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Optimization of Design for Air Type Evaporator Heat Exchanger

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Abstract

This work provides an overview of air type evaporators, specifically focusing on coil type evaporators. It introduces a simplified method for calculating the required heat transfer area of a finned evaporator based on specified thermal and flow parameters, considering air and R134a refrigerant, as well as given design parameters. The study analyzes the influence of individual parameters on the heat transfer surface, using a base example as reference. Furthermore, it examines how changes in various system parameters affect the heat transfer surface. Such parameter analysis can aid in designing new heat exchangers and optimizing existing evaporators.

Key words: heat exchanger, evaporator, heat transfer, optimization

1. Introduction

Heat transport is a phenomenon that can be observed in almost every area of life. For many centuries, people have used the process of providing heat - for thermal processing of equipment and food, or primarily for heating purposes, but even then there were also ways to cool rooms or preserve and protect food - for example, hiding food in ice or creating straw covered pits in the ground - a primitive equivalent of today's refrigerator.

The energy for transport was massively used in the 18th century. By supplying heat to the system, the system "reciprocated" by doing work. This phenomenon was crucial in the design of the first steam engines and machines. The invention of electricity at this time also intensified the pace of development of technology related to heat transport and all forms of energy transport, which is why this time is often called the "industrial revolution".

The reverse process, i.e. the removal of heat from the system, also found its commercial application in the second half of the 19th century. According to most sources, the inventor of the refrigerator was Bavarian engineer Carl von Linde, who developed the process of heat transfer for refrigeration purposes; he used it to enable the production of beer in the summer in one of the breweries. The refrigeration circuit he used in his device is very often used and developed even today. It is based on a series of thermodynamic transformations of the working medium, as a result of which the heat supplied to the system in the lower heat exchanger is transported and released in the upper heat exchanger, and work is supplied to the system in a compressor powered by electricity. Over the years, methods of heat transport for refrigeration purposes have also been invented, using the effects of thermoelectric, thermoacoustic and magnetocaloric phenomena.

The dynamic development of refrigeration is visible in almost every industry branch today. Industrial refrigeration technology is used, among others, in the automotive industry, for cooling industrial equipment in air conditioning and medicine. However, its main application is, of course, in the food industry, where it is a basis for extending the shelf life of products. Nowadays, there are many types of refrigeration equipment. There is a scheme for selecting a specific type of device for specific conditions in which it would best fulfill its role. A different device or its elements will be selected for working with food, and another for cooling the ice rink. The structure and principle of operation of a refrigeration device also determines the temperature

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range that can be obtained with it. The temperature of absolute zero cannot be achieved using a home refrigerator, as it serves its purpose within a specific temperature range.

Therefore, work is constantly ongoing to develop and improve these devices. The main goal is to achieve greater efficiency of such a device in order to reduce the costs of its operation and production of such refrigeration devices. There are many factors that influence its efficiency. That is why all types of analyzes of the system's operation and examination of individual parameters are so important. For a given type of refrigeration device, there is an appropriate algorithm for calculating the theoretical values of individual parameters.

1.1. Heat exchangers in refrigeration and air conditioning devices

Heat exchangers are a key element of refrigeration systems. After all, the principle of operation of a refrigeration system is based on the transport of energy (heat) from one medium, lowering its temperature, to another medium, increasing its temperature. The structure of individual exchangers varies, depending on the principle of operation and the method of energy transport. Due to their construction, the following types of heat exchanger can be met on the market:

- plates type,
- tubular type,
- lamella type,
- shell and tube type,
- tubular coaxial type,
- other.

Due to their function in a refrigeration installation, heat exchangers can be classified into the following types:

- evaporators,
- condensers,
- gas coolers,
- regenerative heat exchangers,
- other exchangers (e.g. oil cooler in the automotive industry).

In refrigeration systems, the cooling effect is provided by evaporators. They capture heat from the system, reducing its temperature. In practice, there are probably as many types of evaporators as there are needs to use them. Evaporators can be divided according to the medium being cooled:

- for cooling gases (air evaporators);
- for cooling liquids;
- for cooling solids;

Fluids can be injected into the heat exchanger co-currently or counter-currently, which changes the parameters of the media at the outlet (Fig. 1.).

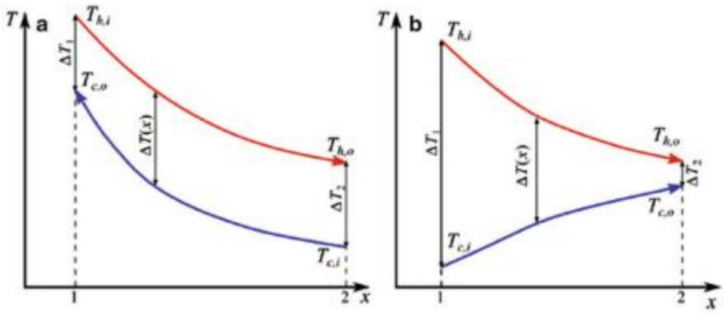

Fig. .1 Temperature changes in the co-current exchanger (on the left) and counter-current exchanger (on the right) [1]

1.2. Heat transfer processes in heat exchangers

In tubular evaporators, heat transfer takes place between two media - one flowing inside the tubes and the other flowing around the outer surface of the tubes. The heat transfer process then takes place, i.e. the complex transport of thermal energy, which consists of convective heat transfer between the cooled fluid and the outer surface of the pipes (and fins, if any), heat conduction inside the partition, convective heat transfer between the inner surface of the pipes and fluid inside the pipes (Fig. 2.).

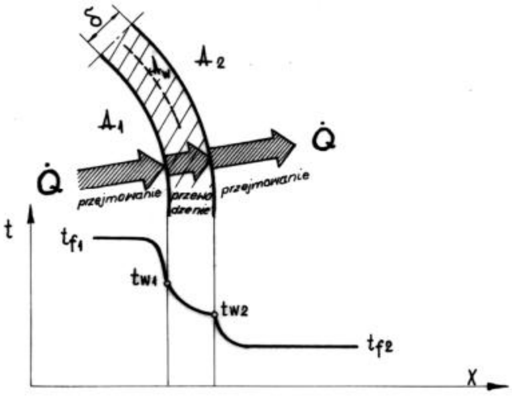

Fig. 2. Temperature distribution during heat transfer when the cooled fluid flows inside the pipe, i.e. $t_{f1} > t_{f2}$ [6]

In special cases, the heat transfer process may also be accompanied by heat radiation. Then the heat transfer coefficient is the sum of both mechanisms - convective heat transfer and radiative heat transfer, while each of these elements can be calculated separately from appropriate dependencies. In evaporators and condensers, there is also a change in the state of matter of a given medium. In condensers, this happens when heat is released and the refrigerant condenses, while in evaporators, the refrigerant boils when heat is absorbed. In such cases, the heat capacity of the medium changing the state of matter is much larger than the medium that absorbs or releases heat (e.g. air). In such considerations, it can be assumed that the temperature during boiling or condensation is constant (Fig. 3.).

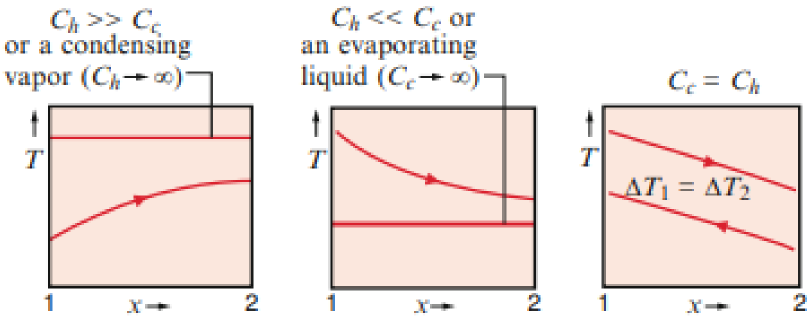

Fig. 3. Temperature distribution of media in heat exchangers.
From the left: during condensation, during evaporation, when the heat capacities of the media are similar.

To calculate the heat capacity $\dot{Q}$ of the heat exchanger, three main parameters can be used - heat transfer coefficient k, heat transfer area A and average logarithmic temperature difference ΔT *(LMTD)*. The relationship can be presented by equation 1:

$$\dot{Q} = k \cdot A \cdot \Delta T \tag{1}$$

The analysis according equation 1 leads to the conclusion that as each of these three components of the equation increases, the efficiency of the exchanger increases. The heat transfer coefficient depends strictly on the material properties of the exchanger and the properties of the factors between which heat exchange takes place, which is why their appropriate selection is so important. The larger the heat exchange surface, the greater the efficiency of the heat exchanger. Therefore, in order to intensify heat transfer in tubular heat exchangers, additional fins are used to ensure that the heat transfer surface is as large as possible.

2. Heat transfer model

2.1. LMTD method

The equation (1) presented in the previous section, describing the heat flow transferred in the exchanger, includes in itself a dependence on the average logarithmic difference in temperatures of the streams of both fluids at the inlet and outlet of the exchanger. Calculating the Logarithmic Mean Temperature Difference is one of the methods for determining the capacity of the exchanger, because the higher the LMTD value, the more intense the heat exchange between the fluids. The LMTD coefficient is determined by the formula:

$$LMTD = \frac{\Delta T_A - \Delta T_B}{ln\left(\frac{\Delta T_A}{\Delta T_B}\right)} \tag{2}$$

where:
ΔT_A – temperature difference between the medium streams at the beginning of the heat exchanger,
ΔT_B – temperature difference between the medium streams at the end of the heat exchanger.

Equation (2) is met for co-current and counter-current flow. If the flow of streams takes place at an angle, i.e. it is cross-shaped, then a correction factor should also be taken into account in equation (1). The LMTD method assumes a steady state of flow. An additional assumption in this method is the constant value of the heat transfer coefficient and its change independent of temperature changes, so it may sometimes be an inaccurate method.

2.2. NTU method

In the case of the method of determining heat transfer efficiency based on the average logarithmic temperature difference, there is one more imperfection. The LMTD method gives correct results for fluids with constant specific heat and small temperature difference ranges. Unfortunately, it cannot be used for thermal calculations of condensers and evaporators because it does not take into account the latent heat associated with the phase change. In such cases, the right method to determine the thermal efficiency of the exchanger is the method using the number of heat transfer units (NTU) (Number of Transfer Units), which is determined by the relationship:

$$NTU = \frac{k \cdot A}{C_{min}} \tag{3}$$

and the ratio of the heat capacities of the streams of both C_r factors is equal to:

$$C_r = \frac{C_{min}}{C_{max}} \tag{4}$$

Then the capacity of the evaporator can be determined as follows:

$$\dot{Q} = \varepsilon \cdot C_{min}(T_{p1} - T_o) \tag{5}$$

where:
T_{p1} - temperature of the cooled medium at the entrance to the exchanger, [°C],
T_o - boiling point of the refrigerant, [°C],
ε - thermodynamic efficiency of the exchanger [-].

The efficiency of the heat exchanger depends on the number of heat transfer units NTU and is respectively:

for co-current flow:

$$\varepsilon = \frac{1 - e^{-NTU(1+C_r)}}{1 + C_r} \tag{6}$$

in cases:

$$C_r \to 0\,, \ \varepsilon = 1 - e^{-NTU}\,;\ C_r \to 1\,, \ \varepsilon = \frac{1}{2} \tag{7}$$

for counter-current flow:

$$\varepsilon = \frac{1 - e^{-NTU(1+C_r)}}{1 - C_r e^{-NTU(1+C_r)}} \tag{8}$$

in cases:

$$C_r \to 0\,, \ \varepsilon = 1 - e^{-NTU};\ C_r \to 1\,, \ \varepsilon = \frac{NTU}{1 + NTU} \tag{9}$$

2.3. Heat transfer coefficient for finned pipes bundle

The heat transfer coefficient on the side of the medium flowing through the finned heat exchanger can be calculated using an algorithm where the heat transfer coefficient:

$$\propto = RCJ\frac{Nu\lambda}{d_z} \tag{10}$$

Where the Nusselt number is determined by the Schmidt formula:

$$Nu = C \cdot Re^{0.6}\left(\frac{A_c'}{A_g'}\right)^{-0.15} Pr^{\frac{1}{3}} \tag{11}$$

where:

$$A_c' = A_ż' + A_{mż}' \tag{12}$$

A_g' - external surface area of an unfinned pipe $[m^2]$,
$A_{mż}'$ - external surface of the pipe between the lamellas, $[m^2]$,
$A_ż'$ - the surface of the lamella $[m^2]$.
RCJ – moisture exchange coefficient, defining the ratio of total heat to the sensible heat of the process; if only air cooling occurs in the exchanger, without dehumidification, then RCJ = 1.0.

The K constant in formula (3.12) for the number of rows greater than four takes its values depending on the type of pipe system in the exchanger. The values in each configuration are presented in table 3.1.

Table 1. Values of the *C* coefficient from the equation (11)

Pipe layout	inline	staggered
K	0.22	0.38

The Reynolds number is then defined by the following equation:

$$Re = \frac{w_0 d_z \rho}{\mu} \tag{13}$$

and it is correct for: $10^3 < Re < 10^5$ and $5 < \frac{A_c}{A_g} \leq 30$.

The velocity w_0 is calculated for the smallest free cross-section. It is determined depending on whether the pipes are arranged in series or in a staggered arrangement in the exchanger (Fig. 4.).

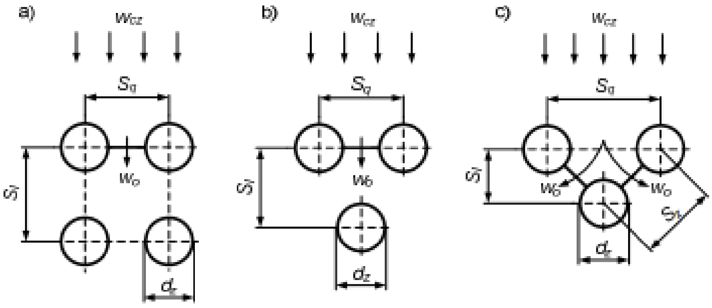

Fig. 4. Pipe arrangement in the tubular exchanger [9]

2.4. Fin efficiency

To fully analyze the heat exchanger, it is necessary to determine the efficiency of the pipe fins (lamellas) used. This efficiency is closely related with the heat transfer coefficient, the method of determining which was presented in the previous section. Additionally, heat exchange through the fin takes place through conduction, so the material selected for the production of lamellas is very important. The higher the conductivity coefficient, the higher the efficiency of the fin. The efficiency of the fins is determined by the formula:

$$\varepsilon_f = \frac{tanh(m \cdot h_f)}{m \cdot h_f} \tag{14}$$

where:

$$m = \sqrt{\frac{2 \cdot \alpha}{\delta_f \cdot \lambda_f}} \tag{15}$$

2.5. Heat transfer coefficient boiling in horizontal pipes

Determining the heat transfer coefficient during the evaporation of a refrigerant is a multidimensional and mathematically complex issue. For design and analysis purposes, the average heat transfer coefficient at the boiling of liquids in horizontal pipes can be determined, assuming complete evaporation of the refrigerant, for example using the generalized formula of Iwaszkiewicz [10], where:

$$\alpha_o = C A_w{}^{-0.7} \tag{16}$$

where:

$$C = 2\frac{\lambda'}{l^{0.3}} Ar^{-0,175} \left(\frac{Pr'}{Ku}\right)^{0.35} \left(\frac{\dot{Q}}{r\mu'}\right)^{0.7} \left(\frac{\rho'}{\rho''}\right)^{0.525} \tag{17}$$

where Ar - Archimedes, Pr - Prandtl and Ku -Kutateladze numbers are. The characteristic linear dimension l is:

$$l = \sqrt{\frac{\sigma}{g(\rho' - \rho'')}} \tag{18}$$

2.6. Algorithm for determining the required heat exchange surface area in a finned evaporator for air cooling

To design a heat exchanger such as an evaporator, but also to analyze it and optimize already used exchangers, an algorithm can be used to calculate the required heat exchange surface with the known efficiency of such an evaporator, using the NTU method. For the algorithm, it was assumed that the cooled medium is air, while the refrigerant flows in the pipes. The following thermal and flow parameters should be used for such calculations:

- thermal efficiency of the evaporator $\dot{Q}, [W]$,
- air flow$\dot{V_p}$, $\left[\frac{m^3}{s}\right]$,
- air temperature at the evaporator inlet $T_{p1}, [°C]$,
- relative humidity of the air at the evaporator inlet $\varphi_{p1}, [-]$,
- boiling point of the medium $T_o, [°C]$,

Additionally, the design parameters of the exchanger should also be assumed (Fig. 5.):

- external diameter d_z and internal diameter of pipes $d_w, [m]$,
- lamella pitch $t, [m]$,
- lamella thickness $\delta_f, [m]$,
- longitudinal pitch of the pipe system $S_l, [m]$,
- transverse pitch of the pipe system $S_q, [m]$,
- height of the exchanger front surface $H, [m]$,
- width of the front surface of the exchanger $G, [m]$.

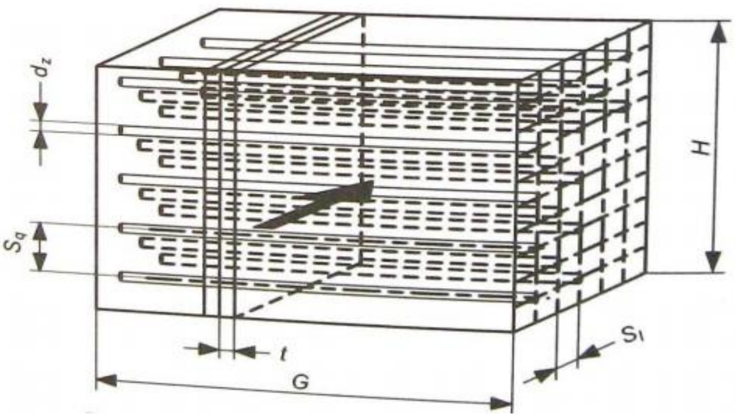

Fig. 5. Design parameters of the heat exchanger

The evaporator capacity can be represented by the equation (19), where the minimum heat capacity will be the heat capacity of the air stream $\dot{W_p}$:

$$\dot{Q} = \varepsilon \dot{W}_p (T_{p1} - T_o) \tag{19}$$

3. Analysis

The following assumptions were made to analyze the finned type air evaporator:

- evaporator capacity is 15 kW,
- the refrigerant is R134a with a boiling point of T_o = -2 °C;
- the cooled medium is the flowing air;
- fins are made of aluminum;
- tubes are made of copper;
- pipe system in the exchanger – inline.

The design parameters of the exchanger were listed in table 2:

Table 2. Design parameters of the evaporator

G, m	H, m	d_w, m	d_z, m	S_q, m	S_l, m	n_r	t, m	$\delta_{\dot{z}}, m$	$\lambda_{\dot{z}}, \frac{W}{mK}$	$\lambda_r, \frac{W}{mK}$
1	0.4	0.01	0.008	0.025	0.025	16	0.002	0.00012	228	390

The thermal and flow parameters of air and its properties for the assumed temperature are shown in table 3:

Table 3. Parameters of the air flowing into the exchanger

$\rho_p, \frac{kg}{m^3}$	$c_{pp}, \frac{J}{kgK}$	$\lambda_p, \frac{W}{mK}$	$\mu_p, Pa \cdot s$	Pr_p	$\dot{V}_p, \frac{m^3}{h}$	T_{p1}, °C	φ_{p1}
1.226	1005	0.0255	0.00001785	0.7035	4200	15	0,65

3.1. Calculation results

An Excel workbook was used for calculations and analyses, and the formulas implemented inside in individual columns allowed for the appropriate behavior of the system for the changed parameters. The calculation results for the input data are summarized in table 4.

Table 4. Calculation results of selected parameters for the starting example

Parametr	Calculated value	Unit
Evaporator		
A_w	3.13	m^2
NTU	0.95	-
ε	0.62	-
k_{Aw}	437.52	$\frac{W}{m^2K}$
N_{rz}	8	-
Air side		
α_p	43.54	$\frac{W}{m^2K}$

$\dot{W}_p$	1437.49	$\frac{W}{K}$
Re_p	3551.89	-
Pr_p	0.70	-
w_{cz}	2.92	$\frac{m}{s}$
w_o	5.17	$\frac{m}{s}$
T_{p2}	4.57	$°C$
Fins		
ε_f	0.86	-
h_f	0.127	m
Refrigerant side		
α_o	889.50	$\frac{W}{m^2K}$
Ar	188725.39	-
Pr'	3.93	-
Ku	0,00079	-
C	1975,33	-

3.2. Analysis of the influence of tube thickness on the required heat transfer surface

The created computational model was used to analyze the impact of design parameters on the required heat exchange surface area. The influence of the tube wall thickness with a constant external diameter of the tube was examined. The thickness range was tested in the range of 0.3 - 1 mm. The calculation results for selected system parameters are presented in table 5.

Table 5. Tube thickness impact on individual parameters of the evaporator

δ_r, m	A_w, m^2	$\alpha_p, \frac{W}{m^2K}$	$k_{Aw}, \frac{W}{m^2K}$	$\alpha_o, \frac{W}{m^2K}$	$T_{p2}, °C$	$w_o, \frac{m}{s}$	N_{rz}, -
0.0010	3.13	43.54	437.52	889.51	4.57	5.17	8
0.0009	3.19	43.54	429.29	877.76	4.57	5.17	8
0.0008	3.25	43.54	421.39	866.42	4.57	5.17	8
0.0007	3.31	43.54	413.79	855.46	4.57	5.17	8
0.0006	3.36	43.54	406.49	844.86	4.57	5.17	8
0.0005	3.42	43.54	399.46	834.60	4.57	5.17	8
0.0004	3.48	43.54	392.69	824.67	4.57	5.17	8
0.0003	3.54	43.54	386.16	815.05	4.57	5.17	7

The results in the table show that the smaller the tube wall thickness, the greater the demand for the necessary heat exchange surface.

3.3. Analysis of the influence of lamella (fin) thickness on the required heat transfer surface

Similarly to the case of tube wall thickness, using the presented model for determining the heat transfer surface for a lamella heat exchanger, it is also possible to examine the algorithm's response to changes in fin thickness. However, the thickness of the slats is a complex phenomenon because the thickness of the slats also affects air resistance. The algorithm presented in the previous chapters contains some simplifications, including in this respect. The presented results were calculated without additional air resistance, but, similarly to the tube wall thickness. The results of the analysis of the influence of lamella thickness are summarized in Table 6.

Table 6. The impact of the fin thickness on individual parameters of the device

δ_f, m	A_w, m^2	$\alpha_p, \frac{W}{m^2K}$	$k_{Aw}, \frac{W}{m^2K}$	$\alpha_o, \frac{W}{m^2K}$	$T_{p2}, °C$	$w_o, \frac{m}{s}$	N_{rz}, -	ε_f, -
0.00012	3.13	43.54	437.52	889.51	4.57	5.17	8	0.86
0.00014	3.07	43.82	445.70	901.11	4.57	5.23	8	0.87
0.00016	3.02	44.11	452.58	910.83	4.57	5.28	8	0.89
0.00018	2.98	44.41	458.56	919.24	4.57	5.34	7	0.90
0.00020	2.95	44.71	463.90	926.72	4.57	5.40	7	0.91

3.4. Analysis of the influence of fin pitch on the required heat transfer surface

Another design parameter that was analyzed using the created model is the influence of the slat pitch, i.e. the distance between subsequent slats. With the initial assumption that the length of the exchanger is to be unchanged and be 1 m, a change in the pitch is associated with a change in the number of fins, so such an analysis will also indicate the impact of fins on the obtained results, mainly on the coefficients related to the intensity of heat transfer. Same as in the case of changes in the thickness of the lamellas, the impact of related changes is omitted in the analysis with air flow resistance. The results of the above analysis are summarized in Table 7.

Table 7. Impact of changing the pitch between lamellas on individual parameters of the design

t, m	A_w, m^2	$\alpha_p, \frac{W}{m^2K}$	$k_{Aw}, \frac{W}{m^2K}$	$\alpha_o, \frac{W}{m^2K}$	$T_{p2}, °C$	$w_o, \frac{m}{s}$	N_{rz}, -	n, -	ε, -
0.002	3.13	43.54	437.52	889.51	4.57	5.17	8	500	0.86
0.003	4.07	45.51	335.94	739.32	4.57	5.06	10	333	0.85
0.004	4.90	47.03	279.35	649.76	4.57	5.01	12	250	0.85
0.005	5.63	48.27	242.80	589.02	4.57	4.98	14	200	0.84
0.006	6.30	49.30	217.07	544.59	4.57	4.96	16	167	0.84

In the table 7, the symbol "n" means the number of lamellas.

3.5. Analysis of the influence of air flow from fans on the required heat exchange surface

The analysis of a fin heat exchanger also requires checking the influence of other factors than just the design parameters. An important factor in the process, air flowing onto the evaporator, so it was also checked what impact the air flow regulation, and more specifically the assumed hourly air flow rate, had on the obtained results. The analysis was carried out for the air flow range of 3200 – 5200 m^3/h. The calculation results for selected parameters are summarized in table 8.

Table 8. The impact of changing the set expenditure on individual parameters of the system

$\dot{V_p}, \frac{m^3}{h}$	A_w, m^2	$\alpha_p, \frac{W}{m^2K}$	$k_{Aw}, \frac{W}{m^2K}$	$\alpha_o, \frac{W}{m^2K}$	$T_{p2}, °C$	$w_o, \frac{m}{s}$	N_{rz}, -	ε, -
5200	2.41	49.49	505.87	1067.46	6.57	6.40	6	0.50
5000	2.51	48.34	493.54	1036.25	6.23	6.16	6	0.52
4800	2.63	47.17	480.64	1003.15	5.87	5.91	7	0.54

$\dot{V}_p, \frac{m^3}{h}$	A_w, m^2	$\alpha_p, \frac{W}{m^2K}$	$k_{Aw}, \frac{W}{m^2K}$	$\alpha_o, \frac{W}{m^2K}$	$T_{p2}, °C$	$w_o, \frac{m}{s}$	N_{rz}, -	ε, -
4500	2.85	45.38	460.02	949.37	5.26	5.54	7	0.57
4200	3.13	43.54	437.52	889.51	4.57	5.17	8	0.61
3900	3.50	41.65	412.52	821.74	3.76	4.80	9	0.66
3600	4.04	39.69	383.93	743.17	2.83	4.43	10	0.72
3200	5.34	36.99	336.17	611.73	1.30	3.94	13	0,.81

3.6. Analysis of the influence of the air temperature flowing into the evaporator on the required heat exchange surface

During the analysis of the computational model, the influence of the assumed inlet air temperature was also examined. The calculation results for such assumptions are presented in Table 9.

Table 9. Inlet air temperature influence on the heat transfer parameters

$T_{p1}, °C$	A_w, m^2	$\alpha_p, \frac{W}{m^2K}$	$k_{Aw}, \frac{W}{m^2K}$	$\alpha_o, \frac{W}{m^2K}$	$T_{p2}, °C$	$w_o, \frac{m}{s}$	N_{rz}, -	ε, -
18	2.15	43.54	493.59	1156.64	7.57	5.17	5	0.52
17	2.40	43.54	477.14	1070.15	6.57	5.17	6	0.55
16	2.72	43.54	458.61	981.24	5.57	5.17	7	0.58
15	3.13	43.54	437.52	889.51	4.57	5.17	8	0.61
14	3.67	43.54	413.19	794.39	3.57	5.17	9	0.65
13	4.45	43.54	384.61	695.11	2.57	5.17	11	0.70
12	5.61	43.54	350.24	590.40	1.57	5.17	14	0.75
11	7.59	43.54	307.40	478.07	0.57	5.17	19	0.80

3.7. Analysis of the impact of the refrigerant used on the required heat transfer surface

As part of the system analyses, the influence of the refrigerant properties on selected system parameters was also checked. In the base case, the calculations were performed for the R134a factor, but for the given temperatures it will also be appropriate to perform an analysis, e.g. for the R32 factor, for the same evaporation temperature. The physical properties of the refrigerant, similarly to the base case, were taken from the program for simulating refrigeration processes - Solkane. The results of calculations of individual parameters for both factors are summarized in table 10.

Table 10. The influence of the selected refrigerant on heat transfer parameters

$Refrigerant$	A_w, m^2	$\alpha_p, \frac{W}{m^2K}$	$k_{Aw}, \frac{W}{m^2K}$	$\alpha_o, \frac{W}{m^2K}$	$T_{p2}, °C$	$w_o, \frac{m}{s}$	N_{rz}, -	ε, -
R134a	3.13	43.54	437.52	889.51	4.57	5.17	8	0.61
R32	2.47	43.54	552.89	1544.92	4.57	5.23	6	0.61

It can be noticed that when using the R32 factor, the heat transfer intensity is higher, which reduces the required heat transfer surface. This reduces the number of tube rows needed.

3.8. Analysis of the impact of taking into account the condensation of moisture from the air on the required heat exchange surface

When calculating the heat exchanger, it should be also pay attention to the air temperatures obtained as a result of heat exchange, because when cooling the air, it is often also dried, i.e. the process of latent heat exchange occurs. This results from exceeding the so-called dew point temperature, which causes moisture to condense from the air. The dew point temperature for a given temperature and relative humidity can be read from Molière's chart. The calculations of the base model were made without taking this phenomenon into account. The RCJ coefficient responsible for taking this phenomenon into account in the calculations was equal to 1. In order to examine the impact of this phenomenon on the obtained results, the value of the RCJ coefficient must first be determined. This can be done with the following algorithm:

$$RCJ = 1 + RCP \frac{X_{p1} - X_p''}{T_{p1} - T_z} \tag{19}$$

where for $T_z \geq 0\ °C$:

$$RCP = \frac{25.016 \cdot 10^5 - 4.1868 \cdot 10^3 (T_z + 273.15)}{1006 + 1857\, X_p''} \tag{20}$$

while for $T_z < 0\ °C$:

$$RCP = \frac{28.351 \cdot 10^5 - 2.05 \cdot 10^3 (T_z + 273.15)}{1006 + 1857\, X_p''} \tag{21}$$

The moisture content X_p'' in saturated air at the evaporator surface temperature T_z can be determined by the formula:

$$X_p''(T_z) = 0.622\, \frac{p_w''}{p_a - p_w''} \tag{22}$$

Where is the partial pressure of water vapor p_w'':

$$p_w'' = 610.7 \cdot 10^B \tag{23}$$

and the coefficient B for $T_z \geq 0\ °C$:

$$B = \frac{T_z}{31.6639 + 0.131305 T_z + 2.63247 \cdot 10^{-5} {T_z}^2} \tag{24}$$

and for $T_z < 0\ °C$:

$$B = \frac{T_z}{27.9541 + 0.103141 T_z + 9.30422 \cdot 10^{-6} {T_z}^2} \tag{25}$$

To determine the weighted average surface temperature of the external surface of the evaporator T_z it is necessary to first calculate the temperature of the base of the fins T_{fp} and the average temperature of the entire fin T_{fm}. This is described by the relationships:

$$T_{fp} = T_o + \frac{\dot{Q}}{A_w} \left(\frac{1}{\alpha_o} + \frac{\delta_r}{\lambda_r} \frac{d_w}{d_m} + R_z \right) \tag{26}$$

$$T_{fm} = \underline{T_p} - \varepsilon_f (\underline{T_p} - T_{fp}) \tag{27}$$

For the base case results of the above, the value of the RCJ coefficient is equal to 1.03. The influence of this coefficient was taken into account in the calculations of individual parameters, and the results obtained are summarized in Table 11 with the results when moisture condensation was not taken into account.

Table 11. A summary of the results of individual system parameters, taking into account the condensation of moisture from the air and without taking it into account

$RCJ, -$	A_w, m^2	$\alpha_p, \frac{W}{m^2K}$	$k_{Aw}, \frac{W}{m^2K}$	$\alpha_o, \frac{W}{m^2K}$	$T_{p2}, °C$	$w_o, \frac{m}{s}$	N_{rz}, -	ε, -
1.00	3.13	43.54	437.52	889.51	4.57	5.17	8	0.61
1.03	3.00	44.85	447.74	916.21	4.87	5.17	8	0.60

4. Conclusions

By comparing all the analyzes parameters, it is possible to determine which change of parameters or their inclusion in the calculations will lead to the optimization of the results obtained in the base example. Table 12 shows how an increase in the value of input parameters affects the output parameters obtained from the computational model (the analysis of the factor change and the impact of condensation of moisture from the air are also taken into account).

Table 12. Table showing how to change parameters input affects the output parameters, where: ↗ - value increases, ↘ - value decreases, = - value does not change.

		Output parameter							
		A_w	α_p	k_{Aw}	α_o	T_{p2}	N_{rz}	ε	ε_f
		$[m^2]$	$\left[\frac{W}{m^2 \cdot K}\right]$	$\left[\frac{W}{m^2 \cdot K}\right]$	$\left[\frac{W}{m^2 \cdot K}\right]$	[°C]	[−]	[−]	[−]
Input parameter	Wall thickness δ_r ↗	↘	=	↗	↗	=	↗	=	=
	Fin thickness δ_f ↗	↘	↗	↗	↗	=	↘	=	↗
	Blade pitchi t ↗	↗	↗	↘	↘	=	↗	↘	↘
	Air flow $V_p^{\cdot}$ ↗	↘	↗	↗	↗	↗	↘	↘	↘
	Temp. Inlet T_{p1} ↗	↘	=	↗	↗	↗	↘	↘	↘
	Change to factor R32	↘	=	↗	↗	=	↘	↘	=
	Accounting for moisture condensation(RCJ>1)	↘	↗	↗	↗	↗	↘	↘	↘

Based on the calculations performed for the given input parameters, the required heat exchange area was 3.13 m^2, which, given the assumed geometric dimensions of the exchanger, would require the use of 8 rows of tubes. The analysis of the change in tube thickness showed that by decreasing the tube thickness, the required heat exchange surface increases, but because the internal diameter of the tube increases, the internal surface area of the tube assembly also increases.In this way, by reducing the thickness of the pipe from 0.001 m to 0.0003 m, the required surface area increased from 3.13 m^2 to 3.54 m^2, while the internal surface area of the pipes increased so much that the heat exchange surface needed to achieve the assumed efficiency will be achieved using 7 rows of tubes, not 8 rows as in the initial case. This analysis can have clear financial benefits for manufacturers, as this analysis reduced the amount of tubing needed by 12.5% compared to the base case.

However, the presented results of the system cannot be the final determinant when selecting the tube wall thickness, because reducing this thickness adversely affects the mechanical properties of the tubes, so the operating conditions in which a given evaporator will operate are also very important, especially the working pressure of the refrigerant.

Increasing the lamella thickness in the presented range is also an optimization compared to the value obtained in the base case. By increasing the fin thickness from 0.00012 m to 0.0002 m, for example, the heat exchange surface needed to maintain the efficiency of the evaporator is reduced from 3.13 m^2 to 2.95 m^2, which allows, as in the case of reducing the tube wall thickness, to save one row pipes. Here, however, it is important to take air temperatures into account when determining the thickness of the slats. When air temperatures are negative, the calculations should also take into account the formation of frost on the external surfaces of the exchanger, which may lead to problems with free air flow through the evaporator. To prevent this, there are appropriate methods of defrosting the evaporator, e.g. with electricity or hot water [4,25]

Increasing the pitch of the fins, in turn, affects the number of fins along the length of the exchanger, which, provided that the length and height of the exchanger are to remain constant, only increases the required surface area and the number of rows necessary to be used. In practical conditions, it is also possible to examine the influence of the inclination of the lamellas in the direction of flow, if there were to be fewer of them.

Changing the air flow rate compared to the assumed ones is a more complex phenomenon because, apart from changing the required heat exchange surface and the number of rows of tubes, the efficiency of the exchanger also changes. This analysis may be important in the design, but above all in the operation of the exchanger (evaporator). The same dependencies apply to the increase in the temperature of the air flowing into the exchanger. The higher the inlet temperature, the smaller the heat exchange surface needed to meet the required efficiency of the exchanger, while the efficiency decreases significantly. For the air temperature in the base case (15^{o}C), the efficiency of the exchanger is approximately 61%. By increasing the inlet air temperature by 1 degree, the required area decreases from 3.13 m^2 to 2.72 m^2, but at the expense of a drop in efficiency to approximately 58%. Using the prepared computational model, it was checked what the minimum air flow rate could be while maintaining 8 rows of pipes to ensure the highest efficiency of the entire evaporator. From the analysis, the maximum value of the evaporator efficiency obtained while maintaining 8 rows of pipes and the construction parameters from the base example is approximately 65% with an air flow of 3840 m^3/h. The air velocity reaching the front surface of the exchanger in this case is 2.7 m/s. It is worth noting that this speed should not exceed 4.0 m/s, because beyond this limit the air flow resistance increases, which is accompanied by an undesirable increase in the noise emitted by the evaporator. Increased flow also promotes heat dissipation.

Changing the refrigerant from R134a to R32, assuming the same boiling point, resulted in a reduction in the required heat exchange area from 3.13 m^2 to 2.47 m^2, which in turn also reduces the target number of necessary rows from 8 to 6, i.e. by 25%. However, with this assumption, hydraulic flow calculations should also be made to prove whether the pipe diameters adopted in the base example are suitable for working with R32, because to achieve the mentioned boiling point of R32 at -2^{o}C , the required working pressure of the medium is 7.62 bar, where for the R134a factor it is 2.72 bar - almost three times less. Changing the factor is also often required due to the impact of a given factor on the environment, so many comparative analyzes of factors in given operating conditions are performed to select the appropriate one and optimize the operation of a given evaporator.

Another necessary parameter when considering the design of the evaporator and testing its operation is to take into account the condensation of moisture from the cooled air. There is a phenomenon of latent heat exchange, which, as shown by one of the above analyses, affects the results obtained.For the given parameters from the initial example, the RCJ coefficient taking into account the process of moisture condensation was 1.04. Correction of the calculations taking this factor into account resulted in a reduction of the calculated required heat exchange area from 3.13 m^2 to 2.93 m^2, which also resulted in a reduction in the number of necessary rows of pipes from 8 to 7. However, it should be noted that the condensation of moisture from the cooled air also reduces the efficiency of the exchanger. This analysis may lead to the conclusion that failure to take moisture condensation into account may lead to oversizing of the exchanger or to incorrect results of the expected cooling effect, for example the air temperature at the evaporator outlet.

The algorithm used to determine the necessary heat transfer area of a lamella heat exchanger to maintain the set efficiency can serve as a basis both for the design of new exchangers, but also for the analysis of already used solutions. As the analysis showed, for the base example, in order to reduce the necessary heat exchange surface and the size of the exchanger (number of rows of pipes), one could, for example, reduce the thickness of the pipe wall or increase the thickness of the fins. The analysis showed that the assumed initial value of the set air flow while maintaining the geometry of the exchanger can be reduced from 4200 m^3/h to 3840 m^3/h, which will improve the efficiency of the exchanger from about 58% to about 61%. An important conclusion from the conducted research is also the fact of moisture condensation, which must be included in the calculations if you want to obtain correct results. In each case, however, one should remember about additional limitations (e.g. hydraulic flow), which should be taken into account in the obtained results.

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