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SHELL AND TUBE HEAT EXCHANGER – THE HEAT TRANSFER AREA DESIGN PROCESS

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Abstract: Nowadays, the operating nuclear reactors are able to utilise only 1 % of mined out uranium. An effective exploitation of uranium, even 60 %, is possible to achieve in so-called fast reactors. These reactors commercial operation is expected after the year 2035. Several design configurations of these reactors exist. Fast reactors rank among the so-called Generation IV reactors. Helium-cooled reactor, as a gas-cooled fast reactor, is one of them. Exchangers used to a heat transfer from a reactor active zone (i.e. heat exchangers) are an important part of fast reactors. This paper deals with the design calculation of U-tube heat exchanger (precisely 1-2 shell and tube heat exchanger with U-tubes): water – helium.

KEYWORDS: Fast reactor, nuclear reactor, Generation IV reactors, GFR, shell and tube heat exchanger, U-tube heat exchanger, helium, DHR, decay heat removal loop.

1 Introduction

A cooling is an integral part of the reactors. In the case of gas-cooled fast reactors (GFR), a refrigerant – helium (He) temperature normally moves from 400 °C to 1000 °C. Helium ensures two roles in the GFR. First, primary, role is to ensure a heat transfer from the nuclear reactor active zone by a main heat exchanger. Second role is to ensure a decay heat removal from the reactor active zone at its unplanned shutdown. In this second case, the decay heat will be leaded away by the heat exchanger for decay heat removal (EDHR). EDHR is a part of the so-called decay heat removal loop. In order to perform these roles by helium, it is necessary to choose a suitable heat exchanger for the given purpose and conditions, and subsequently to realise the design calculations of chosen heat exchanger. The research activities in the given area from several aspects are documented in the various publications, as for example [1 to 7].

2 Design parameters and choice of the heat exchanger for de-cay heat removal

Based on the previously mentioned and on the heat transfer characteristics commonly available in literature [8, 9], it is possible to propose a suitable heat exchanger for a given application. First, it is necessary to select a heat exchanger type and then its design variant. In this case, U-tube heat exchanger was selected. This one belongs to the group of the shell and tube heat exchangers (i.e. the recuperative heat exchangers). The operational parameters of the proposed heat exchanger and its design parameters are listed in table 1 and 2.

The design parameters, unlike the data listed in table 1, were not primarily specified. These have been properly chosen (table 2). However, these data can be chosen if they are not specified, only then, when it is selected a particular design variant of the heat exchanger. In this case, the pressure drops have not been specified. If pressure drop is not

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given explicitly, it is commonly considered in technical practice with the pressure drop value corresponding to (5 - 10) % of the inlet pressure. Nevertheless, there are many cases when pressure drop plays a decisive role. In such cases, a heat exchanger is optimised to maintain the specified value of pressure drop.

In general, there are different design variants within some heat exchanger type. In the case of shell and tube heat exchanger, there are four main design arrangements: with fixed tubesheets, with fixed tubesheets and expansion joint in the shell, with U-tubes and with floating head.

Tab. 1 Operational parameters of the heat exchanger for the decay heat removal.

- heat flow rate ($\dot{Q}$): 240 000 W	- type and design variant of the heat exchanger: primarily unknown
- cooled (hot) medium: helium	- helium temperature at the inlet: 520 °C (T_{h1} = 793,15 K)
- heated (cold) medium: untreated water	- helium temperature at the outlet: 250 °C (T_{h2} = 523,15 K)
- helium operating pressure: 7 MPa	- water temperature at the inlet: 15 °C (T_{s1} = 288,15 K)
- water operating pressure: 0,6 MPa	- water temperature at the outlet: 40 °C (T_{s2} = 313,15 K)

Tab. 2 Design parameters of the heat exchanger for the decay heat removal.

- variant construction: U-tube heat exchanger	- angle arrangement of the tubes in the bundle: 45 °
- passes number on the tube side: 2	- baffles: single segmental baffles
- passes number on the shell side: 1	- sealing strips: none
- tubes outer diameter (d_2): 25,4 mm	- wall thickness of the tubes: 2,77 mm

The heat exchanger type selection as well as its design variant choice depends on the various factors [8, 9]. The U-tube heat exchanger has been chosen because others are considered to be unsuitable in terms of the given data. This choice follows from the analysis of the shell and tube heat exchanger design variants as it is shown below.

Table 1 shows, that the difference between mean temperatures of the streams is 357,5 °C. It means, that the heat exchanger with fixed tubesheets cannot be used because it is not fulfilled the condition concerning the material temperatures from which the shell and tubes are made. The condition expresses, if a hot medium flows in the tube side, then the difference between mean temperature of the shell wall and mean temperature of the tube wall should not exceed the value approximately equal to 40 °F (22,2 °C) [10]. Otherwise, when a hot medium flows in the shell side and a cold medium in the tube side, then the difference between the mean temperature of the walls is to be no more than 150 °F (83,3 °C) [10]. The heat exchanger with fixed tubesheets and expansion joint in the shell could be used, but this alternative is significantly limited by a high price of the suitable expansion joint. From the mentioned above, it follows that two variants of the shell and tube heat exchanger are still usable. However, there is an additional requirement that the heat exchanger should be perfectly tight. The heat exchanger tightness is an important condition because the heat exchanger is to serve as an equipment for heat transfer from the reactor active zone. Only the U-tube heat exchanger can provide this condition. In addition, the floating head heat exchanger cannot be used for pressure greater than 6,3 MPa [8, 9]. Based on the above, the U-tube heat exchanger, with flows arrangement: one-shell pass and two-tube passes (i.e. 1-2 shell and tube heat exchanger with U-tubes), seems to be the most suitable alternative.

It should also be pointed out that the heat exchanger main dimensions (i.e. its maximum diameter and length) need not be specified. In this case, a size of the heat transfer area is determined generally such that the calculations for the specified heat exchanger are made for several alternatives. However, the design parameters, which can be independently chosen, do not change in these calculations. The chosen design parameters, which have been considered as constant for the design of the heat exchanger, are listed in table 2. It also applies that helium will flow in the tubes of this heat exchanger. This is because a medium with a higher operating pressure should flow preferentially in the tubes of the shell and tube heat exchangers.

3 Transport and thermodynamic properties of helium and water

In the calculation of a heat exchanger, it is necessary to know the properties of heat transfer mediums. These properties are necessary for determining the size of a heat transfer area. There are the relationships for the mediums properties calculation in table 3 and 4 [11, 12]. If necessary, the values of the other quantities (e.g. c_p) can be determined by quantities listed in tables 3 and 4.

Tab. 3 Relationships for determination of the transport and thermodynamic properties of helium. (Note: In the relationships, T is in Kelvin and p is in bar.)

- density: $\rho = 48{,}14\dfrac{p}{zT} = 48{,}14\dfrac{p}{T}\left(1+0{,}4446\,pT^{-1{,}2}\right)^{-1}$ [kg/m³]
- dynamic viscosity: $\eta = 3{,}674.10^{-7}T^{0{,}7}$ [Pa.s]
- thermal conductivity: $\lambda = 2{,}682.10^{-3}\left(1+1{,}123.10^{-3}p\right)T^{\left(0{,}71\left(1-2.10^{-4}p\right)\right)}$ [W/(m.K)]
- Prandtl number: $\Pr = \dfrac{\eta c_p}{\lambda} = \left(\dfrac{0{,}7117}{1+1{,}123.10^{-3}p}\right)T^{-\left(0{,}01-1{,}42.10^{-4}p\right)}$ [1]

Tab. 4 Relationships for determination of the transport and thermodynamic properties of water. (Note: In the relationships, t is in degrees Celsius.)

- density: $\rho = 1000{,}1234+9{,}6221.10^{-3}t-5{,}6947.10^{-3}t^2+$ $+1{,}7345.10^{-5}t^3-3{,}0362.10^{-8}t^4$ [kg/m³]
- dynamic viscosity: $\eta = \exp\left(7{,}455-3{,}0636.10^{-2}t+1{,}853.10^{-4}t^2-\right.$ $\left.-7{,}7189.10^{-7}t^3+1{,}823.10^{-9}t^4-1{,}7938.10^{-12}t^5\right)10^{-6}$ [Pa.s]
- thermal conductivity: $\lambda = \left(58{,}206+16{,}633.10^{-2}t-6{,}24.10^{-4}t^2+\right.$ $\left.+1{,}0463.10^{-7}t^3\right)10^{-2}$ [W/(m.K)]
- Prandtl number: $\Pr = \eta c_p/\lambda = 0{,}1706\exp\left(0{,}228+2{,}02.10^{-3}t\right)^{-1}$ [1]

4 The heat transfer area calculation of the decay heat removal exchanger

The heat losses to the surroundings are normally neglected ($\dot{Q}_S = 0$) in the specification of heat exchanger size [9]. Then the following enthalpy balance equation can be written for a heat exchanger with two input and two output streams:

$$\dot{Q} = \dot{m}_h\left(h_{h1}-h_{h2}\right) = \dot{m}_s\left(h_{s2}-h_{s1}\right). \tag{1}$$

Equation (1) is valid provided that the mass flow-rates are constant and $\dot{Q}_S = 0$. However, the quantity $\dot{Q}$ can also be expressed by general rate equation of heat transfer:

$$\dot{Q} = \bar{k}A\Delta\bar{T} = \bar{k}A\left(\bar{T}_h - \bar{T}_s\right). \tag{2}$$

This equation is valid for all geometric con-figurations of the heat exchangers. The size of the heat transfer area (A) is generally determined by equation (2). In the shell and tube heat exchanger, the quantity A usually represents the overall outside surface of tubes, i.e. $A = A_2$ in equation (2).

In the literature, the equation (2) is commonly written in the form: $\dot{Q} = \bar{k}A\varepsilon_{\Delta\bar{T}}\Delta\bar{T}_{\ln}$, wherein $\Delta\bar{T}_{\ln}$ (also known as *LMTD*; $LMTD = \Delta\bar{T}_{\ln}$) is given by equation:

$$\Delta\bar{T}_{\ln} = \frac{\dot{Q}}{\bar{k}A\varepsilon_{\Delta\bar{T}}} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1/\Delta T_2\right)} = \frac{\left(T_{h1} - T_{s2}\right) - \left(T_{h2} - T_{s1}\right)}{\ln\left[\left(T_{h1} - T_{s2}\right)/\left(T_{h2} - T_{s1}\right)\right]}. \tag{3}$$

The value of quantity $\varepsilon_{\Delta\bar{T}}$ can also be expressed by the relationship: $\varepsilon_{\Delta\bar{T}} = \dfrac{\Delta\bar{T}}{\Delta\bar{T}_{\ln}}$ resulting from equations (2) and (3). In a heat exchanger, quantity $\varepsilon_{\Delta\bar{T}}$ represents the deviation of $\Delta\bar{T}$ value from maximum possible value of $\Delta\bar{T}$ which equals the value of $\Delta\bar{T}_{\ln}$.

Equation (1) can be written in the form: $\dot{Q} = \dot{m}_h\bar{c}_{ph}\left(T_{h1} - T_{h2}\right) = \dot{m}_s\bar{c}_{ps}\left(T_{s2} - T_{s1}\right)$ or in the form that defines the quantity *R*, i.e.:

$$\dot{m}_s\bar{c}_{ps}\big/\dot{m}_h\bar{c}_{ph} = \bar{C}_{ps}\big/\bar{C}_{ph} = \left(T_{h1} - T_{h2}\right)/\left(T_{s2} - T_{s1}\right) = R. \tag{4}$$

The so-called weak stream, i.e. a flow in which $\bar{C}_{pi}$ is smaller ($\bar{C}_{p\min} = \min\left\{\bar{C}_{ph}, \bar{C}_{ps}\right\}$), decides about a heat exchanger behaviour [8, 12]. This is the reason why the equation (2) in the literature [8, 13] rewrites by $\bar{C}_{p\min}$ into the form:

$$\bar{k}A\big/\bar{C}_{p\min} = \dot{Q}\big/\left(\bar{C}_{p\min}\Delta\bar{T}\right) = \dot{Q}\big/\left(\bar{C}_{p\min}\varepsilon_{\Delta\bar{T}}\Delta\bar{T}_{\ln}\right) = NTU_{\min}. \tag{5}$$

Equation (5) defines quantity $NTU_{\min}$ which represents the so-called number of transfer units. This quantity is usually quoted without index min, i.e. $NTU = NTU_{\min}$ [8, 13].

There are also other variables which are defined by $\bar{C}_{p\min}$ and used to describe the heat exchangers. In a heat exchanger, the weak stream can be theoretically heated (possibly cooled) to the inlet temperature of second (stronger) stream [14]. In the first case, it would apply the equality: $T_{s2} = T_{h1}$ (fig. 1a), and in the second case: $Th_{s2} = T_{s1}$ (fig. 1b). In such cases (i.e. in a perfect heat transfer) the heat flow rate of the heat exchanger would be maximum ($\dot{Q} = \dot{Q}_{\max}$). In the meaning of fig. 1, $\dot{Q}_{\max}$ can be expressed by relationship:

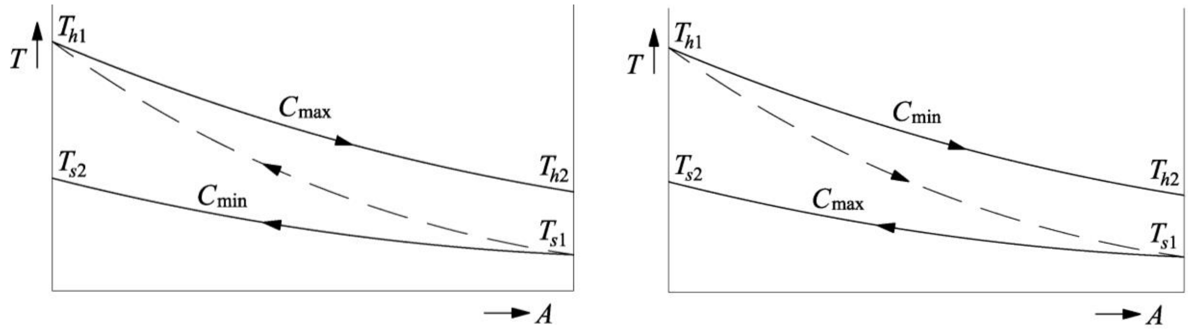
$$\dot{Q}_{\max} = \pm\bar{C}_{p\min}\left(T_{\max 1} - T_{\min 1}\right) = \bar{C}_{p\min}\left(T_{h1} - T_{s1}\right). \tag{6}$$

In equation (6), sign + is applicable for cases corresponding to the fig. 1a, and sign – for cases illustrated in fig. 1b.

The ratio of the actual heat flow rate ($\dot{Q}$) to the maximum possible heat flow rate ($\dot{Q}_{\max}$) defines the so-called the heat exchanger efficiency (Φ):

$$\Phi = \dot{Q}/\dot{Q}_{\max} = \left[\overline{C}_{ph}\left(T_{h1} - T_{h2}\right)\right]/\left[\overline{C}_{p\min}\left(T_{h1} - T_{s1}\right)\right] = \left[\overline{C}_{ps}\left(T_{s2} - T_{s1}\right)\right]/\left[\overline{C}_{p\min}\left(T_{h1} - T_{s1}\right)\right]. \quad (7)$$

Equation (7) may also be written in the form: $\Phi = \overline{C}_{ph}P_h/\overline{C}_{p\min} = \overline{C}_{ps}P_s/\overline{C}_{p\min}$ which determines two quantities P_h and P_s. In the calculations of the heat exchangers, the quantity P_s. is used most often. This quantity is normally denoted only by the symbol P [13] and is expressed by equation: $P = P_s = \left(T_{s2} - T_{s1}\right)/\left(T_{h1} - T_{s1}\right)$.



a) b)

Fig. 1 The temperature profile along the heat transfer area of the heat exchanger with the two inlet and the two outlet streams. (Note: Dashed lines represent ideal heat exchange.)

In equation (7), the quantities $\overline{C}_{ph}$ and $\overline{C}_{ps}$ can be replaced by the symbol $\overline{C}_{p\max}$. The inverse of the ratio $\overline{C}_{p\max}/\overline{C}_{p\min}$ is denoted by the symbol C (i.e. $C = \dfrac{\overline{C}_{p\min}}{\overline{C}_{p\max}}$) in the literature. The indexes s and h can also be replaced by min or max for other quantities (see e.g. equation (6)). On this basis and with reference to fig. 1, equation (1) can be transformed into the form:

$$\dot{Q} = \overline{C}_{ph}\left(T_{h1} - T_{h2}\right) = \overline{C}_{ps}\left(T_{s2} - T_{s1}\right) = \pm\overline{C}_{p\min}\left(T_{\min 2} - T_{\min 1}\right). \quad (8)$$

If the symbol $\dot{Q}$ in equation (5) is replaced by equation (8) so the following equation will be obtained:

$$\Delta\overline{T} = \dot{Q}/\left(\overline{C}_{p\min}NTU_{\min}\right) = \pm\left(T_{\min 2} - T_{\min 1}\right)/NTU_{\min}. \quad (9)$$

In equation (9), the signs + and – have the same meaning as in equation (6). Equation (9) can be rewritten into the form: $\dfrac{\Delta\overline{T}}{T_{\max 1} - T_{\min 1}} = \dfrac{\pm\left(T_{\min 2} - T_{\min 1}\right)}{\left(T_{\max 1} - T_{\min 1}\right)NTU_{\min}}$ [8] or with reference to fig. 1 as:

$$\theta = \frac{\Delta\overline{T}}{T_{h1} - T_{s1}} = \frac{\varepsilon_{\Delta\overline{T}}\Delta\overline{T}_{\ln}}{T_{h1} - T_{s1}} = \frac{\left(T_{\min 2} - T_{\min 1}\right)}{\left(T_{\max 1} - T_{\min 1}\right)NTU_{\min}} = P_{\min}/NTU_{\min}. \quad (10)$$

Equation (10) defines the quantity $P_{\min} = \left(T_{\min 2} - T_{\min 1}\right)/\left(T_{\max 1} - T_{\min 1}\right)$ which is often denoted by the symbol ε, possibly $\varepsilon_{\min}$ (i.e. $P_{\min} = \varepsilon_{\min} = \varepsilon$) [8].

The quantities thus defined (i.e. R, P, C, $P_{\min}$ and $NTU_{\min}$) have a specific meaning for the heat transfer description and calculations of the heat exchangers. As example, it can be mentioned that the following equation allows to determine the quantity $\varepsilon_{\Delta\bar{T}}$ on the basis of C, $NTU_{\min}$ and $P_{\min}$ values:

$$\varepsilon_{\Delta\bar{T}} = \ln\left(\frac{1-CP_{\min}}{1-P_{\min}}\right) \Big/ \left(NTU_{\min}(1-C)\right). \tag{11}$$

This equation arises from equations (3) and (10), while variables in equation (3) are expressed by quantities C and $P_{\min}$. Equation (11) is valid for any heat exchanger. The values of variables C and $P_{\min}$ can be determined relatively easily, however, the value of $NTU_{\min}$ cannot be determined easily anymore. Nevertheless, the value of $NTU_{\min}$ can be calculated from another independent equation. Such equation can be obtained based on the differential equations related to the differential section of a heat exchanger. In the literature, such equations are normally presented in the form of diagrams. The stationary operation of the 1-2 shell and tube heat exchanger with U-tubes can be described by the following equation [15, 16, 17]:

$$2/P_{\min} = (1+C) + \left(\sqrt{C^2+1}\right)\left(\frac{1+\exp\left(-NTU_{\min}\sqrt{C^2+1}\right)}{1-\exp\left(-NTU_{\min}\sqrt{C^2+1}\right)}\right). \tag{12}$$

Equation (12) allows to determine the value of $NTU_{\min}$ and then also the values of other variables (i.e. θ, $\varepsilon_{\Delta\bar{T}}$ and $\Delta\bar{T}$) that are included in equation (10). If the values of $NTU_{\min}$ and $\varepsilon_{\Delta\bar{T}}$ are correctly specified so these have to also correspond with the values calculated by equation (11) which is documented by fig. 2. The so-called mean value of the overall heat transfer coefficient ($\bar{k}$) is the last quantity which need to be known for the calculation of the value A_2. This follows from equation (2). The inverse of the quantity $\bar{k}$ represents the total thermal resistance (R_c) to heat transfer from a hot stream of fluid to a cold one. In the recuperative heat exchangers, R_c is given by the equation: $R_c = \frac{1}{\bar{k}} = R_h + R_{zh} +$ $+ R_w + R_{zs} + R_s$. In some special cases (e.g. in heat transfer through the planar or cylindrical wall) the values of variables R_h, R_{zh}, R_w, R_{zs} and R_s can be determined by other variables. The heat transfer area in the current heat exchanger is represented by the tubes outside surface. This means that equation for the calculation of the quantity $\bar{k}$ may be written by [18] in the form:

$$\frac{1}{\bar{k}} = \frac{d_2}{d_h\bar{\alpha}_h} + \frac{d_2 r_{fh}}{d_h} + \frac{d_2 \ln\left(\frac{d_2}{d_1}\right)}{2\bar{\lambda}} + \frac{d_2 r_{fs}}{d_s} + \frac{d_2}{d_s\bar{\alpha}_s} = R_h + R_{zh} + R_w + R_{zs} + R_s \tag{13}$$

Equation (13) determines the relations for calculation of the quantities R_h, R_{zh}, R_w, R_{zs} and R_s.

In practical calculations, the quantities $\bar{\alpha}_h$ and $\bar{\alpha}_s$ in equation (13) are calculated from the Nusselt number (Nu). The values of Nu are determined from the criterion relations [8, 9, 15, 17] valid for the given design arrangement of the heat exchanger, the hydrodynamic conditions and fluids properties. The values of the quantities r_{fh} and r_{fs} may obtain for some fluids in literature [10, 11]. In this case, it will be assumed that r_{fs} = 0.00035 m^{2}K/W and r_{fh} = 0.0 m^{2}K/W.

There is one more unknown quantity in the equation (13). It is the mean value of the thermal conductivity of the wall material ($\bar{\lambda}$) from which the heat transfer area is made. The material from which the heat exchanger is to be made, is unknown. It will be assumed that the tubes are made from an austenitic stainless steel for which $\bar{\lambda} =$ 15 W/(mK). It will also be supposed that the absolute roughness of tubes, which form the tube bundle of the heat exchanger, is 0.04 mm (i.e. e = 0.04 mm).

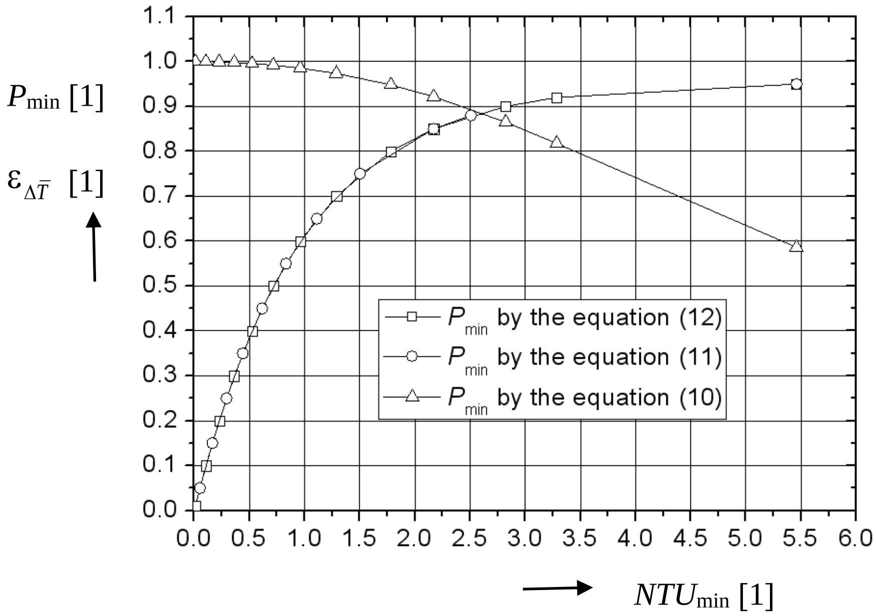


Fig. 2 The dependence of P_{min} and $\varepsilon_{\Delta\bar{T}}$ on NTU_{min} compiled from the data listed in tables 1, 3 and 4, and assuming that T_{s2}-T_{s1}= 25 K

5 Results of calculations

Thermal and hydraulic calculations of EDHR have been performed by means of the computational program. This program was developed on the basis of knowledge presented above and by the equations given in the literature [9, 15, 17]. Calculations have been performed for various values of D_i and S/D_i (see figs. 3 and 5). The values of quantities A_2 and $\Delta p = \Delta p_r + \Delta p_m$, as well as the values of other variables describing EDHR in terms of its design, are basic output from these calculations.

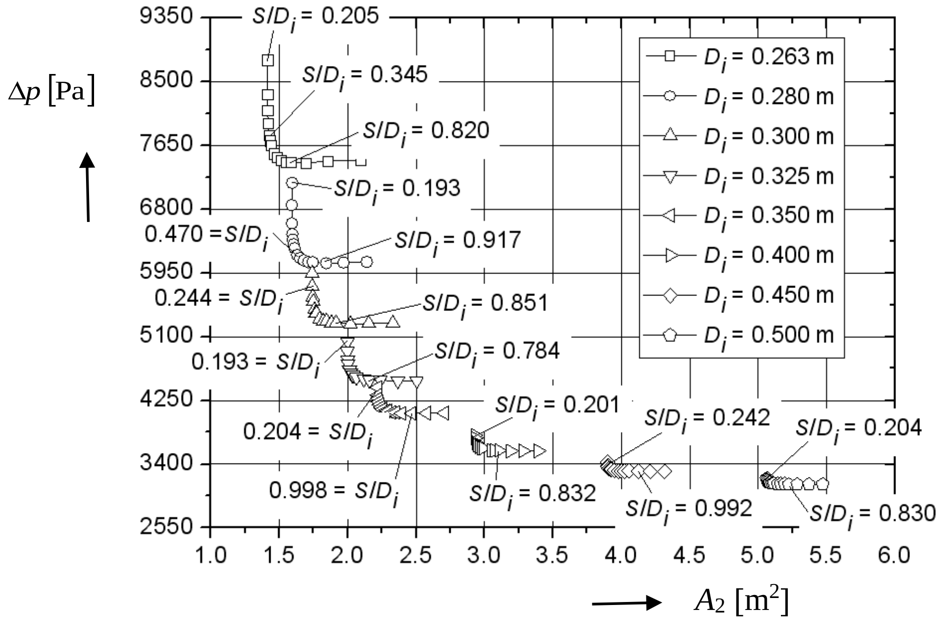


Fig. 3 The dependence of total pressure drop (Δp) on the heat transfer area A_2

Fig. 3 shows that total pressure drop (Δp) decreases with the increasing value of D_i which also applies in case of ratio S/D_i. It is necessary to say that the number of tubes forming a tube bundle increases with the increasing value of D_i and the length of the heat exchanger decreases. The number of tubes is always constant for given D_i. It may be noted that straight length of the tubes, in case of U-tubes the active length of the tube bundle (l_r), should range from $3D_i$ to $15D_i$ by [2] (i.e. $l_r \in \langle 3D_i; 15D_i \rangle$). The baffle spacing ($S$) (fig. 4) is the most important design parameter for which the following relation applies: $S/D_i \in \langle 0{,}2; 0{,}8 \rangle$ [9]. For illustration, fig. 3 also shows the calculation results for other values of ratio S/D_i. The height of baffle cut (H) is another optional variable in the design of the heat exchanger (fig. 4). The value of this variable should be selected by [1] in order that a value of ratio H/D_i falls into a range of the values from 0,2 to 0,35 (i.e. $H/D_i \in \langle 0{,}2; 0{,}35 \rangle$). However, the value H should be chosen preferably so that S_0 (fig. 4) achieves approximately 25 % of the value $S_k = \pi D_i^2/4$ [19].

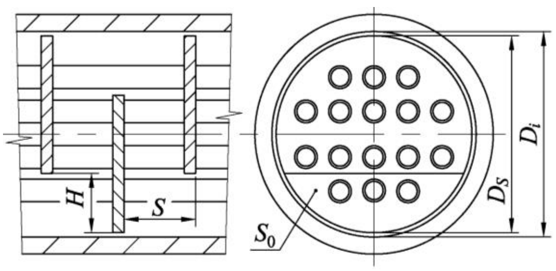


Fig. 4 The design dimensions of the single segmental baffles of the heat exchanger

Fig. 5 shows the dependence of the tube-side pressure drop (Δp_r) on the area A_2 provided that $H/D_i = 0{,}295$. In fig. 5, the dependence for helium is plotted. Similar dependence would also be plotted for water. Such dependencies are important for cases where the pressure drops must not exceed the permissible values. These dependencies allow to choose the value of quantity D_i. It is again necessary to consider the ratio of variables l_r and D_i that should be between the values 3 and 15 (i.e. $l_r/D_i \in \langle 3;15 \rangle$). By [9], if $l_r < 3D_i$, it is necessary to choose the smaller tubes diameter forming the tube bundle, in case that $l_r > 15D_i$, it should be considered with several heat exchangers connected in series.

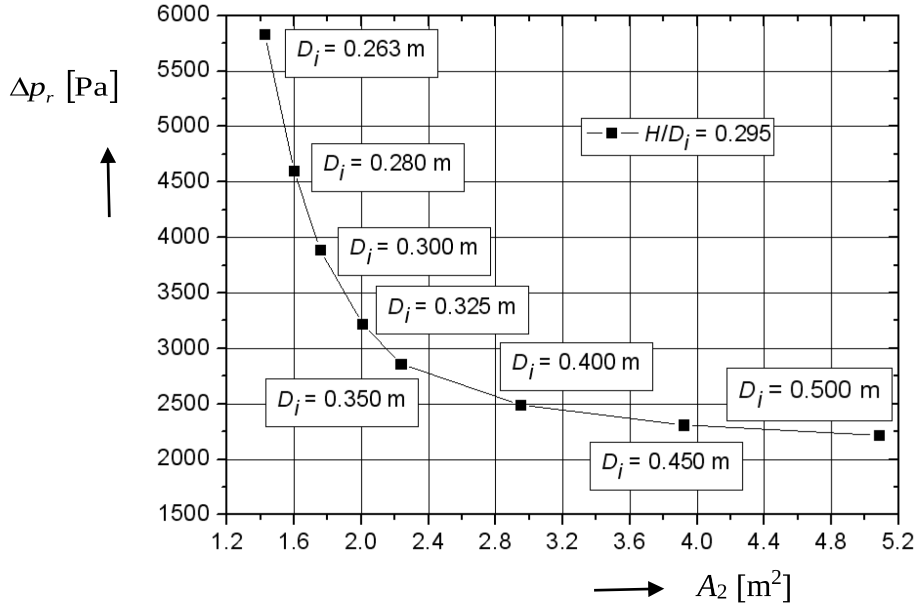


Fig. 5 The dependence of the tube-side pressure drop (Δp_r) on the heat transfer area A_2

6 CONCLUSION

The paper documents which variables are necessary to know and which ones have to be suitably chosen for the heat transfer area design of the recuperative heat exchanger. Then within a design variant of the heat exchanger, the different values of the heat transfer area and the pressure drops can be obtained for the given heat flow rate. The values of the variables presented in this article by means of graphs are a necessary basis for the correct choice of the suitable alternative of the heat exchanger. In the design of equipment, it cannot be forgotten to perform the economic calculations that complement the thermal and hydraulic calculations. The economic calculations always decide which variant of equipment is the most advantageous (in this case, the equipment is the heat exchanger). However, it is necessary to remember that economical aspect may not always be decisive for the design of some equipment. The field of nuclear energetic is such example where is required a high safety and a reliability of the equipments operation. After the determination of the optimal alternative of equipment (the heat exchanger), the strength calculations have to be realised. On their basis, if it is necessary, the design dimensions of the heat exchanger can still be changed. This means that the final variant of the heat exchanger is obtained after the strength calculations because these can influence some dimensions of the heat exchanger. The final variant of the heat exchanger, if its dimensions have been changed after strength calculations, is to be still verified by thermal and hydraulic calculations.

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SYMBOLS

Latin Letters

A	Heat transfer area [m^2]
d	Tube diameter [m]
D_i	Shell inside diameter [m]
D_S	Baffle diameter [m]
e	Tube absolute roughness [m]
C	Heat capacity rate ratio [1]
c_p	Specific heat capacity at constant pressure [J/(kgK)]
C_p	Heat capacity rate at constant pressure [W/K]
H	Height of baffle cut [m]
k	Overall heat transfer coefficient [W/(m^{2}K)]
$LMTD$	Logarithmic mean temperature difference [K]
l_r	Active length of the tube bundle [m]
$\dot{m}$	Mass flow rate [kg/s]
h	Specific enthalpy [J/kg]
NTU	Number of transfer units [1]
Nu	Nusselt number [1]
p	Operating pressure [Pa]
P	Dimensionless temperature change [1]
Pr	Prandtl number [1]
R	Heat capacity rate ratio [1]
R	Thermal resistance [m^{2}K/W]
r_f	Fouling factor [m^{2}K/W]
R_w	Thermal resistance of the wall [m^{2}K/W]
R_z	Thermal resistance due to fouling [m^{2}K/W]
S	Baffle spacing [m]
S_0	Baffle window area [m^2]
S_k	Cross-sectional area of the heat exchanger [m^2]
T	Temperature [K]
$\dot{Q}$	Heat flow rate [W]
$\dot{Q}_s$	Heat loss rate [W]
z	Compressibility factor [1]

Greek Letters

α	Heat transfer coefficient [W/(m^{2}K)]
Δp	Total pressure drop [Pa]
Δp_r	Tube-side pressure drop [Pa]
Δp_m	Shell-side pressure drop [Pa]
ΔT	Temperature difference [K]
$\Delta \bar{T}_{\ln}$	Logarithmic mean temperature difference [K]
ε	Dimensionless temperature change [1]
$\varepsilon_{\Delta \bar{T}}$	Logarithmic mean temperature difference correction factor [1]
Φ	Heat exchanger efficiency [1]
η	Dynamic viscosity [Pa.s]
λ	Thermal conductivity [W/(mK)]
θ	Dimensionless mean temperature difference [1]
ρ	Density [kg/m^3]

Subscripts

c	Total
h	Hot medium, On the side of the hot medium
max	Maximum, Stronger stream
min	Minimum, Weak stream
1	At the inlet, Inner

i	ith medium	2	At the outlet, Outer, Outside surface of the tubes
s	Cold medium, On the side of the cold medium	$\bar{}$	Mean value

Acronyms

DHR	Decay heat removal	GFR	Gas-cooled fast reactor
EDHR	Heat exchanger for decay heat removal	He	Helium

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