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MAXIMUM ENTROPY GENERATION RATE IN A HEAT EXCHANGER AT CONSTANT INLET PARAMETERS

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Abstract: The main goal of the paper is to provide a condition for which a maximum entropy generation occurs in a heat exchanger at constant inlet parameters (temperatures and mass flow rates). Knowing this condition is essential during the design of the heat exchanger as it allows designers to avoid one of its most unfavourable operating conditions in terms of thermodynamics. Entropy generation resulting from the resistance of heat-transferring fluids to flow was not taken into account. Entropy generation was analysed as a function of a heat flow rate at constant parameters at the inlet of a condenser and a counter-flow double-pipe heat exchanger. The analysis showed that for the condenser the entropy generation rate increases with the increase in the heat flow rate. The maximum entropy generation rate occurs for the maximum flow rate of the heat that can be transferred according to the definition of heat transfer effectiveness. For the counter-flow heat exchanger, the entropy generation as a function of the heat flow rate reaches maximum at constant inlet parameters (temperatures and mass flow rates). It appeared that the peak entropy generation, or the largest exergy loss, occurs when the outlet temperatures of the fluids are equal. This assertion was verified against data obtained from a simulator of the counter-flow heat exchanger for two different relations between heat capacity rates.

Keywords: maximum entropy generation rate, heat exchanger

1. INTRODUCTION

Heat exchangers are widely used in industrial applications to transfer heat from a higher to a lower temperature fluid.

Heat transfer effectiveness is commonly used to compare heat exchangers and assess their performance, it is defined as the ratio of the actual to the maximum rate of heat flow which can be transferred in the heat exchanger [1–4]:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} \cdot \tag{1}$$

The heat transfer effectiveness, which is used as an indicator in the comparison of heat exchangers, is a function of two parameters: *NTU* (number of heat transfer units) which is equal to a product of an overall heat transfer coefficient (U) and a heat transfer surface area (A), divided by the smaller fluid heat capacity rate

$$NTU = \frac{UA}{C_{\min}(C_1, C_2)}, \tag{2}$$

and the ratio of heat capacity rates:

$$C_r = \frac{C_{\min}(C_1, C_2)}{C_{\max}(C_1, C_2)}, \tag{3}$$

where the heat capacity rate is a product of the fluid mass flow rate and its specific heat at constant pressure:

$$C = c_p\, \dot{m} \cdot \tag{4}$$

Since irreversible processes occurring during the heat transfer are not directly taken into account in the definition of the heat transfer effectiveness, the second law of thermodynamics was applied to enable a more comprehensive assessment of the heat exchanger performance. According to the second law of thermodynamics, irreversible processes take place in the heat exchanger during the heat flow, their measure being the rate of entropy generation. The entropy generation rate in the heat exchanger results from the heat flow and the resistance of heat-transferring fluids to flow (pressure losses). The heat exchangers should be designed so that the losses due to irreversible processes accompanying the heat flow are kept as small as possible. MacClintock [5] and Prigogine [6] were the first to assess heat exchanger performance by introducing the minimization of the entropy generation. Bejan [7, 8] developed the approach of the entropy generation minimization (EGM) and proposed an entropy generation number N_S defined as the entropy generation rate ($\dot{S}$) divided by the lower heat capacity rate:

$$N_s = \frac{\dot{S}}{C_{\min}}, \tag{5}$$

where the entropy generation rate for the heat exchanger in which the heat is transferred between two fluids is equal to:

$$\dot{S} = \dot{m}_1 \Delta s_1 + \dot{m}_2 \Delta s_2 \geq 0. \tag{6}$$

In his approach, Bejan considered two types of irreversibility: one resulting from the heat transfer, and one associated with the resistance of heat-transferring fluids to flow. For variables (temperatures and mass flow rates) at the heat exchanger inlet and for known heat exchanger geometry, Bejan introduced a parameter N_S as a function of heat transfer effectiveness ε, and observed that the heat transfer effectiveness does not always increase with a decrease in entropy and does not always reach maximum. Bejan named such a behaviour of N_S with respect to the heat transfer effectiveness a paradox. This paradox was explained in [9, 10]. To explain it, some researchers introduced a revised parameter N_S.

For example, the parameter proposed by Bejan (N_S) was developed in [10-12] wherein another definition was given under the name of an entropy generation index, equal to the sum of entropy generation rates ($\dot{S}$) divided by a product of an overall heat transfer coefficient (U) and the heat transfer surface area (A):

$$\Gamma = \frac{\dot{S}}{UA}. \tag{7}$$

A parameter proposed by Hesselgreaves [13] is equal to the sum of entropy generation rates for the heat exchanger divided by the heat flow rate considering the inlet temperature of the colder fluid so that the proposed parameter is dimensionless (the revised entropy generation number [14]):

$$N_{s1} = \frac{T_{2i}\,\dot{S}}{\dot{Q}}. \tag{8}$$

Shah and Skiepko [15] presented relations between the heat transfer effectiveness and entropy generation for various types of heat exchangers. They demonstrated that for a minimum entropy generation the effectiveness can have an intermediate maximum or minimum value.

In the literature a number of papers [16-23] can be found wherein the entropy generation minimization is analysed regarding various types of heat exchangers. The EGM method is mainly employed in selecting optimum geometric parameters of the heat exchanger, such as the tube inner diameter [24-28]

Mohammed [29] introduced the parameter N_S as a function of the heat transfer effectiveness, the ratio of heat capacity rates and temperature ratio at the heat exchanger inlet. He demonstrated that the heat transfer effectiveness of a heat exchanger should approach one, as in such a case irreversible processes of a smaller extent (lower entropy generation) are expected. Mohammed [29] proved that the entropy generation associated with the resistance of flow is much lower than that resulting from the heat transfer in the heat exchanger.

Ahmad Fakheri [30] indicated that the entropy minimization should not be an objective function for the heat exchanger, and proposed an entropy flux which is defined by Eq. (6).

In most of the papers cited, the entropy generation minimization was analysed for variable parameters (temperatures and mass flow rates) at the heat exchanger inlet and for known heat exchanger geometry, or for constant inlet and outlet parameters (temperatures and mass flow rates) so that an optimal geometry, e.g. the tube diameter or the pipes pitch can be chosen. In the author's view, it is important to determine not only the optimal parameters but also those operating conditions that are the most unfavourable in terms of thermodynamics so that they can be avoided during the design. For this purpose a heat flow rate was sought for which the largest entropy generation rate (the largest exergy loss) occurs at known (constant) inlet parameters.

The largest exergy loss occurs for the largest entropy generation rate. According to the Guoy-Stodola theorem, the exergy loss equals the product of the entropy generation rate and the ambient temperature [31, 32]:

$$\delta B = T_o \dot{S}. \tag{9}$$

The entropy generation was analysed as a function of the heat flow rate for known parameters (temperatures, mass flow rates) at the inlet in the case of a heat exchanger with phase change (a condenser) and without it (a counter-flow double-tube heat exchanger).

2. ANALYSIS OF ENTROPY GENERATION: A PHASE-CHANGE HEAT EXCHANGER (A CONDENSER)

In a condenser, the colder fluid receives heat from the condensing fluid which, depending on the state of steam, gives up some or all heat of evaporation. In the condenser under consideration, water flows through tubes and receives the heat from the condensing steam flowing by the outer surface of the tubes. Pressure losses were not taken into account on the water and steam sides. The temperature and mass flow rate of cooling water and the saturation temperature of steam were assumed as the known parameters.

The entropy change due to the phase transition of saturated steam is [1-3, 32]:

$$\Delta s_1 = -\frac{r}{T_s} \cdot \tag{10}$$

The entropy generation rate for water equals [1-3, 32]:

$$\Delta s_2 = c_w \ln\left(\frac{T_{2o}}{T_{2i}}\right) \cdot \tag{11}$$

The sum of entropy generation rates for both the fluids can be expressed by Eq. (6).

By inserting Eqs. (10) and (11) into Eq. (6), it can be obtained:

$$\dot{S} = -\frac{\dot{m}_1 r}{T_s} + \dot{m}_2 c_w \ln\left(\frac{T_{2o}}{T_{2i}}\right) \geq 0 \cdot \tag{12}$$

On considering the equations for the heat flow rate:

$$\dot{Q} = \dot{m}_1 r, \tag{13}$$

$$\dot{Q} = \dot{m}_2 c_w \left(T_{2o} - T_{2i}\right), \tag{14}$$

Eq. (12) takes the form:

$$\dot{S} = -\frac{\dot{Q}}{T_s} + \dot{m}_2 c_w \ln\left(\frac{\dot{Q}}{\dot{m}_2 c_w T_{2i}} + 1\right) \geq 0 \cdot \tag{15}$$

Eq. (15) shows a relation between the entropy generation rate in the condenser and the heat flow rate. In order to determine the maximum value of the entropy generation rate for the condenser at constant temperature and mass flow rate of cooling water and saturation temperature of steam, one has to calculate a derivative:

$$\frac{d\dot{S}}{d\dot{Q}} = -\frac{1}{T_s} + \frac{\dot{m}_2 c_w}{\dot{Q} + \dot{m}_2 c_w T_{2i}} = 0 \cdot \tag{16}$$

When the condition (16) is taken into account, the value of the heat flow rate for the maximum entropy generation rate in the condenser is obtained as equal to the maximum heat flow rate according to the definition of the heat transfer effectiveness (1):

$$\left.\dot{Q}\right|_{\dot{S}_{\max}} = \dot{m}_2 c_w (T_s - T_{2i}) = \dot{Q}_{\max} \cdot \tag{17}$$

By inserting (17) into (15), it can be written:

$$\dot{S}_{\max} = -\frac{\dot{m}_2 c_w (T_s - T_{2i})}{T_s} + \dot{m}_2 c_w \ln\left(\frac{T_s}{T_{2i}}\right) \geq 0 \cdot \tag{18}$$

The first term in Eq. (18) is the maximum entropy generation rate on the steam side:

$$-\dot{m}_1 \Delta s_{1\max} = -\dot{m}_2 c_w \left(1 - \frac{T_{2i}}{T_s}\right) = \frac{-\dot{Q}_{\max}}{T_s} \cdot \tag{19}$$

The second term in Eq. (18) is the maximum entropy generation rate on the water side:

$$\dot{m}_2 \Delta s_{2\max} = \dot{m}_2 c_w \ln\left(\frac{T_s}{T_{2i}}\right) \cdot \tag{20}$$

On considering Eqs. (19) and (20), Eq. (18) takes the form:

$$\dot{S}_{\max} = \dot{m}_1 \Delta s_{1\max} + \dot{m}_2 \Delta s_{2\max} \geq 0 \tag{21}$$

In the case of a phase-change heat exchanger (condenser) at constant temperature and mass flow rate of cooling water and steam saturation temperature, with the increase in the heat flow rate, the entropy generation rate increases and reaches the maximum value for the heat flow rate equal to Q_{max} according to Eq. (17), being the maximum flow rate of the heat that can be transferred in the heat exchanger. Then, according to Eq. (1), the heat transfer effectiveness equals one (ε = 1), while the outlet temperature of cooling water equals the saturation temperature of steam (T_{2o}=T_s).

3. ANALYSIS OF ENTROPY GENERATION: A HEAT EXCHANGER WITHOUT PHASE CHANGE

The entropy generation in the case of a heat exchanger without phase change was analysed for a counter-flow double-tube heat exchanger with water as heat transferring fluids. To simplify the analysis, no pressure losses for both fluids were taken into account. The temperatures at the heat exchanger inlet and mass flow rates of both the fluids were assumed as the known parameters.

The entropy generation rate in the case of the counter-flow heat exchanger in question has the form of Eq. (6).

The entropy generation rate for the hotter fluid has the form:

$$\Delta s_1 = c_{p1} \ln\left(\frac{T_{1o}}{T_{1i}}\right). \tag{22}$$

The entropy generation rate for the colder fluid has the form:

$$\Delta s_2 = c_{p2} \ln\left(\frac{T_{2o}}{T_{2i}}\right). \tag{23}$$

Combining Eqs. (22) and (23) the total entropy generation rate in heat exchanger can be written as:

$$\dot{S} = -\dot{m}_1 c_{p1} \ln\left(\frac{T_{1i}}{T_{1o}}\right) + \dot{m}_2 c_{p2} \ln\left(\frac{T_{2o}}{T_{2i}}\right) \geq 0, \tag{24}$$

or:

$$\dot{S} = \dot{m}_1 c_{p1} \ln\left(\frac{T_{1o}}{T_{1i}}\right) + \dot{m}_2 c_{p2} \ln\left(\frac{T_{2o}}{T_{2i}}\right) \geq 0. \tag{25}$$

On considering the equations for the heat flow rate:

$$\dot{Q} = \dot{m}_1 c_{p1}\left(T_{1o} - T_{1i}\right), \tag{26}$$

$$\dot{Q} = \dot{m}_2 c_{p2}\left(T_{2o} - T_{2i}\right), \tag{27}$$

Eq. (25) takes the form:

$$\dot{S} = \dot{m}_1 c_{p1} \ln\left(\frac{-\dot{Q}}{\dot{m}_1 c_{p1} T_{1i}} + 1\right) + \\ + \dot{m}_2 c_{p2} \ln\left(\frac{\dot{Q}}{\dot{m}_2 c_{p2} T_{2i}} + 1\right) \geq 0 \quad . \tag{28}$$

At constant inlet temperatures and mass flow rates of both fluids, in order to evaluate the maximum value of the entropy generation rate for the heat flow rate, one has to calculate the derivative of Eq. (28).

Taking into account the condition for the extremum of entropy generation rate ($d\dot{S}/d\dot{Q} = 0$), the value of the heat flow rate for the maximum entropy generation rate is obtained in the form:

$$\dot{Q}\Big|_{\dot{S}_{\max}} = \frac{\dot{m}_2 c_{p2} \dot{m}_1 c_{p1}\left(T_{1i} - T_{2i}\right)}{\dot{m}_1 c_{p1} + \dot{m}_2 c_{p2}}. \tag{29}$$

Eq. (29) can be expressed as a function of the maximum heat flow rate according to Eq. (1) and the ratio of heat capacity rates:

$$\dot{Q}\Big|_{\dot{S}_{\max}} = \frac{\dot{Q}_{\max}}{1 + C_r}. \tag{30}$$

The heat transfer effectiveness for the maximum entropy generation rate equals:

$$\varepsilon = \frac{1}{1 + C_r}. \tag{31}$$

By inserting (29) into (28), we obtain the maximum entropy generation rate:

$$\dot{S}_{\max} = \dot{m}_1 c_{p1} \ln\left[\frac{\dot{m}_2 c_{p2} T_{2i} + \dot{m}_1 c_{p1} T_{1i}}{(\dot{m}_1 c_{p1} + \dot{m}_2 c_{p2}) T_{1i}}\right] + \\ + \dot{m}_2 c_{p2} \ln\left[\frac{\dot{m}_2 c_{p2} T_{2i} + \dot{m}_1 c_{p1} T_{1i}}{(\dot{m}_1 c_{p1} + \dot{m}_2 c_{p2}) T_{2i}}\right] \geq 0 \quad . \tag{32}$$

By comparing Eqs. (25) and (32), we obtain two conditions:

$$T_{1o} = \frac{\dot{m}_2 c_{p2} T_{2i} + \dot{m}_1 c_{p1} T_{1i}}{(\dot{m}_1 c_{p1} + \dot{m}_2 c_{p2})}, \tag{33}$$

$$T_{2o} = \frac{\dot{m}_2 c_{p2} T_{2i} + \dot{m}_1 c_{p1} T_{1i}}{(\dot{m}_1 c_{p1} + \dot{m}_2 c_{p2})}. \tag{34}$$

The comparison between the conditions (33) and (34) indicates that the maximum entropy generation rate in the heat exchanger (without considering pressure losses), at constant temperatures at the inlet and mass flow rates of both the fluids, occurs when their outlet temperatures are equal:

$$T_{2o} = T_{1o}. \tag{35}$$

4. A DESCRIPTION OF A COUNTER-FLOW DOUBLE-PIPE HEAT-EXCHANGER SIMULATOR

The above derived relations were verified against data obtained from a simulator of the counter-flow heat exchanger. A diagram of the counter-flow double-pipe heat exchanger with commonly used symbols is shown in Fig. 1. It was assumed that the hotter fluid flows in a tube of smaller diameter d = 0.03 m. The colder fluid flows between two tubes. The diameter of the outer tube is D = 0.043 m. The heat transferring fluids were assumed to be water. The temperature of the water giving up heat at the heat exchanger inlet is 52°C. The temperature of the heated water at the heat exchanger inlet is 17°C.

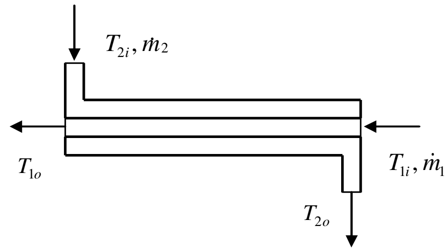

Fig. 1. A diagram of the counter-flow double-pipe heat exchanger with commonly used symbols

The simulator input data were temperatures at the inlet and mass flow rates of both fluids, and geometrical data (inner and outer diameters, length of the heat exchanger). The output (calculated) data were temperatures of both fluids at the heat exchanger outlet, overall heat transmission coefficient, heat flow rate and the heat transfer effectiveness.

Calculations were performed for two relations between the heat capacity rates: $C_2 > C_1$ and $C_1 > C_2$. For the first one, the mass flow rates were $\dot{m}_2 = 0.1$ kg/s for the colder fluid, and $\dot{m}_1 = 0.03$ kg/s for the hotter fluid. For the second one, $\dot{m}_1 = 0.1$ kg/s and $\dot{m}_2 = 0.03$ kg/s were assumed.

The calculations were performed for constant inlet temperatures and mass flow rates of both fluids for a heat exchanger length from 1 to 10 m.

5. RESULTS

The output data, i.e. heat flow rate, and heat exchanger effectiveness, obtained from the counter-flow heat-exchanger simulation, are shown in the following figures for the two relations between heat capacity rates: $C_2 > C_1$ and $C_1 > C_2$.

The heat flow rate as a function of the heat exchanger length is displayed in Fig. 2.

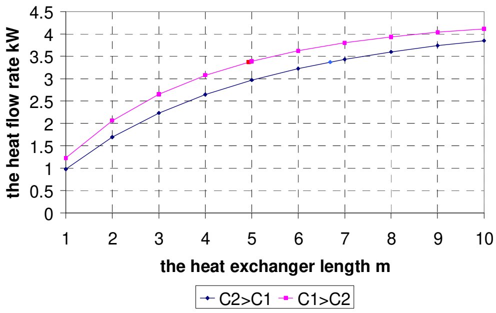

Fig. 2. The heat flow rate as a function of the heat exchanger length

The heat transfer effectiveness of the heat exchanger as a function of the heat exchanger length is shown in Fig. 3.

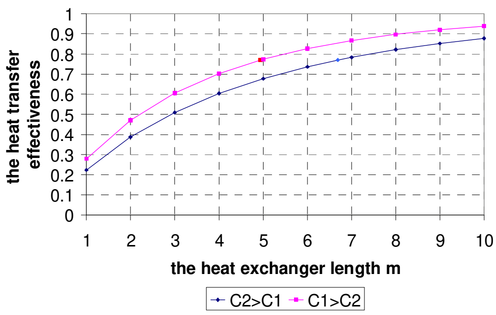

Fig. 3. The heat transfer effectiveness of the heat exchanger as a function of the heat exchanger length

At constant parameters (temperatures and mass flow rates) at the heat exchanger inlet in Figs. 2 and 3, a regularity of changes in the heat flow rate and heat transfer effectiveness can be observed. With increasing heat transfer surface area the heat flow rate and the heat transfer effectiveness of the heat exchanger increases.

The overall heat transfer coefficient for the relation $C_1 > C_2$ is greater than $C_2 > C_1$, this is the reason why the performance (heat flow rate, effectiveness) of the heat exchanger for the relation $C_1 > C_2$ is greater.

The entropy generation rate as a function of the heat exchanger length is displayed in Fig. 4 with clear maximum values.

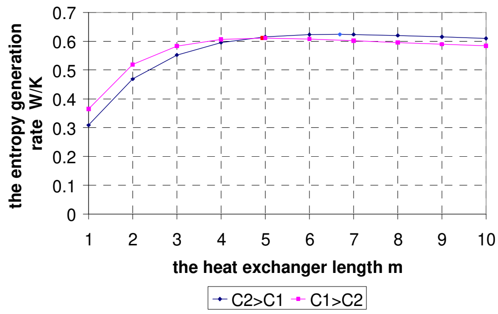

Fig. 4. Entropy generation rate as a function of the heat exchanger length

Based on data shown in Fig. 4, the maximum entropy generation rate in the heat exchanger can be found. For short heat exchanger pipes a substantial rise of entropy generation rate is observed with increasing heat transfer surface area until the maximum value is reached. Following the maximum value of the entropy generation rate, further increase in the heat transfer surface area results in a slow decrease in the entropy generation rate in the heat exchanger.

Due to the higher the overall heat transfer coefficient for the relation $C_1 > C_2$ the shorter length of the heat exchanger in order to achieve equal exit temperature is needed. Therefore, the maximum entropy generation rate for relation $C_1 > C_2$ occurs for a smaller length of the heat exchanger.

For the relation between the heat capacity rates $C_2 > C_1$ the maximum entropy generation rate was $\dot{S}$ = 0.624 W/K for the heat exchanger length of $L = 6.688$ m. For this length of the heat exchanger, the heat flow rate equals $\dot{Q}$ = 3.375 kW, the heat transfer effectiveness is $\varepsilon = 0.769$, while the temperatures at the heat exchanger outlet are equal: $t_{2o} = t_{1o} = 25.068$°C.

For the relation between the heat capacity rates $C_1 > C_2$ the maximum entropy generation rate was $\dot{S}$ = 0.611 W/K for the heat exchanger length of $L = 4.937$ m. For this length of the heat exchanger, the temperatures at its outlet are equal: $t_{2o} = t_{1o} = 43.921$°C; the values of heat transfer rate and effectiveness are the same as in the case of the relation between the heat capacity rates $C_2>C_1$, which follows from the assumed values of mass flow rates and Eqs. (30) and (31).

The change in parameter Γ as defined by Eq. (7) with increasing heat exchanger length is shown in Fig. 5.

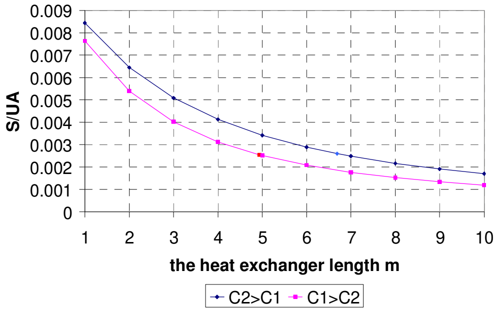

Fig. 5. The change in parameter Γ as defined by Eq. (7) as a function of the heat exchanger length

The change in parameter Ns_1 as defined by Eq. (8) with increasing heat exchanger length is shown in Fig. 6.

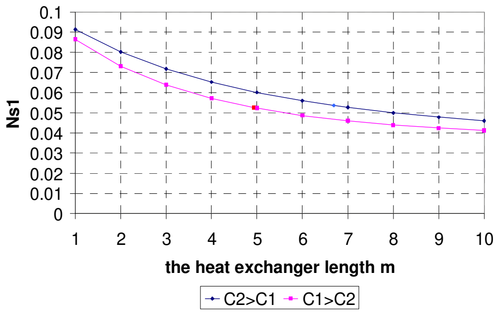

Fig. 6. The change in parameter Ns_1 as defined by Eq. (8) as a function of the heat exchanger length

For parameters defined by Eqs. (7) and (8), no extremum exists with increasing heat transfer surface area for constant parameters (temperatures and mass flow rates) of both the fluids at the heat exchanger inlet for the two relations between the heat capacity rates ($C_2>C_1$ and $C_1>C_2$).

Exergy loss (Eq. (9)) with clear maximum values is shown in Fig. 7. Reference temperature (T_o) was assumed to be equal to the temperature of the colder fluid (water) (T_{2i}) at the heat exchanger inlet. As for entropy generation rate (Fig. 4) heat transfer surface area at which exergy losses reach maximum depends on the ratio of C_1 and C_2. This extremum can be easily determined from the charts presented or, more precise-ly, with the use of heat exchanger computa-tional simulator.

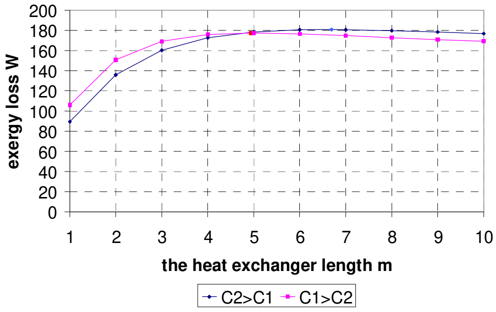

Fig. 7. Exergy loss (9) as a function of the heat exchanger length

6. CONCLUSIONS

The main goal of the paper is to provide a condition for which a maximum entropy generation occurs in a heat exchanger at constant inlet parameters (temperatures and mass flow rates). If such a working point is known, one of the most unfavourable operating conditions can be avoided during the design of the heat exchanger.

The paper presents an analysis of the entropy generation for a condenser and a counter-flow double-pipe heat exchanger. In the paper, the entropy generation resulting from the heat flow was considered, whereas the entropy generation due to the resistance of heat-transferring fluids to flow was not taken into account.

The entropy generation rates were presented as functions of the heat flow rate. For constant parameters (temperatures and mass flow rates) at the heat exchanger inlet, a heat flow rate for which the largest entropy generation rate occurs and the conditions for such a rate were sought.

For the condenser, the maximum entropy generation rate occurs for the maximum flow rate of the heat that can be transferred according to the definition of heat transfer effectiveness (1), which corresponds to the equality of the outlet temperatures of the coolant and condensing fluid.

For the counter-flow heat exchanger, the entropy generation reaches maximum as a function of the heat flow rate. It appeared that at constant parameters (temperatures and mass flow rates) at the heat exchanger inlet the maximum entropy generation, or the largest exergy loss, occurs when the outlet temperatures of the fluids are equal. This feature was verified against data obtained from a simulator of the counter-flow heat exchanger. At constant inlet parameters (temperatures and mass flow rates), heat transfer surface areas were increased, and calculations were performed. The data from the counter-flow heat-exchanger simulator confirmed that the maximum entropy generation rate (the largest exergy loss) occurs under the condition that the outlet temperatures of the fluids are equal. The entropy generation for the counter-flow heat exchanger was analysed for two relations between the heat capacity rates of fluids: $C_2 > C_1$ and $C_1 > C_2$.

Nomenclature

Symbols

A – heat transfer area, m^2
C – heat capacity rate, W/K
C_r – ratio of heat capacity rates $C_r = C_{min}/C_{max}$, –
c_p – specific heat at constant pressure, J/(kg K)
c_w – specific heat of water, J/(kg K)
d – diameter of the smaller tube, m
D – diameter of the larger tube, m
L – length of the heat exchanger, m
$\dot{m}$ – mass flow rate, kg/s
NTU – the number of heat transfer units $NTU = UA/C_{min}$, –
N_{s1} – the revised entropy generation number, –
r – phase transition heat, J/kg
s – specific entropy, J/kg/K
$\dot{S}$ – rate of entropy generation, W/K
T – temperature, K
t – temperature, °C
U – overall heat transfer coefficient, W/(m^2 K)
$\dot{Q}$ – heat flow rate, W
δB – exergy loss, W
ε – heat transfer effectiveness, -
Γ – the entropy generation index, entropy flux, –

Indices

1 – hot fluid
2 – cold fluid
i – inlet
o – outlet, reference conditions
s – saturated conditions
max – maximum value
min – minimum value

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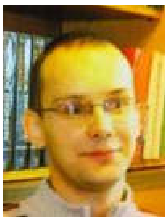

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