Exergy of heat flows in exchanger consisting of gravity heat pipes

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ABSTRACT: The paper deals with the analysis of the impact of inlet air temperature on the exergy efficiency and exergy of the losing heat flow and determination of the relation between the exergy and thermal efficiency in an exchanger consisting of gravity heat pipes. The assessment of heat processes quality and transformation of energy in the exchanger are also dealt with.

Keywords: energy; anergy; exergy; efficiency; heat exchanger; heat pipe; heat flow

Exergy analysis of heat systems is a method established in 1970's. Aiming at improving the efficiency of heat processes, it makes use of thermodynamic optimization, i.e. the method of minimizing the growth of entropy (EGM – entropy generation minimization). Numerous studies have dealt with this analysis since the very beginning, with the Bejan (1996, 2002) publications representing the summary focusing on theoretical fundamentals relating to heat exchangers and other thermal systems.

The term of exergy is generally used to express the utilizable energy in energy development and consumption systems; its principal advantage is that it enables a comparison of various heat flows on a common basis. Another advantage lies in the fact that it makes it possible to define the exergy loss rate in the sum of all exergy flows within the system that is proportionate to the growth of entropy in the system (Bejan 2002). The process of exergy wasting is a transformation into a less valuable energy – anergy.

These possibilities established by the 2nd law of thermodynamics enable the monitoring of the quality of heat processes in heat exchangers. With this knowledge, it is subsequently possible to define the parameters and steps leading to designing a heat exchanger with a minimum exergy transformation, i.e. minimum entropy growth and high efficiency. Most publications describe the growth of entropy in the heat exchanger as the sum of the partial change in entropy caused by heat exchange (temperature difference) and a partial change of entropy caused by the fluid matter friction (pressure drop). As regards the air-to-air heat exchangers, the partial change caused by the difference in pressures is often neglected (Sekulic, Baclic 1978; Sekulic 1990) so as to facilitate the definition of the optimum progress of heat processes.

The generation of entropy in a heat pipe is dealt with in the publication by Khalkhali et al. (1996). To optimize the processes in the heat pipe, the authors set three basic parameters conducing to the growth of entropy. The first is the difference in temperature between the evaporation and condensation sections of the tube; the second one is the temperature drop in the flow of working medium vapour during its transport between both the ends of the tube, and the third one is the impact of friction related with the flows of vapour and condensate. Based on the first and second parameters, the publication defined the optimum temperature of the environment in the condensation section of the tube, as well as the requirement for the minimum of heat transfer coefficient in the transport section of the tube. The third parameter defining the friction of the working medium in the tube implies the shortest possible length of the evaporation part of the tube and the maximum possible wick cross-sectional hydraulic diameter.

The requirements for the wick cross-sectional hydraulic diameter of the condensate flow-off area and the minimum coefficient of thermal transmission in the transport section were also confirmed by a theoretical study (Abtahi et al. 1988) on the basis of the standard balance of energy flows between the cooled and heated air currents, the coat and working medium of the tube.

The purpose of this paper is to analyze the impact of the inlet air temperatures on exergy efficiency values, exergy values of heat flows and to determine the relation between the exergy efficiency and thermal efficiency in a heat pipe heat exchanger. The paper also deals with evaluating the quality of thermal processes and exergy transformation in the heat exchanger.

METHOD

Theoretical analysis

The exergy of the mass flow of 1 kg of gas during its passage through an open thermodynamic system can be determined from the following adiabatic (isoentropic) and isothermal expansion illustrated in Fig. 1.

Gas expands from the initial state determined by the pressure p_1 and temperature T_1 to the state of the surrounding environment determined by the pressure p_0 and temperature T_0. In the case of an adiabatic expansion, the temperature drops to $T_2 = T_0$ and the pressure $p_2 > p_0$ is reached. During the subsequent reversible isothermal expansion with a simultaneous heating, the gas pressure drops to $p_3 = p_0$.

In this case, the maximally utilizable energy (exergy) is equal to the technical work $a_{t1.3}$ carried out between the initial and final states. The first law of thermodynamics implies that:

$$da_t = dq - dh \qquad \text{(J/kg)} \qquad (1)$$

where: da_t – change in specific technical work (J/kg),
dq – specific transferred heat (J/kg),
dh – change in specific enthalpy (J/kg).

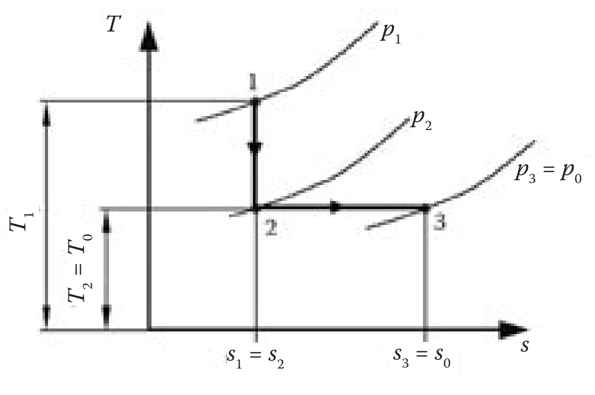

Fig. 1. Adiabatic and isoentropic gas expansion

Integrating equation (1), we obtain the following result:

$$a_{t1.3} = q_{1.3} - (h_3 - h_1) = h_1 - h_3 + q_{1.3} \qquad \text{(J/kg)} \qquad (2)$$

As 1–2 is $dq = 0$ in adiabatic expansion, the following applies to transferred heat:

$$q_{1.3} = q_{2.3} = T_0(s_3 - s_2) = -\,T_0(s_2 - s_3) \qquad \text{(J/kg)} \qquad (3)$$

Substituting the figure in equation (2), we obtain the following result:

$$a_{t1.3} = h_1 - h_3 - T_0(s_2 - s_3) \qquad \text{(J/kg)} \qquad (4)$$

This maximum technical work $a_{t1,3}$ corresponding to the reversible process between the gas state determined by $h_1 = h$ and $s_1 = s$ and the state of the environment determined by $h_3 = h_0$ and $s_3 = s_0$ represents the maximum work content of the substance, i.e. exergy e_x:

$$e_x = h - h_0 - T_0(s - s_0) \qquad \text{(J/kg)} \qquad (5)$$

The non-utilized specific energy, i.e. anergy a_n, is h_0 at T_0 and shared heat at T_0:

$$a_n = h_0 + T_0(s - s_0) \qquad \text{(J/kg)} \qquad (6)$$

According to equation (5), the change in the specific exergy of a heat flow during any process is as follows:

$$de_x = dh - T_0 ds \qquad \text{(J/kg)} \qquad (7)$$

In the event of an isobaric heat transfer, the specific enthalpy change dh is equal to the transferred heat dq. The specific entropy change is $ds = dq/T$ in case of a reversible process. The above-mentioned facts indicate that equation (7) can be put as follows:

$$de_x = dq - T_0 \times \frac{dq}{T} = \left(1 - \frac{T_0}{T}\right) dq \qquad \text{(J/kg)} \qquad (8)$$

Equation (6) can be put as follows:

$$da_n = \frac{T_0}{T}\, dq \qquad \text{(J/kg)} \qquad (9)$$

When heat is transferred between cooled air i and heated air e, heat flow dQ transmits exergy flow dE_i (10) according to equation (8). Heated air e, however, receives only the exergy flow dE_e (11):

$$dE_i = \left(1 - \frac{T_0}{T_i}\right) dQ \qquad \text{(W)} \qquad (10)$$

$$dE_e = \left(1 - \frac{T_0}{T_e}\right) dQ \qquad \text{(W)} \qquad (11)$$

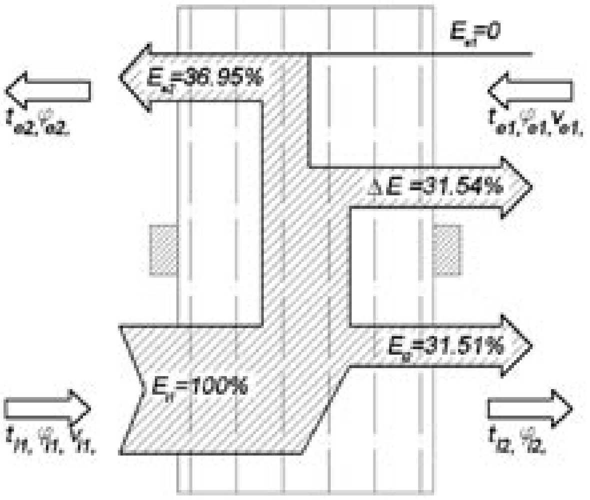

Fig. 2. Scheme of measurement and exergies of exchanger heat flows

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

The exergy of the losing heat flow dE can be expressed by the following equation:

$$dE = dE_i - dE_e = T_0 dQ \left(\frac{T_i - T_e}{T_i \times T_e}\right) \qquad \text{(J/kg)} \quad (12)$$

The balance of exergies of exchanger heat flows according to Fig. 2 implies the following equation:

$$\Delta E + E_{i2} + E_{e2} = E_{i1} + E_{e1} \qquad \text{(J/kg)} \quad (13)$$

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

The proportion between the heat flow exergy E and heat flow energy Q is referred to as the exergetic number by RANT (1956) who introduced the term of exergy (1953) for the transformable part of energy and the term of anergy (1955) for the non-transformable part of energy; other authors refer to it as the exergetic factor (WALL 1998).

Exergetic factors can be defined for heated air $\varepsilon_{ex,e}$ (14) and cooled air $\varepsilon_{ex,i}$ (15):

$$\varepsilon_{ex,e1} = \frac{E_{e1}}{Q_{e1}}\ 0(-), \qquad \varepsilon_{ex,e2} = \frac{E_{e2}}{Q_{e2}}\ (-) \qquad (14)$$

$$\varepsilon_{ex,i1} = \frac{E_{i1}}{Q_{i1}}\ (-), \qquad \varepsilon_{ex,i2} = \frac{E_{i2}}{Q_{i2}}\ (-) \qquad (15)$$

We calculated the exergies of heat flows, proceeding from equations (10) and (11) according to the following formula (ADAMOVSKÝ, KÁRA 1997; ADAMOVSKÝ, ADAMOVSKÝ 2004):

$$E = \frac{V \times \rho}{(1 + x)}(h - h_{e1}) \times \left(1 - \frac{T_{e1}}{T}\right) \qquad \text{(W)} \quad (16)$$

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

Equation (16) implies that the inlet heated air heat flow exergy $E_{e1} = 0$ as $T = T_{e1}$.

Besides heat flows exergies, the following parameters were determined for the purpose of evaluating exchangers:

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

$$\eta_{ex.p} = \frac{E_{e2} - E_{e1}}{E_{i1} - E_{i2}} \qquad (-) \quad (17)$$

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

$$\eta_{ex.i} = \frac{E_{i1} - E_{i2}}{E_{i1}} = 1 - \frac{E_{i2}}{E_{i1}} \qquad (-) \quad (18)$$

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

$$\eta_{ex.c} = \eta_{ex.p} \times \eta_{ex.i} = \frac{E_{e2} - E_{e1}}{E_{i1}} \qquad (-) \quad (19)$$

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

$$\eta_t = \frac{t_{e2} - t_{e1}}{t_{i1} - t_{e1}} \qquad (-) \quad (20)$$

e) Exergy of losing heat flow ΔE from equation (13).

The energetic efficiency of heat exchanger is generally defined by the following formula:

$$\eta = \frac{Q_R}{Q_{i1} - Q_{e1}} \qquad (-) \quad (21)$$

where: $Q_R = Q_{e2} - Q_{e1}$ – reutilized heat flow (W),
Q_{i1} – inlet cooled air heat flow (W),
Q_{e1} – inlet heated air heat flow (W),
Q_{e2} – outlet heated air heat flow (W).

Substituting figures from equations (14, 15) and (19) in equation (21), the following result is obtained:

$$\eta = \eta_{ex,c} \times \frac{\varepsilon_{ex,i1}}{\varepsilon_{ex,e2}} \qquad (-) \quad (22)$$

Measurement technique

The measurement and results evaluation were supported by Research Plan VZ03 CEZ MSM 6840770003 Development and application of computer simulation algorithms in engineering. The measurement was carried out at the experimental centre of the Technical Faculty of Czech University of Agriculture in Prague, implemented within the framework of Project No. 972/1999 of the University Development Fund.

Technical parameters of the tested HPHE exchanger are:

- finned pipe, extruded, pure aluminium;
- pipe outer diameter (excluding the fins): 25 mm;
- pipe outer diameter (including the fins): 53 mm;
- effective length of pipes: 1,450 mm;
- fin spacing: 2.5 mm;
- number of fins per 1 m of pipe length: 330;
- number of pipe rows: 6;
- number of pipes per row: 8;
- total number of pipes: 48;
- linear pitch of pipes in rows: 62 mm;
- transversal pitch of tubes: 62 mm;
- bundle width: 455 mm;
- bundle depth: 345 mm;
- front-end bundle cross-section in the inlet and outlet air chamber: 0.426 m^2,
- maximum reduction of cross sectional area of flow: 0.345.

The following values were measured to determine the exergies of the exchanger heat flows according to the scheme given in Fig. 2:

- cooled air temperature values t_{i1}, t_{i2} (°C);
- cooled air relative humidity values φ_{i1}, φ_{i2} (–);
- heated air temperature values t_{e1}, t_{e2} (°C);
- heated air relative humidity values φ_{e1}, φ_{e2} (–);
- cooled and heated air flow rates v_i, v_e (m/s).

The air temperature and relative humidity values were measured at volume flow rates of cooled and heated air of

$V_i/V_e = \kappa \cong 1$

The Ahlborn 9636 thermometers and psychrometers connected to the THERM 3280-84 recorder were used for measuring the temperature and relative humidity values. The air flow rates were measured with the THIES CLIMA propeller-type anemometer connected to the Ahlborn 2290 recorder.

The values of specific enthalpy, humidity and specific weight were determined according to known formulas (Recknagel et al. 1994).

RESULTS AND DISCUSSION

The results of measurements and calculations of heat flows exergies, exergy efficiencies and exergetic factors are illustrated by Figs. 2–4.

The chart in Fig. 3 implies that the exergy efficiency of the utilization of the heat from cooled air $\eta_{ex,i}$ increases with the increasing temperature difference $\Delta T_{i,e1} = T_{i1}-T_{e1}$. The growth $\Delta T_{i,e1} = 7.5K$ results in the increase of $\eta_{ex,i}$ by 0.17. The characteristics of the behaviour $\eta_{ex,i}= f(\Delta T_{i,e1})$ are confirmed by equations (16, 18). With the growing temperature difference $\Delta T_{i,e1}$, the exergy of the heat flow E_{i1} ($E_{e1} = 0$) increases according to equation (16). At the same time, with the growing temperature difference $\Delta T_{i,e1}$ the exergy of the heat flow E_{i2} decreases according to the laws of heat transfer by conduction and convection while the efficiency $\eta_{ex,i}$ grows in compliance with equation (18). In practice, efficiency $\eta_{ex,i}$ can be improved by increasing the area of heat-exchanging surfaces and optimizing their shapes in terms of heat transfer.

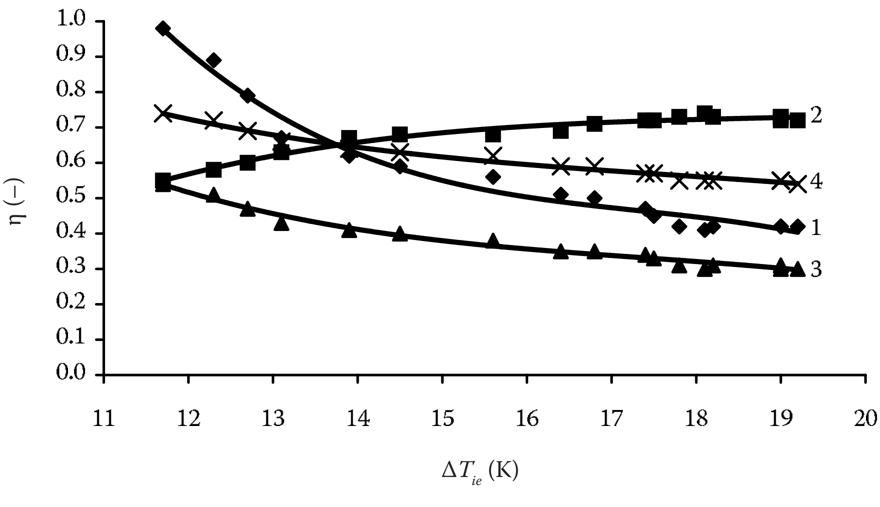

Fig. 3. The change of exergy efficiencies and thermal efficiency of the exchanger versus the difference of inlet temperatures

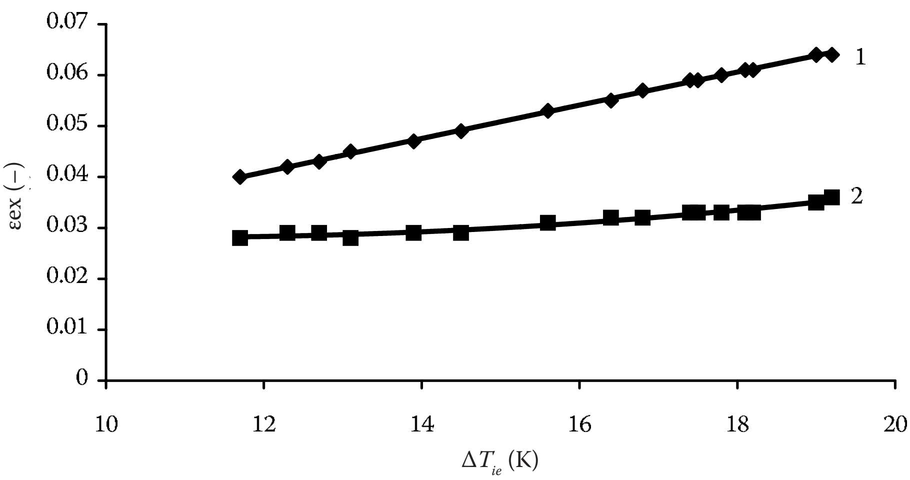

Fig. 4. The change of exergetic factors of cooled air versus the difference of inlet temperatures

The exergy efficiency of the heat transfer from cooled into heated air $\eta_{ex,p}$ decreases with the increasing temperature gradient $\Delta T_{i,e1}$. The growth $\Delta T_{i,e1} = 7.5K$ implies a drop of $\eta_{ex,p}$ by 0.56. The characteristics of the behaviour $\eta_{ex,p} = f(\Delta T_{i,e1})$ result from equations (16, 17). With increasing $\Delta T_{i,e1}$, E_{i1} grows while E_{i2} drops, and according to equation (17), $\eta_{ex,p}$ also decreases. The efficiency $\eta_{ex,p}$ is an important criterion of utilizing waste heat contained in cooled air. The $\eta_{ex,p}$ value is influenced primarily by the quantity of the losing heat flow ΔE that increases with the increasing $\Delta T_{i,e1}$. The efficiency $\eta_{ex,p}$ can be influenced by selecting a suitable working medium for the heat pipes, evaporation pressure, length of the evaporation section of the pipe and wick cross-sectional hydraulic diameter of the condensate (Khalkhali et al. 1996).

The overall exergy efficiency of the exchanger $\eta_{ex,c}$ decreases with the growing increase of the temperature difference $\Delta T_{i,e1}$. The growth $\Delta T_{i,e1} = 7.5K$ implies a decrease of $\eta_{ex,c}$ by 0.24. Within the range of $\Delta T_{i,e1} = 11–20\ K$, it reaches values of 0.30–0.54.

The thermal efficiency of the exchanger η_t drops with the growing temperature difference $\Delta T_{i,e1}$ which is significantly greater than the growth of difference in temperatures $\Delta T_e = T_{e2} - T_{e1}$. The growth $\Delta T_{i,e1} = 7.5K$ implies a drop of η_t by 0.20. Within the range of $\Delta T_{i,e1} = 11–20\ K$, it reaches values of 0.54–0.74. Estimating that at $V_i = V_e$, $(1 + x_e) = (1 + x_i)$, $c_{pe} = c_{pi}$, and $\rho_e = \rho_i$, equation (16) and (20) imply the following relation between the overall exergetic $\eta_{ex,c}$ and thermal η_t, efficiency of the exchanger:

$$\eta_{ex.c} = \frac{T_{i1}}{T_{e2}} \times \eta_t^2 \qquad (–) \quad (23)$$

At the conditions specified above, the thermal efficiency of the exchanger η according to equation (20) is equal to the energetic efficiency η according to equation (21).

The scheme in Fig. 2 illustrates the percentual distribution of the average values of the exchanger heat flows exergies calculated from 23 measurements with $\Delta T_{i,e1}$ values ranging from 10 to 20 K. The exergy of the losing heat flow ΔE represents a considerable fraction which, as proved by verifications, increases with the growth of $\Delta T_{i,e1}$ and the ratio of $V_i \times V_e^{-1}$. The trend of increasing the ΔE value is related to the decrease of $\eta_{ex,p}$ and the percentual rate of the exergies of heat flows E_{e2} and E_{i2}. The results of the verification procedure confirmed the validity of equation (12). The exergy of losing heat flow ΔE depends not only on the temperature difference $\Delta T_{i,e1}$, but also on the temperature values. With the same $\Delta T_{i,e1}$, ΔE is lower at higher temperatures and higher at lower ones.

The scheme in Fig. 4 illustrates the dependence of exergetic factors of cooled air on the inlet temperature difference. With the temperature T_{i1} growing in comparison with T_{e1}, the proportion of utilizable energy, i.e. exergy, grows, with the exergetic factor $\varepsilon_{ex,i1}$ increasing. The exergetic factor ranges from 0.040 to 0.064. This implies that when $\varepsilon_{ex,i1} = 0.04$, at $t_{i1} = 18.4°C$ and $t_{e1} = 6.7°C$, the heat flow Q_{i1} contains 4 per cent of exergy and 96 per cent of anergy. The exergetic factor $\varepsilon_{ex,i2}$ slowly grows with the growing temperature difference $\Delta T_{i,e1}$. Its values are obviously lower within the range of 0.028 to 0.036.

CONCLUSION

The analysis of the gravitational heat pipe heat exchanger proved that:

– the exergy efficiency of the utilization of the heat from cooled air $\eta_{ex,i}$ grows with the growing inlet air temperature difference $\Delta T_{i,e1}$;
– the exergy efficiency of the heat transfer from cooled into heated air $\eta_{ex,p}$ drops with the growing inlet air temperature difference $\Delta T_{i,e1}$;

– the relation between the exchanger thermal efficiency η_t and its total exergy efficiency $\eta_{ex,c}$ can be expressed with formula (23);
– the losing heat flow exergy ΔE is lower at higher temperatures and higher at lower temperatures when the temperature difference $\Delta T_{i,e1}$ is constant;
– the losing heat flow exergy ΔE grows with growing $\Delta T_{i,e1}$ and the ratio $V_i \times V_e^{-1}$;
– the values of exergetic factors of inlet cooled air $\varepsilon_{ex,i1}$ and outlet cooled air $\varepsilon_{ex,i2}$ grow with the growing $\Delta T_{i,e1}$.

The calculations of the total exergy efficiency $\eta_{ex,c}$ and thermal efficiency η_t imply a good quality of thermal processes and exergy transformation in the exchanger. These conclusions are also confirmed by low values of the exergetic factor $\varepsilon_{ex,i2}$. Significantly lower values of $\eta_{ex,c}$ and η_t were detected at the same ratio of $V_i \times V_e^{-1}$ when carrying out measurements of a plate heat exchanger (Adamovský, Adamovský 2004).

The transformation of exergy can be improved by optimizing the shapes of heat-exchanging surfaces, which applies primarily to the evaporation section of the heat pipe, and by selecting an appropriate working medium. Another way is to decrease the hydraulic friction of working media vapour and condensate. The purpose of these measures is to decrease the length of the heat pipe evaporation section and to speed up the process of thermodynamic changes in progress.

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Exergie tepelných toků výměníku z gravitačních tepelných trubic

ABSTRAKT: Článek se zabývá analýzou vlivu vstupních teplot vzduchu na exergetické účinnosti a exergii ztrátového tepelného toku a stanovením závislosti mezi exergetickou a teplotní účinností ve výměníku z gravitačních tepelných trubic a hodnocením kvality tepelných procesů a transformace exergie ve výměníku.

Klíčová slova: energie; anergie; exergie; účinnost; výměník tepla; tepelná trubice; tepelný tok

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