Exergy of heat flows of the air-to-air plate heat exchanger

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ABSTRACT: Based on extensive measurements of the temperature, humidity and flow rate of the heated and cooled air in the plate heat exchanger this article analyses the influence of air inlet temperatures on both the exergy efficiency of the heat exchanger and the heat loss exergy. Furthermore, it describes the dependence between the thermal and exergy efficiency of the heat exchanger. The analysis of the tested heat exchanger indicated that the exergy efficiency of heat utilization from cooled air increases with rising inlet air temperature different, while the exergy efficiency of the heat transfer from cool to heated air decreases. In addition, the experiments confirmed the validity of the relationship between heat loss exergy and the values of air inlet temperatures.

Keywords: heat exchanger; air; efficiency; energy; exergy; anergy; heat flow; stables

The employing of recuperative heat exchangers in the utilization of the secondary ventilation air heat in stables aims at lowering the consumption of fuel for heating, and improving the stable's microclimate, along with heat comfort of the animals and protecting external structures against deterioration. Whether or not these results will be achieved depends above all on the construction of the heat exchanger and of the heat exchange surfaces.

In the technical field the energy balances are derived from the 1st thermodynamic law, which covers merely the instantaneous quantitative aspects of heat flows. In terms of energy conservation law these balances are expressed in the form of constant sums of exergy and anergy. And that is why they cannot, or can only to a very limited extent, be used to define the transformation of part of the energy (exergy) to the less valuable energy (anergy), or in other words, the so-called energy losses conditioned by the irreversibility of processes. Using the 2nd thermodynamic law we have a chance to trace the energy transformation through entropy, a quality indicator of every heat process and an accompanying symptom of process irreversibility.

The plate air-to-air heat exchanger, whose exergy analysis is in the spotlight of our interest, operates, in terms of the laws above, in irreversible changes, while the change of entropy induced by the heat flows depends on the construction and surface of the heat exchange areas.

An overview on the utilization of a physical model of entropy change in various technical fields is given in the BEJAN's paper (1996). The EGM (Entropy generation minimization) model, when applied to heat exchangers, expresses the change of total entropy S_{gen} as a sum of partial changes of entropies evoked by heat exchange $S_{gen,\Delta T}$ and fluid friction $S_{gen,\Delta p}$. The ratio of the change of entropy in the real design S_{gen} to the change of entropy in an optimal design, in which the entropy change reaches its minimum $S_{gen,min}$, is called the entropy change coefficient N_S. This method, which is well suited for finding out how close the design comes to the ideal alternative under given conditions, was published by BEJAN (1978).

The SEKULIC paper (1990) mentions an equation for entropy change as a function of temperature and pressure differences. In case of an air-to-air heat exchanger the influence of the pressure difference is later ignored, while a new coefficient, HERN (heat exchanger reversibility norm), is introduced to describe the quality of the energy transfer. This coefficient considers a reversible process as a limiting case and the real irreversible processes as less favourable alternatives. The author assumes that there exist such shapes and dimensions of heat exchanges surfaces, for which the increase of entropy will be minimal.

Another article (DE JONG et al. 1997) presents a simplified method for evaluating air-to-air heat exchangers. In principle, it is based on previous papers dealing with the generation of entropy in an irreversible process, yet the solution is simplified by calculating only the exergy of the sensible heat. According to the authors the method should optimize the heat exchange surfaces in an early stage of the design. An interesting aspect of this article is the fact that the input power of the fan is included into the total exergy balance of the heat exchanger.

A method that takes into account latent heat and condensation of water vapour from air was published by SAN and JAN (2000), who elaborated on the method

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introduced by HOLMBERG (1989). The article defines three variables that characterize the heat exchanger in a certain operation mode: exergy recovery factor r, total heat exchanger efficiency ε and exergy efficiency in terms of the 2nd thermodynamic law η_{II}. The total heat exchanger efficiency ε is a standard ratio between actually and theoretically exchanged heat flow. The exergy efficiency η_{II} is defined as a ratio of the outlet air heat flow exergy to the sum of inlet heat flow exergies. Another very interesting variable is the exergy recovery factor r, defined as a ratio of the outlet air heat flow exergy to the total exergy transformed, including mechanical exergy.

The results of the field and laboratory tests along with our faith in the persuasive capabilities of the method for evaluating heat exchanger constructions through the use of the exergy balance set us to work on this paper. Its objective is to analyse the influence of inlet air temperatures on the exergy efficiency of the heat exchanger, determine the relationship between thermal and exergy efficiency of the heat exchanger and examine the exergies of the heat flows in the exchanger.

METHOD

Theoretical analysis

The change of the specific exergy de of the substance passing through an open thermodynamic system may be expressed in the following form (RANT 1956):

$$de = dh - T_0 \,.\, ds \qquad \text{(J/kg)} \qquad (1)$$

where: dh – the change of the specific enthalpy of a substance (J/kg),
T_0 – thermodynamic temperature of the surrounding environment (K),
ds – the change of the specific entropy of a substance (J/kg/K).

In an isobaric heat transfer process the change of the specific enthalpy dh is, according to the 2nd form of the 1st thermodynamic law, equal to the specific heat that was transferred dq. Provided the heat transfer takes place in a reversible process, the change of specific entropy $ds = dq/T$ and the equation (1) may be transformed into:

$$de = dq - T_0 \cdot \frac{dq}{T} = (1 - \frac{T_0}{T}) \cdot dq \qquad \text{(J/kg)} \qquad (2)$$

Should we consider a partial fragment of the heat exchanger (Fig. 1), in which the cooled air i transfers its heat to the heated air e, then the air i will, through the heat flow dQ and following the equation (2), transfer the exergy flow as follows:

$$dE_i = (1 - \frac{T_0}{T_i}) \cdot dQ \qquad \text{(W)} \qquad (3)$$

The cooled air e, however, will absorb only the exergy flow:

$$dE_e = (1 - \frac{T_0}{T_e}) \cdot dQ \qquad \text{(W)} \qquad (4)$$

where: T_i – thermodynamic temperature of the cooled air i (K),
T_e – thermodynamic temperature of the heated air e (K).

The $dE_i - dE_e$ difference is the heat loss exergy dE, which may be expressed as:

$$dE = dE_i - dE_e = T_0 \cdot dQ \,(\frac{T_i - T_e}{T_i \cdot T_e}) \qquad \text{(W)} \qquad (5)$$

From the exergy balance of the heat flows in the heat exchanger depicted in Fig. 1 it follows that:

$$E_{e1} + E_{i1} = E_{i2} + E_{e2} + \Delta E \qquad \text{(W)} \qquad (6)$$

where: E_{e1} – exergy of the heat flow of the inlet heated air (W),
E_{i1} – exergy of the heat flow of the inlet cooled air (W),
E_{i2} – exergy of the heat flow of the outlet cooled air (W),
E_{e2} – exergy of the heat flow of the outlet heated air (W),
ΔE – heat loss exergy (W).

The exergies of the individual air heat flows at the temperature T were calculated according to the equations (3) and (4), yielding (ADAMOVSKÝ, KÁRA 1997):

$$E = \frac{V \,.\, \rho}{(1 + x)} \,(h - h_{e1}) \,.\,)1 - \frac{T_{e1}}{T}) \qquad \text{(W)} \qquad (7)$$

where: V – volume flow of the air (m^3/s),
ρ – specific density of the air (kg/m^3),
x – specific humidity of the air (kg/kg$_{s.v.}$),
h – specific enthalpy of the air (J/kg$_{s.v.}$).

From the equation (7) it follows that the exergy of the heat flow of the inlet heated air $E_{e1} = 0$, since $T = T_{e1}$.

To be able to evaluate the heat exchanger, we computed, apart from the exergies of heat flows, the following:

a) Exergy efficiency of the heat transfer from cooled to heated air $\eta_{ex,p}$.

$$\eta_{ex,p} = \frac{E_{e2} - E_{e1}}{E_{i1} - E_{i2}} \qquad (-) \qquad (8)$$

b) Exergy efficiency of the utilization of the heat from the cooled air $\eta_{ex,i}$.

$$\eta_{ex,i} = \frac{E_{i2} - E_{i1}}{E_{i1}} = 1 - \frac{E_{i2}}{E_{i1}} \qquad (-) \qquad (9)$$

c) Total exergy efficiency $\eta_{ex,c}$.

$$\eta_{ex,c} = \eta_{ex,p} \,.\, \eta_{ex,i} = \frac{E_{i2} - E_{i1}}{E_{i1}} \qquad (-) \qquad (10)$$

d) Thermal efficiency of the heat exchanger η_t at the volume flows ratio $\kappa = V_i \cdot V_e^{-1} = 1$.

$$\eta_t = \frac{t_{e2} - t_{e1}}{t_{i1} - t_{e1}} \qquad (-) \qquad (11)$$

e) Heat loss exergy ΔE from the equation (6).

Measurement methodology

The measurement was performed at the experimental site of the Technical Faculty, Czech University of Agriculture in Prague, established within the framework of the University Development Fund project No. 972/1999 entitled *Expansion of the experimental and pedagogical centre for energetically and ecologically effective devices*.

Technical parameters of the tested plate heat exchanger are provided below:

– air cross flow,
– the heat exchanger housing is made of retarded plastics,
– maximum volume flow of inlet and outlet air: 600 m^3/h,
– dimensions: height 0.61 m, depth 0.33 m, width 1.30 m,
– weight: 65 kg,
– electrical input power of the fan motors: 2 × 190 W,
– maximum number of revolutions of the fan: 1,850 rev/min,
– the sides of the exchanger at which the air is supplied are fitted with filters.

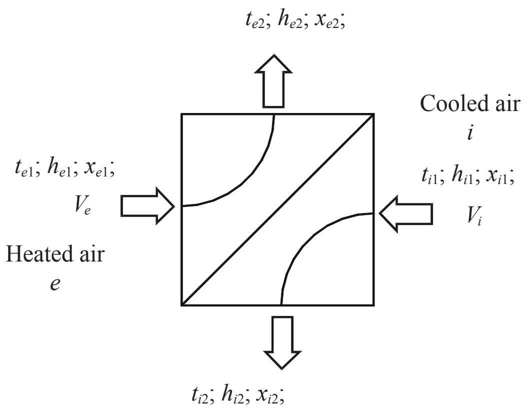

Fig. 1. Block diagram of the recuperative heat exchanger

In order to determine the exergies of the heat flows the following were measured according to the diagram depicted in Fig. 1:

– temperatures of the cooled air t_{i1}, t_{i2} (°C),
– relative humidities of the cooled air φ_{i1}, φ_{i2} (–),
– temperatures of the heated air t_{e1}, t_{e2} (°C),
– relative humidities of the heated air φ_{e1}, φ_{e2} (–),
– velocities of the cooled and heated air flow v_i, v_e (m/s).

The temperatures and relative air humidities were measured at the following volume flow ratios of the cooled and heated air: $V_i/V_e = \kappa \lessgtr 1$.

In measuring the temperatures and relative humidities we used the tested thermometers and psychrometers Ahlborn 9636, connected to the measurement centre THERM 3280-84. The air flow rates were measured with a propeller anemometer THIES CLIMA, connected to the digital measurement centre Ahlborn 2290.

Specific enthalpies, humidities and specific densities were determined according to the well-known formulas (RECKNAGEL et al. 1994).

RESULTS AND DISCUSSION

The results of the measurements and calculations of heat flow exergies and exergy efficiencies are interpreted in the charts and diagrams (Figs. 2 to 7).

From the charts in Figs. 2, 4 and 6, it is clear that the exergy efficiency of the utilization of heat from cooled air $\eta_{ex,i}$ increases with the difference $\Delta t_{i,e1}$. The behaviour of the curve $\eta_{ex,i} = f(\Delta t_{i,e1})$ conforms to Fourier's and Newton's laws of heat conduction and convection. The build-up of heat gradient $\Delta t_{i,e1}$ results in an increase in the heat flow transferred between the cooled *i* and heated air *e*, a decrease in the exergy of heat flow E_{i2} and, pursuant to the equation (9), an increase in the efficiency

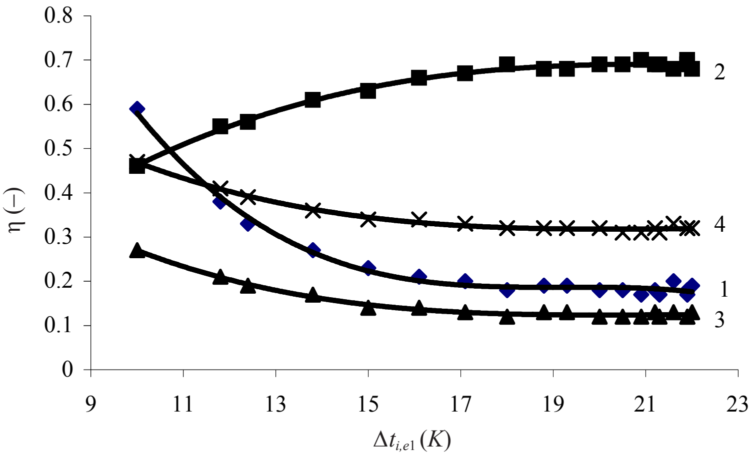

Fig. 2. The change of exergy efficiency and thermal efficiencies of the plate heat exchanger versus the difference of inlet temperatures at κ = 1.02

1. $\eta_{ex,p} = -0.0005\,\Delta t_{i,el}^3 + 0.0293\,\Delta t_{i,el}^2 - 0.5616\,\Delta t_{i,el} + 3.7779$ *[–] (*$R^2 = 0.9905$*)*
2. $\eta_{ex,i} = 0.00007\,\Delta t_{i,el}^3 - 0.0058\,\Delta t_{i,el}^2 + 0.1443\,\Delta t_{i,el} - 0.4776$ *[–] (*$R^2 = 0.9904$*)*
3. $\eta_{ex,c} = -0.0001\,\Delta t_{i,el}^3 + 0.0069\,\Delta t_{i,el}^2 - 0.1448\,\Delta t_{i,el} + 1.1396$ *[–] (*$R^2 = 0.9855$*)*
4. $\eta_t = -0.00009\,\Delta ti_{,el}^3 + 0.0061\,\Delta t_{i,el}^2 - 0.1332\,\Delta t_{i,el} + 1.2852$ *[–] (*$R^2 = 0.9852$*)*

(R^2 – determination coefficient; *R* – correlation coefficient)

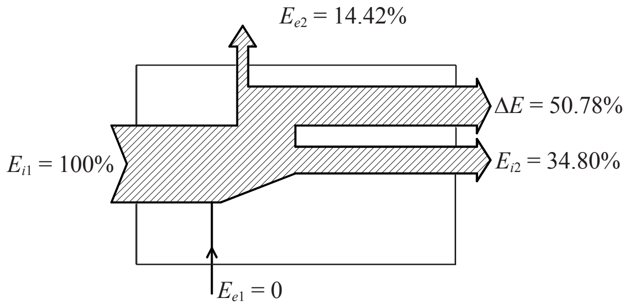

Fig. 3. The diagram of heat flow exergies in the plate heat exchanger at κ = 1.02

The total exergy efficiency of the heat exchanger $\eta_{ex,c}$, expressed in the form of (10), decreases with the increase in the temperature gradient $\Delta t_{i,e1}$. As can be seen from the behaviour of the curves as well as from the forms of the parametric equations, the decrease is lower than that of the efficiency $\eta_{ex,p}$.

The chart in Fig. 2 depicts the relationship between a thermal efficiency of the heat exchanger η_t and the temperature gradient $\Delta t_{i,e1}$ at κ = 1.08. From both the chart and the computations it follows that the equation $\eta_{ex,c} = \eta_t - 0.2$ holds for the heat exchanger in question. Assuming that $\rho_e = \rho_i$, $(1 + x_e) = (1 + x_i)$ and that the

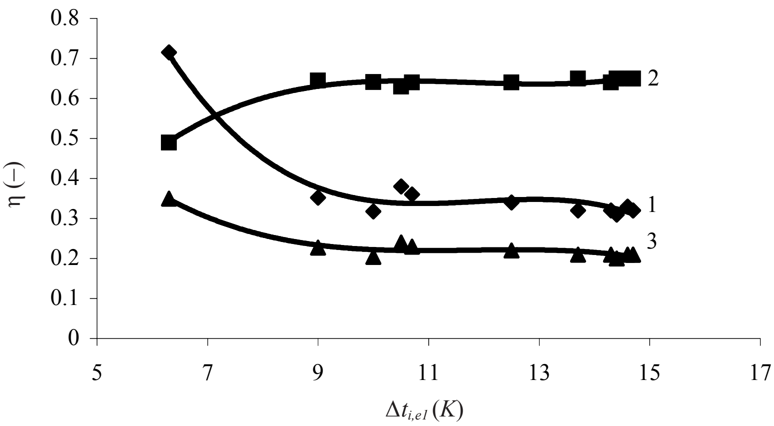

Fig. 4. The change of exergy efficiencies of the plate heat exchanger versus the difference of inlet temperatures at κ = 0.87

1. $\eta_{ex,p} = -0.0024\ \Delta t_{i,el}^3 + 0.0841\ \Delta t_{i,el}^2 - 0.9914\ \Delta t_{i,el} + 4.2086$ [–] ($R^2 = 0.9658$)
2. $\eta_{ex,i} = 0.0011\ \Delta t_{i,el}^3 - 0.0376\ \Delta t_{i,el}^2 + 0.4378\ \Delta t_{i,el} - 1.0396$ [–] ($R^2 = 0.9723$)
3. $\eta_{ex,c} = -0.0008\ \Delta t_{i,el}^3 + 0.0299\Delta\ t_{i,el}^2 - 0.3495\ \Delta t_{i,el} + 1.5757$ [–] ($R^2 = 0.9384$)

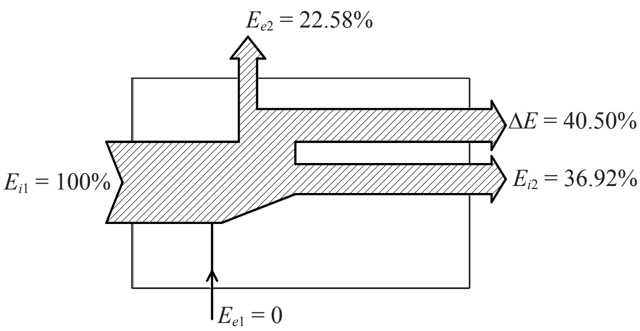

Fig. 5. The diagram of heat flow exergies in the plate heat exchanger at κ = 0.87

$\eta_{ex,i}$. The exergy of the heat flow of the outlet cooled air E_{i2} may be reduced by expanding the heat transfer area of the exchanger or by using a greater degree of turbulent flow, which is given by the air flow rate along with the shape and surface of the heat transfer areas.

With the increasing temperature gradient $\Delta t_{i,e1}$ the exergy efficiency of the heat transfer from the cooled to heated air $\eta_{ex,p}$ decreases. The pattern of the curve $\eta_{ex,p} = f(\Delta t_{i,e1})$ once again follows the laws of heat exchange as indicated above. When the exergy of the heat flow E_{i2} decreases, so does the efficiency $\eta_{ex,p}$ as corresponds to the equation (8).

specific heat capacities at constant pressure are equal, i.e. $c_{p,e} = c_{p,i}$, then the equations (7), (8) and (11) imply that the following formula describes the relationship between the total exergy efficiency $\eta_{ex,c}$ and the thermal efficiency of the heat exchanger η_t:

$$\eta_{ex,c} = \frac{1}{\kappa} \cdot \frac{T_{i1}}{T_{e2}} \cdot \eta_t^2 \qquad (-) \qquad (12)$$

The diagrams depicted in Figs. 3, 5 and 7 show the percentage distribution of the exergies of heat flows in the heat exchanger. At all values of κ the heat loss exergy ΔE accounts for the biggest portion. The measured values along with the computation results confirmed the conclusion arising from the equation (5). Not only does the heat loss exergy ΔE depend on the temperature gradient $\Delta t_{i,e1} = T_i - T_e$, but it is also influenced by the level of temperatures. At the constant temperature difference $\Delta t_{i,e1}$ the heat loss exergy is lower at high temperatures and higher at low temperatures. The diagrams also indicate that as the κ coefficient rises, so does the heat loss exergy ΔE. In addition, the increase in ΔE is closely related to the decline in the exergy of the heat flow of the outlet heated air E_{e2} as well as the exergy efficiency $\eta_{ex,p}$. These conclusions are supported by the equations (6, 8).

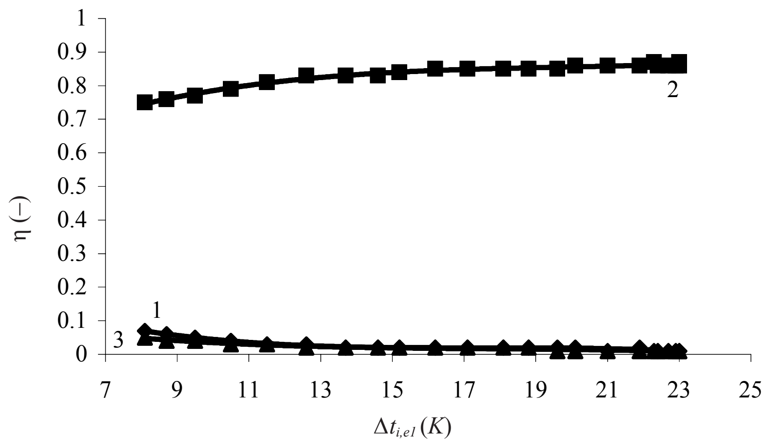

Fig. 6. The change of exergy efficiencies of the plate heat exchanger versus the difference of inlet temperatures at κ = 2.80

1. $\eta_{ex,p} = -0.00006\ \Delta t_{i,e1}^3 + 0.0033\ \Delta t_{i,e1}^2 - 0.0568\ \Delta t_{i,e1} + 0.3486$ [–] ($R^2 = 0.9790$)
2. $\eta_{ex,i} = 0.00005\ \Delta t_{i,e1}^3 - 0.0032\ \Delta t_{i,e1}^2 + 0.0644\ \Delta t_{i,e1} - 0.4036$ [–] ($R^2 = 0.9843$)
3. $\eta_{ex,c} = -0.00003\ \Delta t_{i,e1}^3 + 0.0014\ \Delta t_{i,e1}^2 - 0.0252\ \Delta t_{i,e1} + 0.1752$ [–] ($R^2 = 0.9461$)

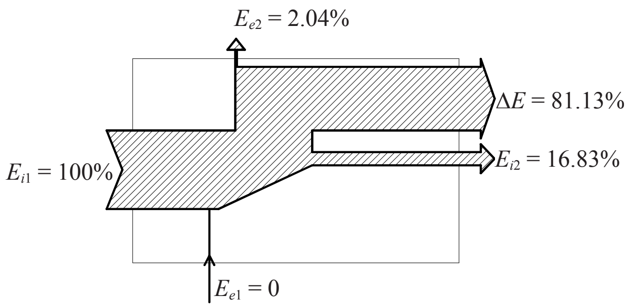

Fig. 7. The diagram of heat flow exergies in the plate heat exchanger at κ = 2.80

CONCLUSION

The exergy analysis of the tested plate heat exchanger showed and proved that:

– the exergy efficiency of the utilization of the heat from cooled air $\eta_{ex,i}$ increases with the difference of inlet air temperatures $\Delta t_{i,e1}$,
– the exergy efficiency of the heat transfer from cooled to heated air $\eta_{ex,p}$ decreases with the difference of inlet air temperatures $\Delta t_{i,e1}$,
– the relationship between the thermal and total exergy efficiency of the heat exchanger, η_t and $\eta_{ex,c}$ respectively, may be expressed in the form of the equation (12),
– at a constant temperature difference $\Delta t_{i,e1}$ the heat loss exergy ΔE is lower at high temperatures and higher at low temperatures,
– as the $\kappa = V_i/V_e^{-1}$ increases, so does the heat loss exergy ΔE.

The results of the tests and computations imply that evaluating the recuperative heat exchanger by means of exergy analysis allows us to get a detailed overview of the thermodynamic processes in action, to qualify the individual heat transfer processes and to acquire the knowledge necessary for the optimization thereof. We should add that this optimization aims not at a complete elimination of the exergy loss, but rather at the most efficient elimination in an economical sense.

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Exergie tepelných toků deskového rekuperačního výměníku vzduch–vzduch

ABSTRAKT: Na základě rozsáhlých měření teplot, vlhkostí a rychlostí proudění ohřívaného ochlazovaného vzduchu v deskovém rekuperačním výměníku jsou analyzovány vlivy vstupních teplot vzduchu na exergické účinnosti výměníku i exergie tepelných toků a naznačena závislost mezi teplotní a exergickou účinností výměníku. Exergická analýza ověřovaného výměníku ukázala, že exergická účinnost využití tepla z ochlazovaného vzduchu stoupá s rozdílem vstupních teplot a exergická účinnost přenosu tepla z ochlazovaného do ohřívaného vzduchu klesá s rozdílem vstupních teplot. Dále je experimentálně potvrzen vztah mezi exergií ztrátového tepelného toku a rozdílem vstupních teplot vzduchu.

Klíčová slova: rekuperační výměník; vzduch; účinnost; energie; exergie; anergie; tepelný tok; stáj

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