difference $t_{e2} - t_{e1}$.

Energetic efficiency η is lower than thermal efficiency η_t because equation 3 takes into account the effect of condensed vapours in the cooled waste air.

According to the manufacturers, the efficiency of top-class exchangers exceeds 70 %. This value might suggest that almost all the air energy available in the given space is utilised. Closer examination reveals that what is presented is thermal efficiency, which is always higher than other kinds of efficiency (see Table 2). Low exergetic efficiency is a sign that there still is a potential in terms of transmission of recovered air utilised energy (exergy).

Key words: plate exchanger, energetic efficiency, heat energy recovery

INTRODUCTION

An option how fossil fuels can be saved is the use of vent air secondary heat. Deployment of recovery exchangers to that end appears an environmental-friendly way, as once mounted, such exchangers only increase the drag inside the air ducting pipes. They are not meant to substitute one fuel with another one, but to utilise energy that has been derived from a fossil fuel at a different time, or to utilise waste energy generated in biological processes or other energy transformations brought about by human activity. As prices of fossil fuels are expected to rise and the development of low-energy houses in urban and extra-urban areas to expand, so there may also be expected an increase in the use of recovering vent air secondary heat. The same reasons will lead to resuming the research and installations of recovery exchangers in agriculture, and in meat production in particular, a field of business with a high level of secondary heat generation. Practical use of such projects has been proved by tests carried out in the 1980s and 1990s [Adamovsky 1996].

Recovery exchangers should be able to add, to the maximum possible extent, to the vent air the energy in the waste air (ducted out of the premises ventilated). The exergetic efficiency of recovery exchangers has been calculated by Adamovsky [2000].

Comprehensive papers on exergetic analysis of heat systems were published by Bejan [1996; 2002], discussing theoretic foundations and their implications for heat exchangers and other heat systems.

This paper deals with the determination of several efficiency types of a cross-current recovery exchanger which is a part of the air venting system in the swimming pool hall on the premises of the Czech University of Life Sciences (CULS) in Prague.

MATERIAL AND METHODS

The exchanger assessed is mounted on the sporting grounds of CULS in Prague that also include a patio building of the physical education department (Fig. 1). This building dates back to the 1960s, with a total built-in area of 2,605 m^2 (gyms 736 m^2; swimming pool hall 25 x 12m, 633 m^2; locker rooms 390 m^2; and corridors and facilities 846 m^2).

The halls housing the gyms and the swimming pool are made as a precast ferroconcrete structure. Both have monolithic peripheral support pillars founded on concrete bases. Foam silicate cladding 1.5 m thick provides the halls' heat insulation. The gable walls are reinforced with monolithic ferroconcrete pillars. The roof is made in ferroconcrete frame girders with roofing panels on them. Partition walls are made in cavity light-weight bricks 10 and 15 cm thick. The ferroconcrete structure backing masonry is made in cinder blocks – locker rooms in light-weight bricks and basement premises in classic-format bricks.

Some parts of the building have basement premises, where technical facilities are situated. The building has been modernised over time (1992 – changing over to a different heat source; 1999 and 2000 – air handling system reconstruction; 2000 – windows replaced; 2006 – heat insulation for the roof; and other projects). As the reconstruction work has prevailingly aimed at reducing the building's energy intensity, recovery exchangers were mounted as part of the swimming pool (1999) and gyms (2000) air handling systems reconstruction.

The air ventilation system consists of three independent parts (swimming pool, locker rooms, and gyms). Heat recovery units have been mounted only in the first two parts. The recovery unit assessed is part of the swimming pool system. It is a product of AL-KO Luffttechnik, more specifically of the engineering plant AL-KO Therm Gmbh in Frankfurt. The units were mounted in place by a Czech firm.

The product is a cross-current plate exchanger with a heat-exchanging surface of antirust aluminium. According to the manufacturer, the exchanger is fit for temperatures common in air ventilation systems. The air is forced in by fans at a flow quantity of 16,000 m^3, maximum speed 2000 rpm, electric motor output 7.5 kW, filters for the air coming from the outside – grade G4, pressure loss from 42 to 200 Pa, filters for the air coming from the inside – grade G3, pressure loss from 46 to 200 Pa.

The ducting of the outside air in and out of the exchanger is provided by air pipes opening in the 'English courtyard' along the perimeter of the building. A specialised firm provided service of the exchanger throughout the working period. The results presented are derived from in-process measurements taken on 31 January 2007 and 7 February 2007, from 9.15 to 11.30 on both days.

Air temperature and air humidity were measured with 9636-51B-type sensors by Ahlborn, connected to the AHLBORN ALMEMO 5990-2 centre. These sensors were placed into each of the four input/output channels, very close to the exchanger itself (Fig. 2). The data measured were saved in the centre memory every minute.

Figure 1. The halls housing the gyms and the swimming pool

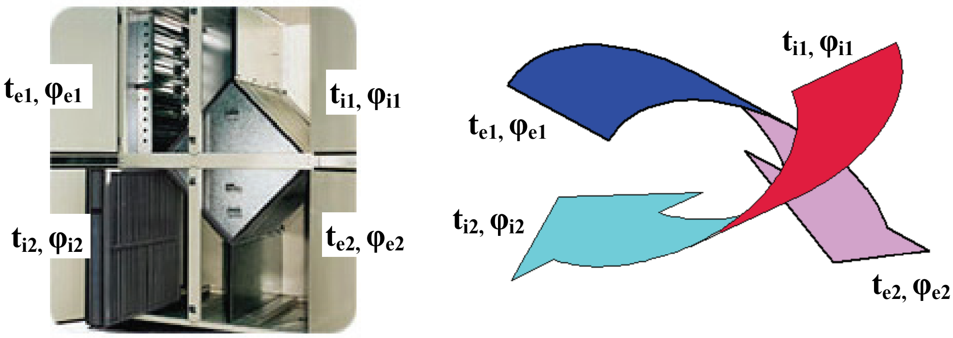

Figure 2. Points of measurement in cross-current exchanger

t_{e1} [°C] – feed ventilation air temperature before exchanger
t_{e2} [°C] – feed ventilation air temperature after exchanger
φ_{e1} [%] – feed ventilation air relative humidity before exchanger
φ_{e2} [%] – feed ventilation air relative humidity after exchanger
t_{i1} [°C] – waste air temperature before exchanger
t_{i2} [°C] – waste air temperature after exchanger
φ_{i1} [%] – waste air relative humidity before exchanger
φ_{i2} [%] – waste air relative humidity after exchanger

In order to ensure the air flow rate through the exchanger, measurements were taken of the air flow rate in air piping, dimensions 100 cm x 63 cm (waste piping before the exchanger) and 80 cm x 80 cm (feed piping after the exchanger). Measurements were taken using a Prandtl dynamic velocity probe connected to the AHLBORN ALMENO 2295-6 evaluation device. As recommended by the measuring equipment manufacturer, measurements were taken in a grid of 3 x 5 points (the waste piping) and that of 3 x 4 points (the feeding piping) in one section of both air ducts. The velocity data obtained were used to compute the volume flow rate in the respective section part and the final flow rate was established by averaging those intermediate values. Velocity measurements with the recovery exchanger inlet flaps at the incoming air side angled (throttled down) at 45° were taken 7 February 2007.

The weather conditions data shown were taken by the CULS weather station in Prague http://meteostanice.agrobiologie.cz/.

Manufacturers of recovery exchangers present thermal efficiency as one of the quality criteria (http://www.atrea.cz/?page=rkup_intro_cz); thermal efficiency can be expressed as follows:

$$\eta_t = \frac{t_{e2} - t_{e1}}{t_{i1} - t_{e1}}. \tag{1}$$

η_t [-] – thermal efficiency
t_{e1} [°C] – feed ventilation air temperature before exchanger
t_{e2} [°C] – feed ventilation air temperature after exchanger
t_{i1} [°C] – waste air temperature before exchanger

Assuming that secondary heat utilisation efficiency η is generally given as the actual obtained (recovered) heat output $\dot{Q}_e$ divided by the total heat output $\dot{Q}_{i,\max}$ to be obtained from the cooled air, it is possible to construct the following relation while considering the different amounts of feed and waste air:

$$\eta = \frac{\dot{Q}_e}{\dot{Q}_{i,\max}} = \frac{\dot{m}_{sv,e} \cdot (i_{e2} - i_{e1})}{\dot{m}_{sv,i} \cdot \left[c_{p,sv} \cdot (t_{i1} - t_{e1}) + (x_{i1} - x''_{e1}) \cdot l_{23}\right]} \tag{2}$$

and the following relation upon substituting for dry air mass flow rates:

$$\eta = \frac{\dfrac{\dot{V}_{e2} \cdot \rho_{vv,e2}}{1 + x_{e2}} \cdot (i_{e2} - i_{e1})}{\dfrac{\dot{V}_{i1} \cdot \rho_{vv,i1}}{1 + x_{i1}} \cdot \left[c_{p,sv} \cdot (t_{i1} - t_{e1}) + (x_{i1} - x''_{e1}) \cdot l_{23}\right]} \tag{3}$$

η [-] – recovery exchanger energetic efficiency

$\dot{Q}_e$ [W] – thermal output obtained

$\dot{Q}_{i,max}$ [W] – maximum obtainable thermal output

$\dot{m}_{sv,e}$ [kg.s^{-1}] – feed flow dry air mass flow rate

$\dot{m}_{sv,i}$ [kg.s^{-1}] – waste flow dry air mass flow rate

$\dot{V}_{e2}$ [m^3.s^{-1}] – feed air volume flow rate at exchanger outlet

$\dot{V}_{i1}$ [m^3.s^{-1}] – waste air volume flow rate at exchanger inlet

$\rho_{vv,e2}$ [kg. m^{-3}] – feed moist air density at exchanger outlet

$\rho_{vv,i1}$ [kg. m^{-3}] – waste moist air density at exchanger inlet

t_{e1} [^{o}C] – feed ventilation air temperature before exchanger

t_{e2} [^{o}C] – feed ventilation air temperature after exchanger

t_{i1} [^{o}C] – waste air temperature before exchanger

x_{e2} [kg. kg $_{sv}$$^{-1}$] – feed air specific humidity after exchanger

x_{i1} [kg. kg $_{sv}$$^{-1}$] – waste air specific humidity before exchanger

x''_{e1} [kg. kg $_{sv}$$^{-1}$] – specific air humidity at temperature t_{e1} and relative humidity φ = 100%

i_{e1} [J. kg $_{sv}$$^{-1}$] – feed air specific enthalpy before exchanger

i_{e2} [J. kg $_{sv}$$^{-1}$] – feed air specific enthalpy after exchanger

$c_{p,sv}$ [J. kg^{-1}.K^{-1}] – dry air specific thermal capacity at constant pressure

l_{23} [J. kg^{-1}] – evaporated water specific latent heat

The variable η can be seen as the recovery exchanger energetic efficiency. The difference $(x_{i1} - x''_{e1})$ is considered only in relation (2) where temperature t_{e1} is lower than the temperature of the dew point of the waste air incoming into the recuperator. This takes into account the possible energy gain from vapour condensation. The denominator (2) disregards the lower heat output, which is several orders lower and can be obtained by cooling down the vapours and condensed water.

For other relations establishing exchanger exergetic efficiencies see Adamovsky [2000].

The following relation is generally mentioned as that of exergetic efficiency of heat transfer from waste air in the heated air $\eta_{ex,p}$ [-] :

$$\eta_{ex,p} = \frac{\dot{E}_{e2} - \dot{E}_{e1}}{\dot{E}_{i1} - \dot{E}_{i2}} \quad . \tag{4}$$

The following relation is generally mentioned as that of exergetic efficiency of waste air heat utilisation $\eta_{ex,i}$ [-]:

$$\eta_{ex,i} = \frac{\dot{E}_{i1} - \dot{E}_{i2}}{\dot{E}_{i1}} = 1 - \frac{\dot{E}_{i2}}{\dot{E}_{i1}}, \tag{5}$$

and for total exergetic efficiency $\eta_{ex,c}$ [-]:

$$\eta_{ex,c} = \eta_{ex,p} . \eta_{ex,i} = \frac{\dot{E}_{e2} - \dot{E}_{e1}}{\dot{E}_{i1}}, \tag{6}$$

where:

$$\dot{E}_{\text{var}} = \frac{\dot{V}_{\text{var}} \cdot \rho_{vv,\text{var}}}{\left(1 + x_{\text{var}}\right)} \left(i_{\text{var}} - i_{e1}\right) \cdot \left(1 - \frac{T_{e1}}{T_{\text{var}}}\right) \tag{7}$$

The var index represents indices $e1$, $e2$, $i1$, and $i2$. Substituting the feed air parameters before exchanger in state $e1$ in equation (7) gives the following relation:

$$\dot{E}_{e1} = \frac{\dot{V}_{e1} \cdot \rho_{vv,e1}}{\left(1 + x_{e1}\right)} \left(i_{e1} - i_{e1}\right) \cdot \left(1 - \frac{T_{e1}}{T_{e1}}\right) = 0 \tag{8}$$

It follows from relation (8) that air state $e1$ can be defined as the zero state. The value obtained from equation (7) for other states is made related to this zero state, and is therefore referred to as exergetic flow rate.

$\dot{E}_{e1}$ [W] – feed air exergetic flow rate before exchanger

$\dot{E}_{e2}$ [W] – feed air exergetic flow rate after exchanger

$\dot{E}_{i1}$ [W] – waste air exergetic flow rate before exchanger

$\dot{E}_{i2}$ [W] – waste air exergetic flow rate after exchanger

T_{var} [K] – thermodynamic air temperature at points of measurement e1, e2, i1, and i2

i_{var} [J. kg $_{sv}^{-1}$] – feed air specific enthalpy at points of measurement e1, e2, i1, and i2

i_{e1} [J. kg $_{sv}^{-1}$] – feed air specific enthalpy before exchanger

i_{e2} [J. kg $_{sv}^{-1}$] – feed air specific enthalpy after exchanger

i_{i1} [J. kg $_{sv}^{-1}$] – waste air specific enthalpy before exchanger

i_{i2} [J. kg $_{sv}^{-1}$] – waste air specific enthalpy after exchanger

RESULTS

The calculated air flow rates are shown in Table 1.

Table 1. Moist air volume flow rate

Flow rate	Date of measurement	
$[m^3.h^{-1}]$	31. 1. 2007	7. 2. 2007
$\dot{V}_{e2}$	15 100	14 554
$\dot{V}_{i1}$	15 400	15 966

Figures 3, 4, 5 and 6 show the temperature and humidity curves at exchanger inlets and outlets on measurement days as well as outside air temperature (t_e) and outside air relative humidity (φ_e) captured by the met station on the CULS premises.

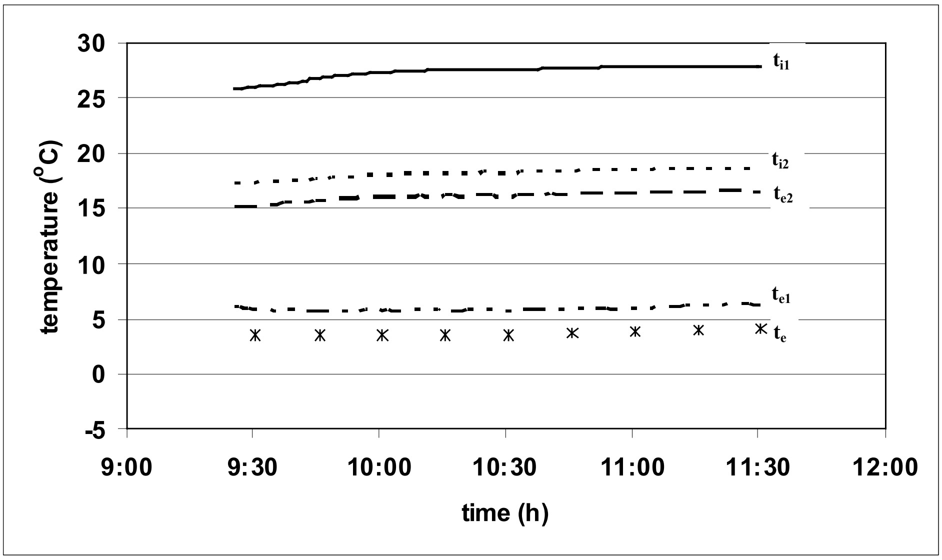

Figure 3. Temperatures at exchanger inlets and outlets on 31 January 2007

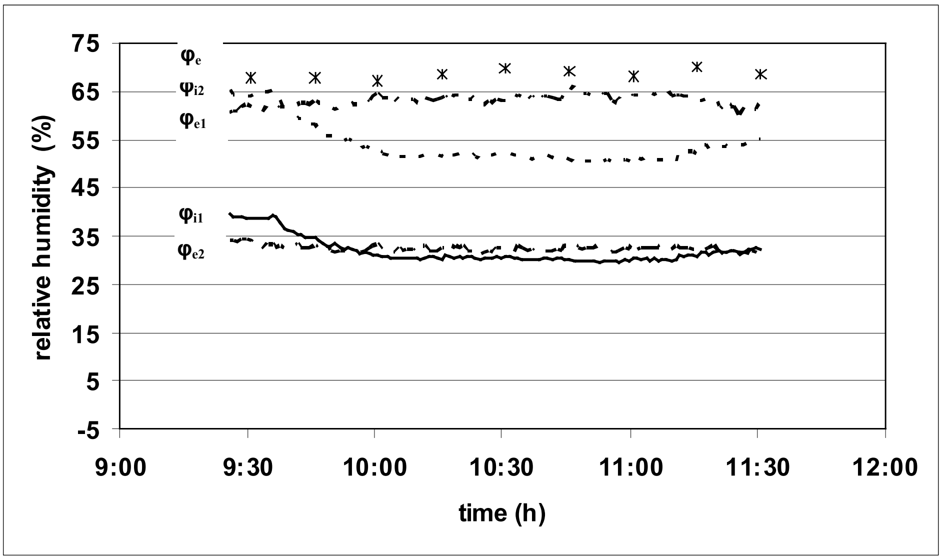

Figure 4. Relative humidity at exchanger inlets and outlets on 31 January 2007

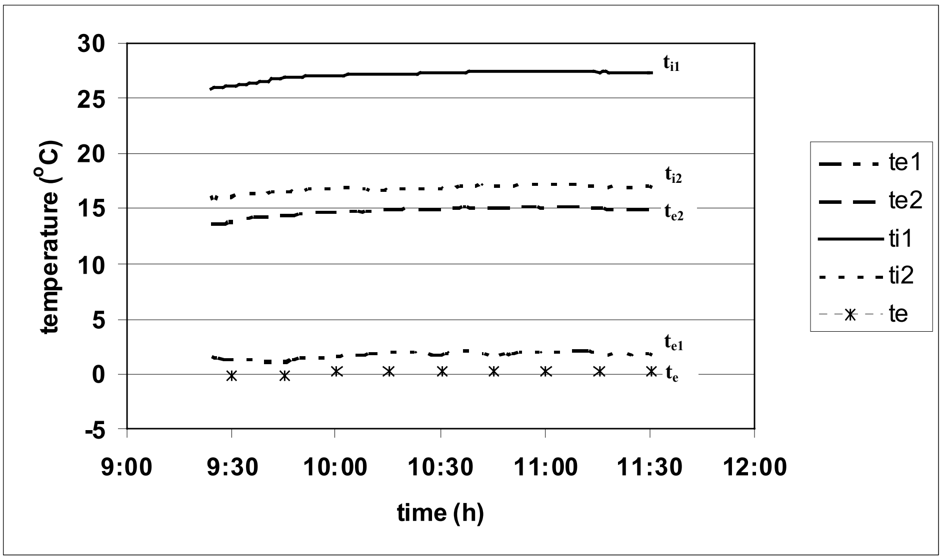

Figure 5. Temperatures at exchanger inlets and outlets on 7 February 2007

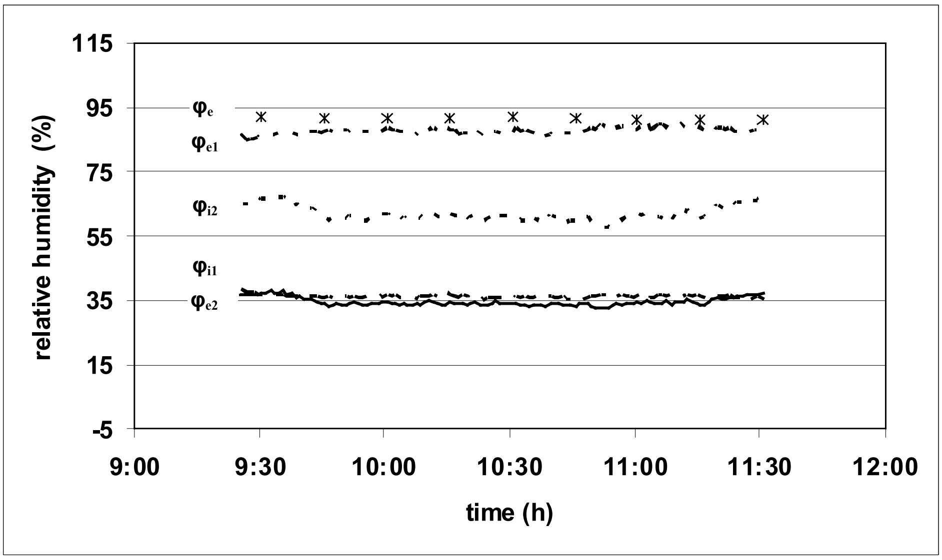

Figure 6. Relative humidity at exchanger inlets and outlets on 7 February 2007

Table 2 shows efficiency ranges calculated according to relations (1), (3), (4), (5), and (6) for air parameters measured at exchanger inlets and outlets on 31 January and 7 February 2007 and the calculated flow rates (Table 1).

Table 2. Calculated efficiency ranges for recovery exchanger performance on 31 January and 7 February 2007

Efficiency [%]	Date of Measurement	
	31. 1. 2007	7. 2. 2007
Thermal efficiency (1)	45.5 ÷ 47.9	49.5 ÷ 52.0
Energetic efficiency (3)	37.8 ÷ 46.4	36.5 ÷ 40.1
Exergetic efficiency of heat transfer from cooled air in heated air (4)	24.7 ÷ 30.7	31.4 ÷ 35.5
Cooled air heat utilisation exergetic efficiency (5)	57.6 ÷ 60.9	56.4 ÷ 59.9
Total exergetic efficiency (6)	14.3 ÷ 18.7	18.1 ÷ 20.6

Exchanger humidity condensation occurred throughout the period of measurement on 7 February 2007.

DISCUSSION AND CONCLUSION

The difference between the outside temperature t_e and outside humidity φ_e values taken by the met station and the temperature t_{e1} and humidity φ_{e1} values measured at the recuperator inlet can be explained as resulting from air being drawn in from the premises affected by the building and 8-m air piping situated in the building's interior. Heat transmission with surrounding air occurs despite the mineral wool heat insulation applied to the air piping.

A slight increase of thermal efficiency observed on 7 February 2007 resulted from throttling down the recuperator feed air inlet flaps. Reducing the heated air discharge volume (see Table 1) resulted in a greater temperature difference $t_{e2} - t_{e1}$.

Energetic efficiency η is lower than thermal efficiency η_t because equation (3) takes into account the effect of condensed vapours in the cooled waste air.

According to manufacturers, the efficiency of top-class exchangers exceeds 70% (http://www.atrea.cz/?page=rkup_intro_cz, www.rekuper.cz/rekuper-rekuperace-regulace-odpadní-vody-cnc-zpracováni-plechu-produkty-klimatizace-co-to-je.htm). This value might suggest that almost all the air energy available in the given space is utilised. Closer examination reveals that what is presented is thermal efficiency, which is always higher than other kinds of efficiency (see Table 2). Low exergetic efficiency is a sign that there still is a potential in terms of transmission of recovered air utilised energy (exergy).

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