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A NEW APPROACH TO THE PREDICTION AND DESIGN OF SHELL AND TUBE HEAT EXCHANGERS

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Abstract. *In this paper, an iterative procedure for thermo-hydraulic calculation of shell and tube heat exchangers according to prescribed pressure drop, has been presented. From this procedure it is possible to predict the heat exchanger geometries for which one can expect in advance to satisfy the thermo-hydraulic conditions of project. It remains to the designer to pay a full attention to the analysis of possible solutions and to the choice of optimal heat exchanger geometry.*

Key words: *shell and tube heat exchanger, design, iterative procedure*

1. INTRODUCTION

Designing of a Shell-and-Tube Heat Exchanger (STHE) can be treated in a few subsequent phases:

- Geometric Designing;
- Checking;
  - thermo-hydraulic calculation;
  - mechanical calculation;
  - techno-economic calculation;
- Optimization;
- Simulation.

Designing is determining the heat exchanger geometry enabling the heat exchange rate between hot and cold fluid, in the frame of the given operating conditions of apparatus.

By checking one can investigate whether the HE of defined geometry (shell diameter, tube diameter, length of tubes, number and arrangement of tubes in bundle, number of passes for shellside and tubeside fluid, number of baffles, ...) can perform the heat exchange between hot and cold fluid for prescribed pressure drop (bounded by allowed pressure drop) or not, i.e. is it possible to reach wanted temperature variation of fluids in given apparatus.

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The aim of optimization is to adopt such a heat exchanger which could be able to perform the basic function and also be reliable in operation with satisfying economic criteria.

By simulation is possible the modeling of the given HE, i.e. the description of the apparatus behavior in the case when the independent process variables are specified, for instance the evaluation of the outlet fluid parameters on the basis of the known inlet fluid parameters.

The designing can be reduced to the systematic analysis of the numerous possible heat exchanger geometries in order to find the apparatus in which the effective heat exchange can be done in the frame of given operating conditions. The design begins by choosing of the starting HE geometry. For chosen starting geometry, the checking of HE by thermo-hydraulic calculation has to be done. From thermo-hydraulic calculation one can obtain:

- the amount of heat has to be exchanged in given apparatus;
- the amount of heat which can be exchanged in the apparatus of given geometry;
- the tubeside pressure drop;
- the shellside pressure drop.

On the basis of results obtained from thermo-hydraulic calculation the estimation of designing has to be performed. If in given HE can not be performed the specified heat exchange between hot and cold fluid or if one or both pressure drops are greater than allowed ones then the apparatus does not satisfy. In that case the geometry has to be changed, i.e. the first greater apparatus (greater heat exchange surface) is to be checked. If one or both pressure drops are much less than allowed ones, the smaller exchangers have to be checked until the pressure drops in HE reach maximal allowed pressure drops. It is to be accepted the smallest HE, i.e. the HE of smallest heat exchange surface by which the heat exchange can be done under pressure drops so much the closer to the allowed values. If there are no geometry satisfying these conditions, there is the possibility of shell coupling in:

- serial, and
- parallel connection.

Pressure drops for fluids flowing through Shell-and-Tube Heat Exchangers, have significant influence on heat exchange. The greater pressure drops the greater fluid velocities leading to the greater heat transfer coefficients and to the smaller dirt. It follows that the greater pressure drops in HE are in correlation to the smaller needed surface for heat transfer. So, in principle to achieve a more effective heat exchange, the pressure drop in HE should be as much as possible closer to the allowed pressure drop.

The tubeside pressure drop is the most influenced by fluid velocity. If the number of tubefluid passes is increased N times, for the same tube length and number of tubes, the tubeside pressure drop will increase N^3 times. The shellside pressure drop is influenced by many parameters as: dimension and shape of baffles, baffle spacing, dimension of baffle windows, arrangement of tubes and tube pitch.

The allowable pressure drop is the maximal pressure drop that may be achieved in section of apparatus where the heat exchange is performing. The values of the allowed pressure drops occurring in the practice, for defined operating pressures in HEs, are given in Table 1 and in Figure 1.

Table 1. Recommended values of allowed pressure drops

Operating pressure (absolute)	Allowed pressure drop
to 1bar	1/10 of operating pressure (absolute)
1 ÷ 2 bar	1/2 of operating pressure (manometric)
over 2bar	0,35bar and greater

The allowed pressure drops are separately defined for shellside fluid and tubeside fluid.

The extreme fluid velocities can lead to erosion of apparatus surfaces (due to the presence of solid particles in fluids) or to the vibration of tubes in bundle, those values of pressure drops must be avoided.

The HEs with small pressure drops are ineffective, since they need great heat transfer area. The main reason is: the small fluid velocities resulting in small values of heat transfer coefficients.

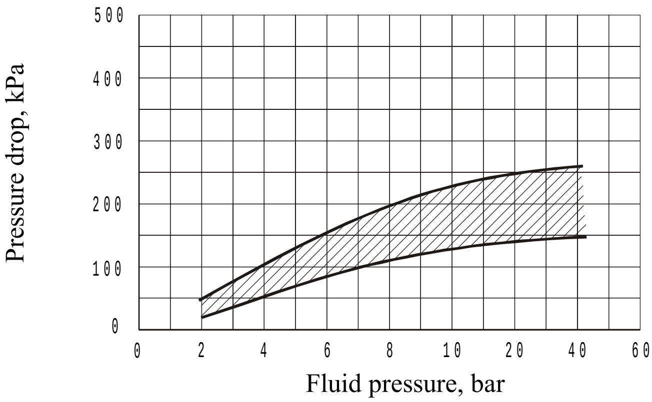


Fig 1. Recommended values of allowed pressure drop

On the basis of the above mentioned, one can conclude that the entire utilizing of pressure drops for both fluids can be performed in the apparatus with minimal heat transfer area and maximal fluid velocities.

It is a common practice to assume, in the first step for the starting geometry of HE, the tube fluid velocity to be within 1-1,5m/s.

In this paper is presented the algorithm for determining the heat transfer area of HE in which the allowed pressure drops will be realized for shellside and tubeside fluids.

2. METHODOLOGY FOR DESIGNING OF STHES

For designing of a heat exchanger it is necessary for a designer to have at disposition the following data:

- service of apparatus;
- space limitations in operation;
- orientation of apparatus;

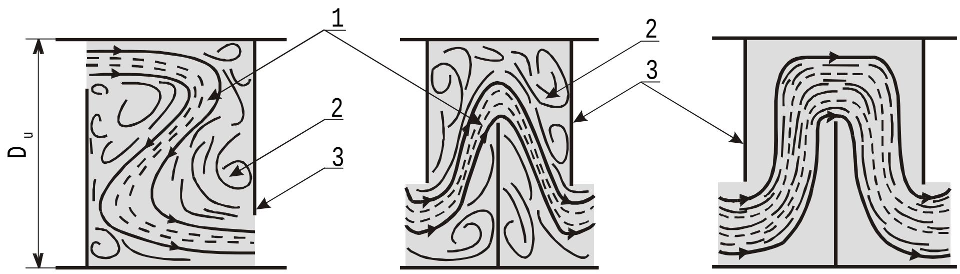
- mass flow rate ($\dot{m}'$) and hot fluid temperatures (t_1' , t_2');
- mass flow rate ($\dot{m}''$) and cold fluid temperatures (t_1'' , t_2'');
- process fluids operating pressures (p' , p'');
- allowed pressure drops ($\Delta p_{all}'$, $\Delta p_{all}''$).

In reference [3] the detailed designing methodology for m-n shell and tube heat exchangers based on Bell's procedure for determining of both shellside heat transfer coefficient and shellside pressure drop, is presented. In fact the basis for the Bell's method is the model suggested by Tinker (1951), later modified by Palen and Taborek (1969).

The fluid flow on the shellside of a STHE with segmental baffles is much more complicated than the tube or channel flow (Fig. 2), and consequently the determination of shellside heat transfer coefficient and pressure drop is more complicated.

An idealized scheme of shellside flow is shown in Fig. 3. One can notice the regions with longitudinal and cross flow with respect to tube bundle.

The main idea of this model is based on the fact that due to resistance to the main flow B (Fig. 4) the bypass streams are occurring. It is necessary to point out that in contrary to other streams, the stream E does not take part in heat transfer.



1 - Main flow, 2 - Eddies, 3 - Baffle

Fig. 2. Shellside flow shame in a STHE

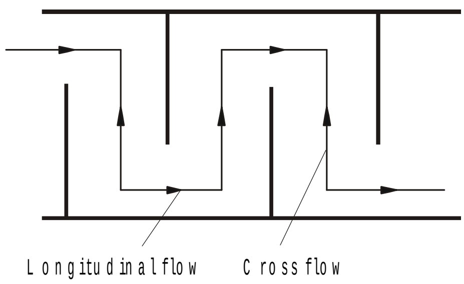


Fig. 3. Idealized flow scheme

Bell's method is based on the determination of shellside heat transfer coefficient in a HE without bypass (or leakage) flows (so called ideal tube bundle) and afterward the correction is done by using factors taking into account the influence of bypass flows.

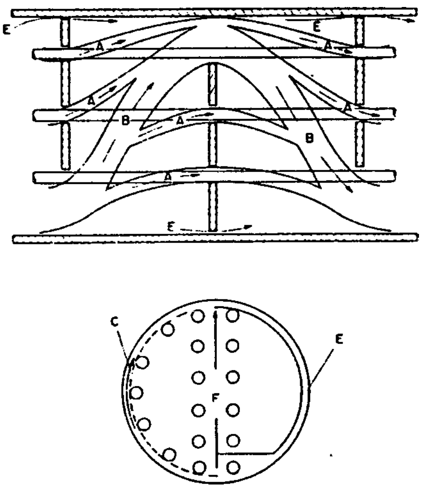


Fig. 4. Real shellside flow stream configuration

The basic equation for calculating the effective average shellside heat transfer coefficient α_o is as follows:

$$\alpha_o = (f_o \cdot f_{AE} \cdot f_{CF} \cdot f_{PK} \cdot f_{lam}) \cdot \alpha_{oid} = f_\alpha \cdot \alpha_{oid} \qquad (1)$$

where:

α_{oid} - heat transfer coefficient for pure cross-flow in an ideal tube bank

f_α - the overall correction factor, which for well designed STHE is typically of the order of 0,6÷0,9.

The total pressure shellside drop over the exchanger is given by:

$$\Delta p_o = \Delta p_{ps} + \Delta p_{ok} + \Delta p_{pk} + \Delta p_{lok} \qquad (2)$$

where:

Δp_{ps} - pressure drop in the interior cross flow sections from baffle tip to baffle tip,

Δp_{ok} - pressure drop in the window sections,

Δp_{pk} - pressure drop in the entrance and exit sections, and

Δp_{lok} - pressure drop due to resistance at headers.

On the similar manner as for heat transfer, the correction factors for calculation of shellside pressure drop, are taking into account the influence of bypass flows.

Although the correction factors vary in wide ranges, depending on HE configuration, the total pressure drop in usual design is for 20÷30% less than ideal bundle pressure drop.

In the most procedures for HE designing, the choice of starting geometry is of subjective nature. Consequently, one can not estimate in advance whether the chosen starting geometry satisfy thermo-hydraulic conditions or not.

Also in the most procedures [7] for calculation of HEs the needed area for heat transfer is determined on the basis of the assumed overall heat transfer coefficient value. The question is how to choose this value from so wide range (f.i. k = 300-900 W/m^{2}K).

2.1. Polley's algorithm

Having in mind the above Polley and co-workers [6] (1991) have been developed the calculation algorithm for HEs according to the allowed pressure drop, based on Kern and Bell procedures. According to Polley's conclusion the determination of heat transfer area based on Bell's procedure is very complex. In that case the coefficients in the expression for the shellside pressure drop depend on the correction factors for shellside heat transfer coefficient, the correction factors for shellside pressure drop, the shell geometry, the tube bundle geometry as well as on the thermo-physical properties of fluid. In this paper the Polley's algorithm derived on the bases of Kern's procedure, is shown.

The basic equations in the algorithm are derived under the following conditions:

- there is no phase change in given HE for operating fluids;
- operating fluids are liquids with small viscosity;
- HE is with one pass of shellside fluid;
- for baffling of shellside fluid the segmental baffles are used;
- tubeflow is a fully developed turbulent flow ($Re_u > 10000$);
- shellside fluid flow is turbulent and fully developed ($Re_o > 100$);
- pressure drops of fluids are equal to allowed pressure drops;
- pressure drop due to local resistance is neglected.

Thermal power of HE

$$\dot{Q} = k \cdot A \cdot \Delta t_{sr} \tag{3}$$

Overall heat transfer coefficient

$$k = \frac{1}{(\frac{1}{\alpha_u} + R_u) \cdot \frac{d_s}{d_u} + \frac{d_s}{2 \cdot \lambda_z} \cdot \ln \frac{d_s}{d_u} + (\frac{1}{\alpha_o} + R_o)} \tag{4}$$

Heat transfer area

$$A = d_s \cdot \pi \cdot L \cdot N_c \tag{5}$$

Number of tubes in tube bundle

$$N_c = \frac{4 \cdot n \cdot \dot{m}_u}{\rho_u \cdot \pi \cdot d_u^2 \cdot w_u} \tag{6}$$

Tube bundle

Re- number for tubeside fluid

$$\mathrm{Re}_u = \frac{w_u \cdot \rho_u \cdot d_u}{\mu_u} \tag{7}$$

Pr- number for tubeside fluid

$$\mathrm{Pr}_u = \frac{c_{pu} \cdot \mu_u}{\lambda_u} \tag{8}$$

Nu- number for tubeside fluid

$$Nu_u = \frac{\alpha_u \cdot d_u}{\lambda_u} \tag{9}$$

For determination of tubeside heat transfer coefficient for STHEs very often the Dittus-Boelter equation has been used

$$Nu_u = 0{,}023 \cdot \mathrm{Re}_u^{0,8} \cdot \mathrm{Pr}_u^{\ 1/3} \cdot \left(\frac{\mu_u}{\mu_{uz}} \right)^{0,14} \tag{10}$$

where μ_{uz} - dynamic viscosity of tubeside fluid for the mean temperature of inner tube surface. In the previous calculation it is assumed to be $(\mu_u/\mu_{uz})^{0,14} = 1$.

If from the equations (1) - (11), the number of tubes in tube bundle, length of tubes and average velocity of tubeside fluid, shall be eliminated, one can obtain, the relation between tubeside pressure drop and tubeside heat transfer coefficient as well as heat transfer area, in the folowing form

$$\Delta p_u = \Delta p_{uall} = C_u \cdot A \cdot \alpha_u^{3,5} \tag{11}$$

where

$$C_u = \frac{1}{0{,}023^{2,5}} \cdot \frac{d_u^{1,5} \cdot \mu_u^{11/6}}{d_s \cdot \dot{m}_u \cdot \rho_u \cdot \lambda_u^{7/3} \cdot c_{pu}^{7/6}} \cdot \left(\frac{\mu_u}{\mu_{uz}} \right)^{-0,63} \tag{12}$$

and the average tubeside fluid velocity

$$w_u = \frac{\mu_u}{\rho_u \cdot d_u} \cdot \left(\frac{\alpha_u \cdot d_u}{0{,}023 \cdot \lambda_u} \right)^{1,25} \cdot \left(\frac{\lambda_u}{c_{pu} \cdot \mu_u} \right)^{0,4167} \cdot \left(\frac{\mu_u}{\mu_{uz}} \right)^{-0,175} \tag{13}$$

Shell

Equivalent diameter

$$d_e = \frac{2\sqrt{3} \cdot t^2 - \pi \cdot d_s^2}{\pi \cdot d_s} \quad \text{- for triangular tube layout} \tag{14}$$

$$d_e = \frac{4 \cdot t^2 - \pi \cdot d_s^2}{\pi \cdot d_s} \quad \text{- for square tube layout} \tag{15}$$

Number of segmental baffles

$$N_p = \frac{L}{L_p} - 1 \tag{16}$$

Shell diameter

$$D_u = \sqrt{\frac{4 \cdot C \cdot t^2 \cdot N_c}{\pi}} \tag{17}$$

where $C = 1$ for square tubelayout and $C = \sqrt{3}/2$ for triangular layout of tubes.

Area of the characteristic cross section for shellside fluid

$$A_o = \frac{\dot{m}_o}{\rho_o \cdot w_o} = L_p \cdot (t - d_s) \cdot \frac{D_u}{t} \tag{18}$$

Re - number for shellside fluid

$$Re_o = \frac{w_o \cdot \rho_o \cdot d_e}{\mu_o} \tag{19}$$

Pr - number for shellside fluid

$$Pr_o = \frac{c_{po} \cdot \mu_o}{\lambda_o} \tag{20}$$

Nu - number for shellside fluid

$$Nu_o = \frac{\alpha_o \cdot d_e}{\lambda_o} = 0{,}36 \cdot \mathrm{Re}_o^{0,55} \cdot \mathrm{Pr}_o^{1/3} \cdot \left(\frac{\mu_o}{\mu_{oz}}\right)^{0,14} \tag{21}$$

Shellside pressure drop

$$\Delta p_o = 4 \cdot 0{,}4475 \cdot \mathrm{Re}_o^{-0,19} \cdot \frac{D_u}{d_e} \cdot (N_p + 1) \cdot \frac{\rho_o \cdot w_o^2}{2} \cdot \left(\frac{\mu_o}{\mu_{oz}}\right)^{-0,14} \tag{22}$$

By eliminating the number of baffles, the baffle spacing, the tube length, the number of tubes in bundle, and the average shellside fluid velocity from equations (3), (12)-(20), one can obtain the relation between the shellside pressure drop and the shellside heat transfer coefficient as well as the heat transfer area

$$\Delta p_o = \Delta p_{oall} = C_o \cdot A \cdot \alpha_o^{5,1} \tag{23}$$

where

$$C_o = \frac{67 \cdot C \cdot t \cdot (t - d_s) \cdot d_e^{1,1} \cdot \mu_o^{1,3}}{\rho_o \cdot \dot{m}_o \cdot d_s \cdot \lambda_o^{3,4} \cdot c_{po}^{1,7}} \cdot \left(\frac{\mu_o}{\mu_{oz}}\right)^{-0,854} \tag{24}$$

and where the average shellside flow velocity is given by:

$$w_o = \frac{\mu_o}{\rho_o \cdot d_e} \cdot \left(\frac{\alpha_o \cdot d_e}{0{,}36 \cdot \lambda_o}\right)^{1/0,55} \cdot \left(\frac{c_{po} \cdot \mu_o}{\lambda_o}\right)^{-1/1,65} \cdot \left(\frac{\mu_o}{\mu_{oz}}\right)^{-0,14/0,55} \tag{25}$$

The coefficients C_u and C_o depend on tube diameter tube pitch, thermo-physical properties and mass flow rates of process fluids.

From equations (1), (2), (9) and (21) one can obtain:

$$C_1 \cdot A^{1/5,1} + C_2 \cdot A^{1/3,5} + C_3 \cdot A + C_4 = 0 \tag{26}$$

where

$$C_1 = \left(\frac{C_o}{\Delta p_{oall}}\right)^{1/5,1} \tag{27}$$

$$C_2 = \left(\frac{C_u}{\Delta p_{uall}}\right)^{1/3,5} \cdot \frac{d_s}{d_u} \tag{28}$$

$$C_3 = -\frac{\Delta t_{sr}}{\dot{Q}} \tag{29}$$

$$C_4 = R_u \cdot \frac{d_s}{d_u} + R_o + \frac{d_s}{2 \cdot \lambda_z} \cdot \ln \frac{d_s}{d_u} \tag{30}$$

Finally, one obtains the non-linear equation (24) with respect to the heat transfer area, which can be solved by using computer.

According to the conclusion of the same author the equation (21) is not satisfactory exact. Because of that Polley suggested for shellside pressure drop the following relation:

$$\Delta p_o = \Delta p_{oall} = C_{op} \cdot A \cdot \alpha_o^{4,412} \tag{31}$$

where the coefficient C_{op} involves the correction factor for shellside heat transfer as a safety factor, which for well designed HE is 0,6.

2.2 Iterative procedure for calculation of HEs

Polley's algorithm based Kern's procedure for thermo-hydraulic calculation of HEs, is very simple and convinient for rapid determination of the heat exchanger geometry enabling the effective heat exchange between process fluids with full utilization of pressure drops. The accepted HE geometry has to be checked according to Bell's procedure for thermo-hydraulic calculation.

The disadvetage of this algorithm (baside the well known disadvantages of Kern's procedure) is first of all that in this algorithm the pressure drop due to local resistances in tube bundle is neglected, which in HE's with multy tube passes can be of the same order of magnitude as the pressure drop due to friction. Secondly, in this algorithm it is not taken into account the decrease of number of tubes with increasing the number of tube passes [2, 3] (equation 15). Anyway, for the designer is also the problem that he has no control on the intermediate results. From this algorithm result often the complet unappropriate geometries of HEs, for instance small number of tubes of great length, or flow velocities much greater of recommended ones ... The reason for that is in the fact that Polley and co-workers have assumed the fluid pressure drops equal to allowed ones.

Having in mind the aboue facts, in this paper an itterative procedure for determining the set of HE geometries satisfying thermo-hydraulic conditions of project, according to prescribed pressure drop, has been developed:

$$\begin{aligned}\Delta p_u &= \Delta p_{uprip} = \Delta p_{utr} + \Delta p_{ulok} = \\ &= 4 \cdot f_{utr} \cdot \frac{L}{d_u} \cdot n \cdot \frac{\rho_u \cdot w_u^2}{2} \cdot \left(\frac{\mu_u}{\mu_{uz}}\right)^{-0,14} + \frac{1}{2} \cdot \rho_u \cdot \sum (\xi_{ui} \cdot w_{ui}^2) \le \Delta p_{uall}\end{aligned} \tag{32}$$

$$\Delta p_o = \Delta p_{oprip} \le \Delta p_{oall} \tag{33}$$

where

ξ_{ui} - local resistance coefficient

w_{ui} - mean fluid flow velocity at the position of local resistance.

Besides return losses in bonnets of HE for 180^o, in local losses are also included the losses due to:

- inlet header;
- outlet header;
- inlet into tubes of tube bundle;
- outlet from tubes of tube bundle;
- flow return for 90^o in bonnet chambers.

Iterative procedure is based on the fact that for the specified apparatus thermal power (apparatus thermal power is given by project task) exists a hyperbolic relation between overall heat transfer coefficient and heat transfer area.

In this paper one part of this itterative procedure is shown that can be applied for rapid determining of the satisfactory STHE geometry of type 1-1 where does not occur phase change of process fluids. In order that the obtained results from the iterative procedure could be compared to the results from Polley's algorithm, the same equations for heat transfer coefficients and pressure drops have been used, (with a small difference, since the pressure drop due to local resistances in tube bundle has been added to the pressure drop due to friction). It is also assumed that the velocity through header is equal to average fluid velocity in tube. Local resistance coefficients are taken from the literature [3].

The itterative procedure has the following steps

1. Assuming the type of STHE and arrangements of process fluids;
2. Assuming the thermo-physical propertis (c_p, ρ, μ, λ) of process fluids;
3. Determination of the HE thermal power

$$\dot{Q} = \dot{m}' \cdot c_p' \cdot (t_1' - t_2') = \dot{m}'' \cdot c_p'' \cdot (t_2'' - t_1'') \tag{34}$$

4. Determination of mean temperature difference

$$\Delta t_{sr} = \frac{(t_1' - t_2'') - (t_2' - t_1'')}{\ln \frac{t_1' - t_2''}{t_2' - t_1''}} \tag{35}$$

5. Assuming the thermal resistances due to fouling (R_u, R_o), tube conductivity (λ_z), tube geometry ($\phi d_s/d_u$), tube pitch (t), tube arrangement and equivalent diameter (d_e) (Eq. 12, 13);

6. Calculation of constants in itterative procedure

$$C_\lambda = R_u \cdot \frac{d_s}{d_u} + R_o + \frac{d_s}{2 \cdot \lambda_z} \cdot \ln \frac{d_s}{d_u} \tag{36}$$

$$C_k = \frac{\dot{Q}}{\Delta t_{sr}} \tag{37}$$

$$C_{wu} = 111{,}6 \cdot d_u^{0,25} \cdot \frac{\mu_u^{0,583}}{\rho_u \cdot \lambda_u^{0,833} \cdot c_{pu}^{0,417}} \left(\frac{\mu_u}{\mu_{uz}}\right)^{-0,175} \tag{38}$$

$$C_{wo} = 6{,}4 \cdot d_e^{0,818} \cdot \frac{\mu_o^{0,394}}{\rho_o \cdot \lambda_o^{1,212} \cdot c_{p0}^{0,606}} \left(\frac{\mu_o}{\mu_{oz}}\right)^{-0,2555} \tag{39}$$

$$C_{pu1} = \frac{12464{,}7}{\dot{m}_u} \cdot \frac{d_u^{1,5}}{d_s} \cdot \frac{\mu_u^{1,833}}{\rho_u \cdot \lambda_u^{2,333} \cdot c_{pu}^{1,167}} \cdot \left(\frac{\mu_u}{\mu_{uz}}\right)^{-0,63} \tag{40}$$

$$C_{pu2} = \frac{1}{2} \cdot \rho_u \cdot C_{lok} \cdot C_{wu}^2 \tag{41}$$

where C_{lok} = 4 for HE with straight tubes and C_{lok} = 2,5 for HE with U-tubes

$$C_{po} = \frac{52{,}7}{\dot{m}_o} \cdot \frac{d_e^{1,1} \cdot (t - d_s)}{t \cdot d_s} \cdot \frac{\mu_o^{1,3}}{\rho_o \cdot \lambda_o^{3,4} \cdot c_{po}^{1,7}} \cdot \left(\frac{\mu_o}{\mu_{oz}} \right)^{-0,854} \tag{42}$$

7. Pressure drop in tube bundle

$$\begin{aligned} \Delta p_{u\,\boldsymbol{j}} &= C_{\boldsymbol{j}} \cdot \Delta p_{uall} \\ C_{\boldsymbol{j}} &= C_{\boldsymbol{j\text{-}1}} - \Delta C, \quad \boldsymbol{j} = 2;3;4;... \end{aligned} \tag{43}$$

$\Delta C = const$, constant step in decreasing of pressure, for $\boldsymbol{j} = 1$ $C_1 = 1$

8. Set of HE geometries is determined by itterative procedure (described in continuation of the text) where the initial (starting) values of heat transfer area is given

$$A_{\boldsymbol{j,i}} = A_{\boldsymbol{j,i\text{-}1}} + \Delta A, \quad \boldsymbol{j}, \boldsymbol{i} = 2;\ 3;\ 4;\ .... \tag{44}$$

$\Delta A = const$, for defined increase of area, for $\boldsymbol{i} = 1$ $A_{j,1} = \Delta A$

9. Calculation of overall heat transfer coefficient

$$k_{\boldsymbol{j,i}} = \frac{C_k}{A_{\boldsymbol{j,i}}} \tag{45}$$

10. Calculation of tubeside heat transfer coefficient

$$\alpha_{u\,\boldsymbol{j,i}} = f(\Delta p_{u\,\boldsymbol{j}}, A_{\boldsymbol{j,i}}, C_{pu1}, C_{pu2}) \tag{46}$$

where

$$\Delta p_{u\,\boldsymbol{j}} = C_{pu1} \cdot A_{\boldsymbol{j,i}} \cdot \alpha_{u\,\boldsymbol{j,i}}^{3,5} + C_{pu2} \cdot \alpha_{u\,\boldsymbol{j,i}}^{2,5} \tag{47}$$

11. Calculation of average fluid velocity in tubes

$$w_{u\,\boldsymbol{j,i}} = C_{wu} \cdot \alpha_{u\,\boldsymbol{j,i}}^{1,25} \tag{48}$$

Limitations $\quad w_{u\,\boldsymbol{j,i}} \geq \dfrac{10000 \cdot \mu_u}{\rho_u \cdot d_u}$ and $w_{u\,\boldsymbol{j,i}} \leq w_{uprep}$

12. Calculation of number of tubes in bundle

$$N_{c\,\boldsymbol{j,i}} = \frac{4 \cdot \dot{m}_u}{\rho_u \cdot \pi \cdot d_u^2} \cdot \frac{1}{w_{u\,\boldsymbol{j,i}}} \tag{49}$$

13. Calculation of tube length

$$L_{\boldsymbol{j,i}} = \frac{A_{\boldsymbol{j,i}}}{d_s \cdot \pi \cdot N_{c\,\boldsymbol{j,i}}} \tag{50}$$

Limitation $\quad L_{\boldsymbol{j,i}} \leq L_{prep}$

14. Calculation of shell diameter

$$D_{u\,\boldsymbol{j,i}} = \sqrt{\frac{4 \cdot C \cdot t^2 \cdot N_{c\,\boldsymbol{j,i}}}{\pi}} \tag{51}$$

15. Calculation of shellside heat transfer coefficient

$$\alpha_{o\,j,i} = \frac{1}{\dfrac{1}{k_{j,i}} - \dfrac{1}{\alpha_{u\,j,i}} \cdot \dfrac{d_s}{d_u} - C_\lambda} \tag{52}$$

16. Calculation of shellside pressure drop

$$\Delta p_{u\,j,i} = C_{po} \cdot \frac{D^2_{u\,j,i}}{N_{c\,j,i}} \cdot A_{j,i} \cdot \alpha^{5,1}_{o\,j,i} \tag{53}$$

Limitation $\Delta p_{o\,j,i} \le \Delta p_{oall}$

17. Calculation of average shellside fluid flow velocity

$$w_{o\,j,i} = C_{wo} \cdot \alpha^{1,818}_{o\,j,i} \tag{54}$$

Condition $w_{o\,j,i} \ge \dfrac{100 \cdot \mu_o}{\rho_o \cdot d_e}$

18. Calculation of characteristic flow cross section for shellside fluid

$$A_{o\,j,i} = \frac{\dot{m}_o}{\rho_o \cdot w_{o\,j,i}} \tag{55}$$

19. Calculation of baffle spacing

$$L_{p\,j,i} = \frac{t \cdot A_{o\,j,i}}{(t - d_s) \cdot D_{u\,j,i}} \tag{56}$$

Condition $0,2 \cdot D_{u\,j,i} \le L_{p\,j,i} \le D_{u\,j,i}$

20. Calculation of number of baffles

$$N_{p\,j,i} = \frac{L_{j,i}}{L_{p\,j,i}} - 1 \tag{57}$$

The whole range of dependence $k = f(A)$ is practically investigated by this iterative procedure, for prescribed tubeside pressure drop. The disadvantage of this method is in the procedure of determination of both shellside heat transfer coefficient and shellside pressure drop. Namely, this method does not introduce the influence of bypass flows, as well as the influence of heat transfer direction.

3. Conclusion

By presented procedure (with using of computer) one can get the set of geometries satisfying the thermo-hydraulic conditions of the project. In such a way the task of a designer is now much easier. The designer has the complete control on the intermediate results of calculation. By imposing extra conditions it is possible to minimize the number of possible solutions. The main aim of the designer should be the analysis and the choice of one of many satisfactory heat exchanger geometries.

Nomenclature

A	- area, m^2
c_p	- specific heat capacity, J/kgK
C	- constant
d, D	- diameter, m
k	- overall heat transfer coefficient, W/m^2K
L	- length, m
m	- number of shellside fluid passes
$\dot{m}$	- mass flow rate, kg/s
n	- number of tubeside fluid passes
N	- number of tubes (baffles)
p	- pressure, Pa
Δp	- pressure drop, Pa
$\dot{Q}$	- thermal power, W
R	- resistance due to fouling, m^2K/W
t	- temperature, ^oC
Δt	- temperature difference, ^oC
w	- velocity, m/s
α	- heat transfer coefficient, W/m^2K
λ	- thermal conductivity, W/mK
μ	- dynamic viscosity, Pa s
ρ	- density, kg/m^3

Index

c	- tube, tube bundle
$dozv$	- allowed
e	- equivalent
o	- shell
p	- baffle
$prep$	- recommended
$prip$	- prescribed
s	- outer
sr	- average
u	- inner
z	- wall
$'$	- hot fluid
$''$	- cold fluid
1	- apparatus inlet
2	- apparatus outlet

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NOVI POSTUPAK ZA PRORAČUN I PROJEKTOVANJE DOBOŠASTIH IZMENJIVAČA TOPLOTE

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U radu je prikazan jedan iterativni postupak za termo-hidraulički proračun dobošastih izmenjivača toplote prema pripisanom padu pritiska. Ovim postupkom moguće je predvideti set geometrija izmenjivača toplote za koje se unapred može očekivati da će zadovoljiti termo-hidrauličke uslove projekta. Na ovaj način projektanti punu pažnju posvećuju analizi mogu}ih rešenja i izboru optimalne geometrije izmenjivača toplote.

Ključne reči: *dobošasti izmenjivač toplote, proračun, iterativni postupak*