Plate heat exchanger design software for industrial and educational applications

Nikola R. Zlatković, Divna M. Majstorović, Mirjana Lj. Kijevčanin, Emila M. Živković

Faculty of Technology and Metallurgy, University of Belgrade, Belgrade, Serbia

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Abstract

Plate heat exchanger is a type of heat exchanger that uses corrugated metal plates to transfer heat between two fluids. The plate corrugations are designed to achieve turbulence across the entire heat transfer area thus producing the highest possible heat transfer coefficients while allowing close temperature approaches. Subsequently, this leads to a smaller heat transfer area, smaller units and in some cases, fewer heat exchangers. In this work, an application for thermal and hydraulic computations of plate heat exchangers had been developed using Sharp Develop, an open source programming platform. During the development process, several literature methods and correlations for calculation of heat transfer coefficient and pressure drop in a plate heat exchanger have been tested and the selected four methods: Martin, VDI, Kumar and Coulson and Richardson have been incorporated into the software. The structure of the software is visually presented through several windows: a window for inserting input data, windows for showing the results of computation by each of the methods, a window for showing comparative analysis of the most important computation results obtained by all of the used methods and a help window for demonstrating the working principle of plate heat exchanger.

Keywords: plate heat exchanger, heat transfer, pressure drop, software.

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The transfer of heat to and from process fluids is an essential part of most chemical processes. Heat exchanger is a heat transfer device that transfers heat between two or more process fluids. One of the most commonly used types of heat exchangers is a plate heat exchanger (PHE).

Plate heat exchangers are widely used in dairy and food processing plants, chemical industries, power plants and central cooling systems. They exhibit excellent heat transfer characteristics, which allows a very compact design, and can easily be disassembled for maintenance, cleaning or for modifying the heat transfer area by adding or removing plates. Also, low temperature approaches can be used, as low as 1 °C, compared to 5 to 10 °C for shell and tube exchangers (SHE). The mean temperature correction factor, *F*, will normally be higher for PHE than for SHE, as the flow is closer to true counter-current flow. Fouling tends to be significantly lower in plate heat exchangers compared to shell and tube heat exchangers due to the higher velocities and the absence of dead angles.

Typically, a plate heat exchanger consists of a stack of corrugated or embossed metal plates in mutual contact, each plate having four openings serving as inlet and outlet ports, and seals designed so as to direct the fluids in alternate flow passages. The flow passages are formed by adjacent plates so that the two streams exchange heat while passing through alternate channels. When a package of plates is assembled, the holes at the corners form continuous tunnels leading the fluids from the inlet into the plate package, where they are distributed into narrow passages between the plates, and then collecting them before the outlet. Fluids are separated by a thin metal wall – corrugated plates. The shape of corrugations is a characteristic of each plate model and is carefully studied by the manufacturers. The purpose of corrugations is to provide turbulence in order to increase the heat transfer coefficients and, at the same time, to increase the structural strength of the assembly. The most common corrugation pattern is the herringbone (chevron) pattern, also studied in this work. Selection of the plates for any application depends on the process requirements in terms of heat transfer coefficients and allowable heat exchanger pressure drop. Patterns that provide higher heat transfer coefficients for a given flow rate also produce higher pressure drops. Thus, for any application, the designer must choose the plate type offering the best balance of the two effects.

The aim of this work was to develop software that could be used for thermal and hydraulic calculations of plate heat exchangers. Due to the great variety of plate dimensions and especially corrugation patterns, every attempt to establish a general calculation method that

Correspondence: D.M. Majstorović, Faculty of Technology and Metallurgy, University of Belgrade, Karnegijeva 4, 11120 Belgrade, Serbia.
E-mail: dbajic@tmf.bg.ac.rs
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can be applied to different types of plate heat exchangers has been unsuccessful so far. Very often correlations established by the producers of plate heat exchangers are used for estimating heat transfer coefficients and pressure drops in certain type of plates. However, several more general methods applicable to the herringbone plate type have been suggested in literature. Some of these methods have been tested and the results for heat transfer coefficient and pressure drop have been compared with the values obtained in the experimental setup already described in our previous papers [1–3]. Finally, four methods: Martin [4], VDI [5], Kumar [6] and Coulson and Richardson [7], have been selected and, in this work, incorporated into an application called PHeatEx Designer 1.0, for thermal and hydraulic design of plate heat exchangers with herringbone type of plates, developed using the Sharp Develop 5.0 Beta 4, the open source development environment. After its validation the application was used for rating of two industrial heat exchangers.

With the development of this software an attempt was made to collect in one place some of the available literature methods for herringbone type of plates, to compare the results obtained by using different calculation methods (*i.e.*, values of heat transfer coefficients, required area for heat transfer or pressure drops) and, if possible, to give some recommendation for their use.

CALCULATION PROCEDURE

In the course of investigations several literature methods for calculation of heat transfer coefficient in plate heat exchangers with a herringbone type of corrugation have been tested: Martin [4], VDI [5], Kumar [6], Coulson and Richardson [7] and generalized Muley and Manglik [8]. All the equations have been incorporated in an Excel sheet and the results have been compared with the values calculated from the experimental correlation obtained from a procedure described in our previous paper [2]. Experimental data have been correlated with the equation:

$$\mathrm{Nu} = 0.39515\mathrm{Re}^{0.6244}\,\mathrm{Pr}^{1/3} \tag{1}$$

in which *Nu*, *Re* and *Pr* are Nusselt, Reynolds and Prandtl numbers defined by the equations:

$$\mathrm{Nu} = \frac{\alpha D_\mathrm{e}}{\lambda} \tag{2}$$

$$\mathrm{Re} = \frac{\rho w D_\mathrm{e}}{\mu} \tag{3}$$

$$\mathrm{Pr} = \frac{\mu c_p}{\lambda} \tag{4}$$

where w is the fluid velocity, D_e is the equivalent diameter for the fluid flow channel and ρ, λ, c_p and μ are fluid density, thermal conductivity, specific heat and dynamic viscosity, respectively.

From the Eq. (1) and the definition of Nusselt number (Eq. (2)) the values of heat transfer coefficient could be easily obtained:

$$\alpha = \frac{\mathrm{Nu}\lambda}{D_\mathrm{e}} \tag{5}$$

Experimental values of pressure drop, determined as the difference between the measured inlet and outlet pressures for the single phase flow [2], have also been compared with results obtained by using literature methods [4–8]. The experimental measurements for single phase flow were conducted with the aqueous 30.6 mass% ethylene glycol solution. The same fluid, the same process parameters and the same geometry of the plates [1–3] were used for literature correlations during the selection process. Thermophysical properties of ethylene glycol and water binary mixture were taken from literature sources [9,10]. The obtained results for several experimental cases are given in Table 1, in the Results and Discussion section of this work.

Table 1. Process parameters for the experimentally investigated single-phase flow of ethylene glycol and water binary mixture

Parameter		Kumar	Martin	VDI	Coulson and Richardson	Muley and Manglik	Exp.
Mass flow rate m / kg s^{-1}	Mass flow rate per channel m_{ch} / kg s^{-1}	Inlet temperature, t_i / °C	Outlet temperature t_o / °C	Density ρ / kg m^{-3}	Viscosity μ / mPa s	Specific heat c_p / kJ kg^{-1} K^{-1}	Thermal conductivity λ / W m^{-1} K^{-1}
2.24	0.448	19.58	14.29	1040.9	2.429	3.790	0.4730
Heat exchanger duty, Q / kW					44.91		
Velocity, w / m s^{-1}					0.277		
Reynolds number		654.3	654.3	654.3	759.0	654.3	654.3
Heat transfer coefficient α / W m^{-2} K^{-1}		5847.9	4883.6	7042.2	4693.2	1372.2	5220.9
Friction pressure drop, Δp_k / Pa		30795	17234	17234	3566	–	–
Total pressure drop, Δp_{tot} / Pa		39750	26189	26189	12521	–	43700

Table 1. Continued

Mass flow rate, m / kg s^{-1}	Mass flow rate per channel, m_{ch} / kg s^{-1}	Inlet tempe-rature, t_i / °C	Outlet tempe-rature t_o / °C	Density ρ / kg m^{-3}	Viscosity μ / mPa s	Specific heat c_p / kJ kg^{-1} K^{-1}	Thermal con-ductivity λ / W m^{-1} K^{-1}
1.015	0.203	16.02	6.583	1045.0	2.828	3.780	0.4668
Heat exchanger duty, Q / kW					36.21		
Velocity, w / m s^{-1}					0.125		
Reynolds number		254.7	254.7	254.7	295.4	254.7	254.7
Heat transfer coefficient, α / W m^{-2} K^{-1}		3257.7	2786.0	4336.9	2677.0	498.1	3017.9
Friction pressure drop, Δp_k / Pa		7174	4804	4804	967	–	–
Total pressure drop, Δp_{tot} / Pa		16123	13753	13753	9916	–	18100
Mass flow rate, m / kg s^{-1}	Mass flow rate per channel, m_{ch} / kg s^{-1}	Inlet tempe-rature, t_i / °C	Outlet tempe-rature t_o / °C	Density ρ / kg m^{-3}	Viscosity μ / mPa s	Specific heat c_p / kJ kg^{-1} K^{-1}	Thermal con-ductivity λ / W m^{-1} K^{-1}
1.557	0.316	16.86	9.95	1043.5	2.669	3.784	0.4692
Heat exchanger duty, Q / kW					41.24		
Velocity, w / m s^{-1}					0.194		
Reynolds number		419.4	419.4	419.4	486.5	419.4	419.4
Heat transfer coefficient, α / W m^{-2} K^{-1}		4465.2	3713.2	5593.0	3692.8	855.9	4056.8
Friction pressure drop, Δp_k / Pa		16199	9632.3	9632.3	2015	–	–
Total pressure drop, Δp_{tot} / Pa		25151	18538	18538	10966	–	28300
Mass flow rate, m / kg s^{-1}	Mass flow rate per channel, m_{ch} / kg s^{-1}	Inlet tempe-rature, t_i / °C	Outlet tempe-rature t_o / °C	Density ρ / kg m^{-3}	Viscosity μ / mPa s	Specific heat c_p / kJ kg^{-1} K^{-1}	Thermal con-ductivity λ / W m^{-1} K^{-1}
1.190	0.238	17.34	8.69	1043.8	2.698	3.783	0.4687
Heat exchanger duty, Q / kW					38.94		
Velocity, w / m s^{-1}					0.147		
Reynolds number		312.9	312.9	312.9	363.0	312.9	312.9
Heat transfer coefficient, α / W m^{-2} K^{-1}		3688.1	3109.4	4776.2	3012.0	425.9	3389.1
Friction pressure drop, Δp_k / Pa		9595	6075	6075	1252	–	–
Total pressure drop, Δp_{tot} / Pa		18539	15018	15018	10195	–	21800
Mass flow rate, m / kg s^{-1}	Mass flow rate per channel, m_{ch} / kg s^{-1}	Inlet tempe-rature, t_i / °C	Outlet tempe-rature t_o / °C	Density ρ / kg m^{-3}	Viscosity μ / mPa s	Specific heat c_p / kJ kg^{-1} K^{-1}	Thermal con-ductivity λ / W m^{-1} K^{-1}
1.730	0.346	19.59	12.98	1041.4	2.468	3.789	0.4723
Heat exchanger duty, Q / kW					43.38		
Velocity, w / m s^{-1}					0.214		
Reynolds number		497.9	497.9	497.9	577.6	497.9	497.9
Heat transfer coefficient, α / W m^{-2} K^{-1}		4899.7	4054.1	6034.4	3951.0	1016.3	4421.0
Friction pressure drop, Δp_k / Pa		19110	11048	11048	2312	–	–
Total pressure drop, Δp_{tot} / Pa		28049	19987	19987	11251	–	30900
Mass flow rate, m / kg s^{-1}	Mass flow rate per channel, m_{ch} / kg s^{-1}	Inlet tempe-rature, t_i / °C	Outlet tempe-rature t_o / °C	Density ρ / kg m^{-3}	Viscosity μ / mPa s	Specific heat c_p / kJ kg^{-1} K^{-1}	Thermal con-ductivity λ / W m^{-1} K^{-1}
1.360	0.272	17.96	10.02	1043.1	2.626	3.785	0.4698
Heat exchanger duty, Q / kW					40.92		
Velocity, w / m s^{-1}					0.168		
Reynolds number		367.9	367.9	367.9	426.7	367.9	367.9

Table 1. Continued

Mass flow rate, m / kg s^{-1}	Mass flow rate per channel, m_{ch} / kg s^{-1}	Inlet temperature, t_i / °C	Outlet temperature t_o / °C	Density ρ / kg m^{-3}	Viscosity μ / mPa s	Specific heat c_p / kJ kg^{-1} K^{-1}	Thermal conductivity λ / W m^{-1} K^{-1}
1.360	0.272	17.96	10.02	1043.1	2.626	3.785	0.4698
Heat transfer coefficient, α / W m^{-2} K^{-1}		4075.8	3407.0	5177.5	3314.0	736.7	3721.7
Friction pressure drop, Δp_k / Pa		12292	7499	7499	1562	–	–
Total pressure drop, Δp_{tot} / Pa		21234	16441	16441	10504	–	22800
Mass flow rate, m / kg s^{-1}	Mass flow rate per channel, m_{ch} / kg s^{-1}	Inlet temperature, t_i / °C	Outlet temperature t_o / °C	Density ρ / kg m^{-3}	Viscosity μ / mPa s	Specific heat c_p / kJ kg^{-1} K^{-1}	Thermal conductivity λ / W m^{-1} K^{-1}
1.910	0.382	17.51	11.74	1042.5	2.576	3.786	0.4705
Heat exchanger duty, Q / kW					41.72		
Velocity, w / m s^{-1}					0.236		
Reynolds number		526.1	526.1	526.1	610.3	526.1	526.1
Heat transfer coefficient, α / W m^{-2} K^{-1}		5139.7	4244.7	6293.8	4151.2	1091.8	4628.4
Friction pressure drop, Δp_k / Pa		23039	13217	13217	2762	–	–
Total pressure drop, Δp_{tot} / Pa		31994	22172	22172	11717	–	35900
PD (α) / %		10.03	7.99	38.43	10.70	79.60	–
PD (Δp_{tot}) / %		10.43	33.02	33.02	59.42	–	–

Based on the obtained results, four literature methods were selected and incorporated in the software for thermal and hydraulic design of plate heat exchangers. These methods are: Martin [4], VDI [5], Kumar [6] and Coulson and Richardson [7]. The selection of methods was mostly based on their general applicability and, in some cases, on good agreements with values for heat transfer coefficient obtained from the experimental correlation (1) and Eq. (2), reported in Table 1. All of the selected methods consist of three phases of calculation: 1) calculation of basic geometrical characteristics of the exchanger and other general characteristics of the process, 2) heat transfer calculation and 3) pressure drop calculation.

Heat exchanger duty is determined from one of the following equations:

$$Q = m_h c_{p,h}\left(t_{hi} - t_{ho}\right) \tag{6}$$

$$Q = m_c c_{p,c}\left(t_{co} - t_{ci}\right) \tag{7}$$

where m is the mass fluid flow rate, c_p is the fluid specific heat, and t_i and t_o are inlet (i) and outlet (o) temperatures, while subscripts h and c designate hot and cold fluid, respectively.

Available area for heat transfer, A_s, is defined by the equation:

$$A_s = L_p W\left(N_{pl} - 1\right) \tag{8}$$

in which L_p is the plate length, W is the plate width and N_{pl} is the number of plates in the heat exchanger. In the Eq. (8), the effective plate length L_m, if known, could be used instead of the plate length L_p.

Required area for heat transfer A_p can be calculated from the equation:

$$A_p = \frac{Q}{KF\Delta t_{ln}} \tag{9}$$

where K is the overall heat transfer coefficient, Δt_{ln} is the logarithmic mean temperature difference and F is the correction factor for Δt_{ln}. For plate heat exchangers the correction factor is often close to 1.0 since the flow configuration is closer to true counter-current flow than in the shell and tube heat exchangers. Overall heat transfer coefficient K (W m^{-2} K^{-1}) is determined from the equation:

$$\frac{1}{K} = \frac{1}{\alpha_h} + \frac{1}{\alpha_c} + \frac{\delta_{pl}}{\lambda_{pl}} + R_{p,h} + R_{p,c} \tag{10}$$

where α_h and α_c are heat transfer coefficients for hot and cold fluid, respectively, $R_{p,h}$ and $R_{p,c}$ are corresponding fouling factors, δ_{pl} is the plate thickness and λ_{pl} is the thermal conductivity of the plates.

Reserve in heat transfer area ΔA (%) is calculated from the equation:

$$\Delta A = 100\frac{A_s - A_p}{A_s} \tag{11}$$

Cross sectional area of the channel S is defined by the equation:

$$S = BW \tag{12}$$

where B is the average distance between adjacent plates determined as:

$$B = 2a \tag{13}$$

in Martin [4] and VDI [5] correlations. In Kumar [6] and Coulson and Richardson [7] methods parameter B is calculated from the expression:

$$B = p_\mathrm{p} - \delta_\mathrm{pl} \tag{14}$$

in which p_p (m) is the plate profile pitch and δ_pl (m) is the plate thickness.

Equivalent diameter D_e for the fluid flow channel in Coulson and Richardson [7] method is determined as:

$$D_\mathrm{e} = 2B \tag{15}$$

In Martin [4], VDI [5] and Kumar[6] methods the following equation for equivalent diameter is used:

$$D_\mathrm{e} = \frac{2B}{\mu_\mathrm{p}} \tag{16}$$

where μ_p is the plate enhancement factor due to corrugation. If not given by the plate manufacturer, it could be calculated as a function of average distance between adjacent plates (B) and wave length of the plate (Λ) according to the described procedure [4,5]:

$$\mu_\mathrm{p} = \frac{1}{6}\left(1 + \sqrt{1+X^2} + 4\sqrt{1+X^2/2}\right) \tag{17}$$

where X is calculated as:

$$X = \frac{B\pi}{\Lambda} \tag{18}$$

Heat transfer coefficient in Martin [4] and VDI [5] methods is determined from the Eq. (5). The difference between these two methods is in the equations used for calculating the Nusselt number. In Martin [4] method, the following equation is used:

$$\mathrm{Nu} = 0.122\mathrm{Pr}^{1/3}(\mu/\mu_\mathrm{w})^{1/6}\left(\zeta \sin(2\alpha_\mathrm{pl})\mathrm{Re}^2\right)^{0.374} \tag{19}$$

while in VDI correlation [5], the Nusselt number is calculated from the equation:

$$\mathrm{Nu} = 1.615[(\xi \mathrm{Re}/64)\mathrm{Re}\mathrm{Pr}D_\mathrm{e}/L]^{1/3} \tag{20}$$

In both equations Re and Pr are Reynolds and Prandtl numbers of the fluid, μ (Pa s) and μ_w (Pa s) are viscosities of the fluid at the mean fluid temperature and wall temperature, respectively, while ξ represents the fluid friction coefficient. The friction coefficient ξ depends on three friction components (ξ_0, $\xi_{1,0}$, ξ_1) and plate corrugation angle α_pl. Friction components (ξ_0, $\xi_{1,0}$, ξ_1) are functions of the Reynolds number and operating conditions accounted for by empirical coefficients and constants. More details on calculation procedures could be found elsewhere [4,5]. The characteristic length L (m) could be determined as a function of the corrugation angle (α_pl) and wave length (Λ), from the equation:

$$L = \frac{\Lambda}{\sin(2\alpha_\mathrm{pl})} \tag{21}$$

The heat transfer coefficient in Kumar method [6] is calculated using the expression:

$$\alpha = \frac{j_\mathrm{h}\lambda \mathrm{Pr}^{0.33}\left(\frac{\mu}{\mu_\mathrm{w}}\right)^{0.17}}{D_\mathrm{e}} \tag{22}$$

Heat transfer factor (j_h) is determined as a function of Reynolds number (Re) and plate profile angle ($\beta = 90 - \alpha_\mathrm{pl}$) according to the described procedure [6].

Coulson and Richardson [7] method uses the following expression for calculation of the heat transfer coefficient:

$$\frac{\alpha D_\mathrm{e}}{\lambda} = 0.26\mathrm{Re}^{0.65}\mathrm{Pr}^{0.4}\left(\mu/\mu_\mathrm{w}\right)^{0.14} \tag{23}$$

The total fluid pressure drop Δp_tot is the sum of three values: a) pressure drop in exchanger channels (friction pressure drop), b) pressure drop in nozzles, and c) pressure drop due to elevation. The pressure drop in exchanger channels according to Martin [4] and VDI [5] methods is calculated:

$$\Delta p_\mathrm{k} = N\frac{\xi\rho w^2 L_\mathrm{p}}{2D_\mathrm{e}} \tag{24}$$

in which N represents the number of fluid passes through the exchanger while ξ is the fluid friction coefficient, already mentioned in the Eqs. (19) and (20). In the Kumar method [6], friction coefficient ξ is replaced by the Fanning – type friction factor f, and a correction factor due to the wall temperature influence is taken into account:

$$\Delta p_\mathrm{k} = N\frac{4f\rho w^2 L_\mathrm{p}}{2D_\mathrm{e}\left(\frac{\mu}{\mu_\mathrm{e}}\right)^{0.17}} \tag{25}$$

The friction factor, similar to the heat transfer factor (j_h), is determined as a function of the Reynolds number (Re) and plate profile angle ($\beta = 90 - \alpha_\mathrm{pl}$) according to the procedure described in [6].

According to Coulson and Richardson [7], pressure drop in exchanger channels is calculated:

$$\Delta p_k = N\frac{8 j_f \rho w^2 L_p}{2D_e} \qquad (26)$$

while the friction factor j_f can be determined from the expression:

$$j_f = 0.6\mathrm{Re}^{-0.3} \qquad (27)$$

Pressure drop in nozzles, in all four selected methods, is calculated from the equation:

$$\Delta p_p = N\frac{1.3 w_p^2 \rho}{2} \qquad (28)$$

where w_p represents the fluid velocity in nozzles.

Pressure drop due to elevation has to be taken into account in the case of uneven number of fluid passes, and can be determined from the equation:

$$\Delta p_{el} = \rho g L_p \qquad (29)$$

Software description

The software PHeatEx Designer 1.0 for thermal and hydraulic calculation of plate heat exchangers has been developed in Sharp Develop 5.0 Beta 4, the open source development environment. The software is activated by clicking the shortcut shown in Figure 1a. After running the application, the **Login Form** window, shown in Figure 1b, will appear.

After entering the correct password Application Loading message will appear and the main window, shown in Figure 2, will open.

From the menu in the upper part of the window, the following sections could be chosen: **Insert Input Data, Methods of Calculation, Working Principle** and **Analysis of Results.** After choosing **Insert Input Data**, a new form, **Insert Input Data** window, shown in Figure 3, will open.

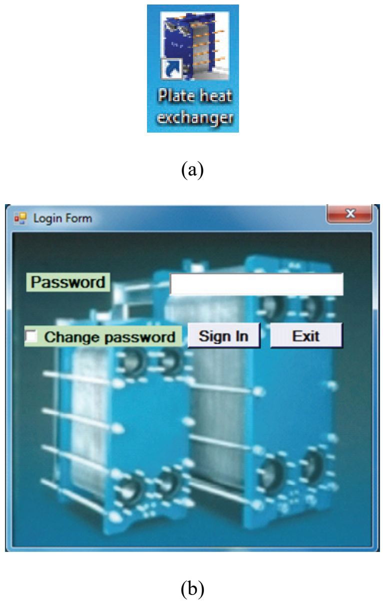

Figure 1. Application shortcut (a) and login form window (b).

After choosing one of the calculation methods from **Methods of Calculation** drop-down menu, a new form which depends on the chosen method will open. If any of the input parameters necessary for calculation are missing, a message, in a form of Message Box as shown in Figure 4, will appear.

When one of the methods of calculation is chosen, a form which describes that method will open as can be seen in Figure 5 for Kumar method. In the upper part of the window four commands are offered: **Back –** for returning to the main window, **Start Calculating**, **Save Results**–for saving the calculation results in a form of a Notepad Document, Microsoft Word Document or Microsoft Excel Document and **Delete Results**.

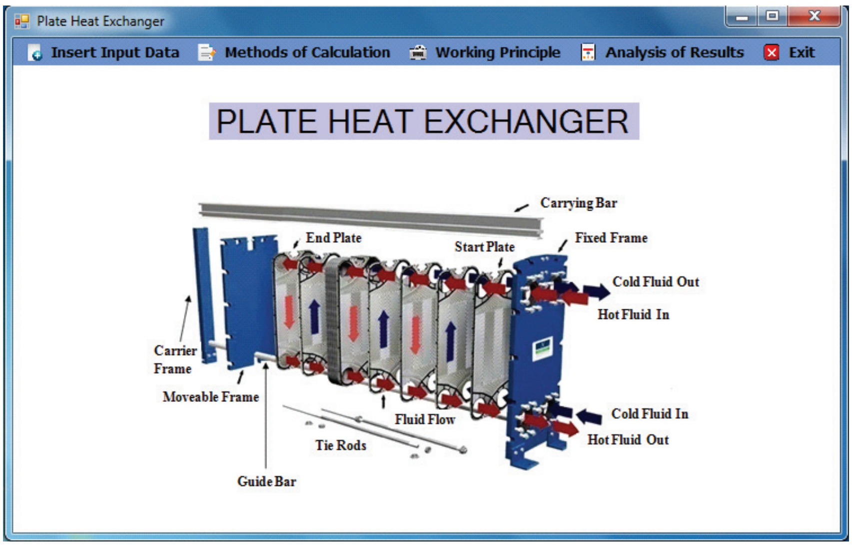

Figure 2. Application main window.

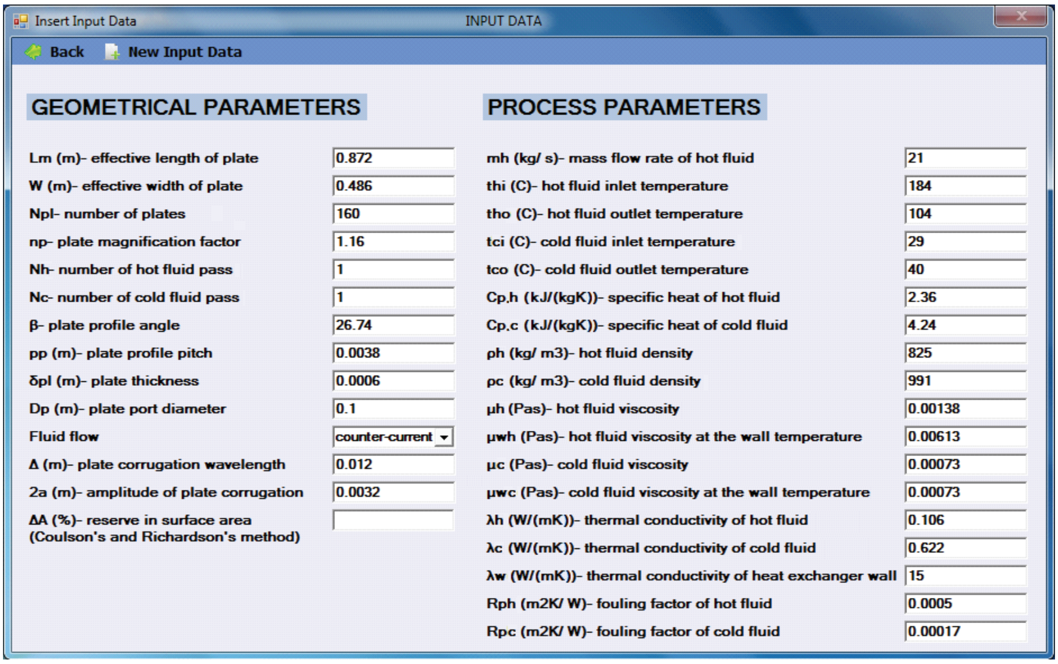

Figure 3. Insert input data window.

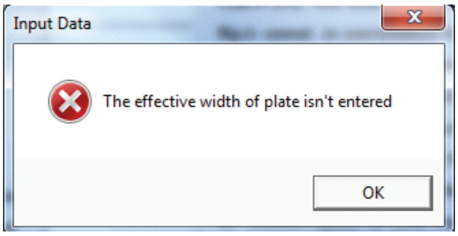

Figure 4. Missing parameter warning window.

After running the calculation, all output parameters will be shown in a separate window and the most important results (heat transfer coefficients of both fluids, overall heat transfer coefficient, available surface area, required surface area, total pressure drops of both fluids, etc.) will be also shown in a Message Box.

The **Working Principle** window, shown in Figure 6, contains two video clips which show working principle and components of a plate exchanger.

The **Analysis of Results** window is a part of the software, which allows comparison of the results obtained by different methods. If so desired, some of the methods could be excluded from the comparative analysis, as shown in Figure 7.

RESULTS AND DISCUSSION

The process of selection of literature methods to be used for the development of the application PHeatEx Designer 1.0 for thermal and hydraulic calculations of plate heat exchangers, was based on comparison of the values determined using literature methods [4–9] with experimental data measured on the setup described in our previous work [2]. A survey of the obtained results for seven experimental cases is given in Table 1. Literature methods were assessed by calculating percentage deviation for heat transfer coefficient and pressure drop from the equation:

$$PD(Y) = \frac{100}{m}\sum_{i=1}^{m}\left|\frac{Y_{exp}-Y_{cal}}{Y_{exp}}\right|_i \tag{30}$$

where Y_{exp} and Y_{cal} denote experimental and calculated values of the heat transfer coefficient and pressure drop, while m is the number of experimental data points.

From the Table 1 it can be concluded that the best agreement with experimental values for heat transfer coefficient, with around 8% deviation, was obtained by using the Martin method [4]. This result is considered as very good since the experimental uncertainty in determination of heat transfer coefficient is estimated between 10% and 15% [2]. Satisfactory results, below the value of experimental uncertainty, were also obtained with Kumar [6] and Coulson and Richardson [7] methods, with the difference that the Kumar method [6] gave higher and Coulson and Richardson method [7] lower values of the heat transfer coefficient than the experimental ones. VDI method [5], which is based on similar calculation procedure as the Martin method [4], unexpectedly gave much higher deviation, around 38%. The Muley and Manglik method [8] has been known for giving considerably lower values for Nusselt number and heat transfer coefficient than the other literature methods, and this was confirmed by our results, given in Table 1. Consequently, this method

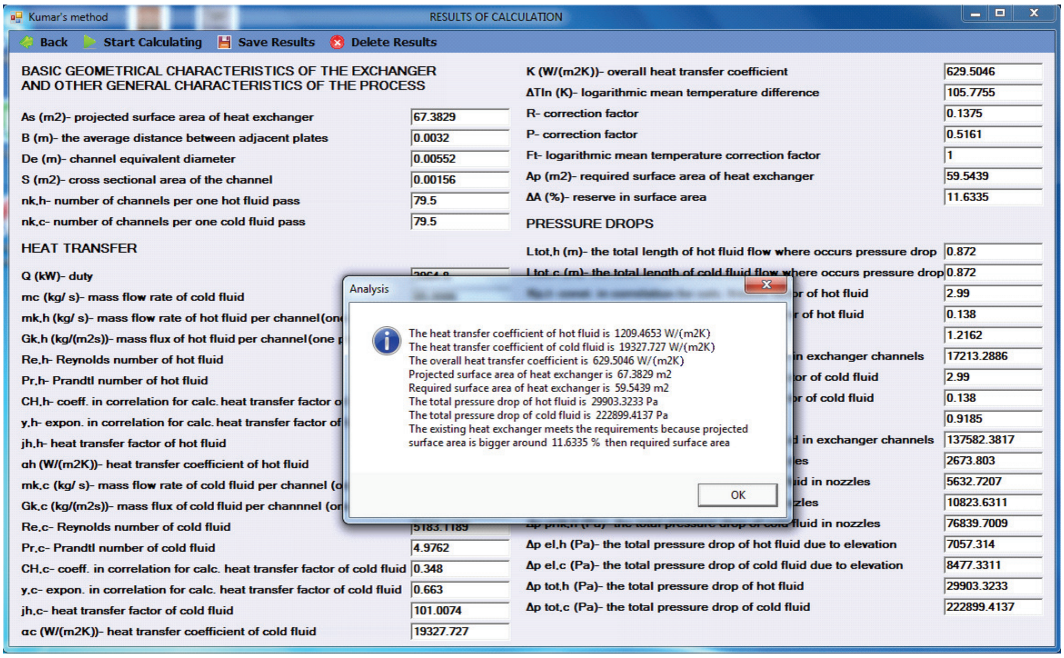

Figure 5. Kumar method window.

was not included in the software for thermal and hydraulic design of plate heat exchangers.

The best agreement with experimental values for pressure drop, below estimated experimental uncertainty of 15–17% [2] was obtained by using the Kumar method [6]. Martin [4] and VDI [5] methods gave 33% lower values than the measured pressure drop. The highest deviation from experimental data was obtained by using the Coulson & Richardson method [7], due to very low values of friction pressure drop calculated by the equation (26).

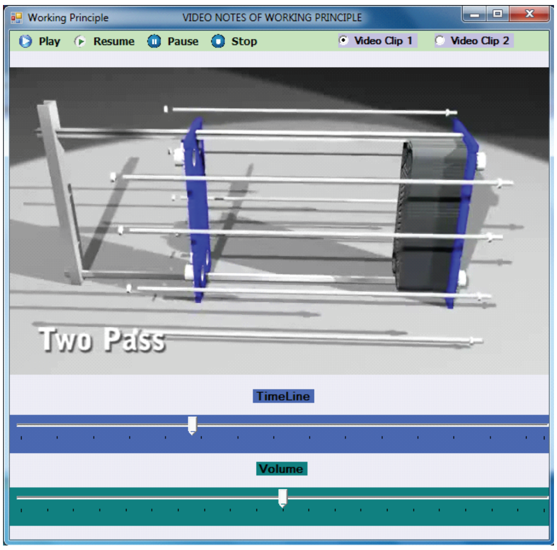

Figure 6. PHE working principle window.

Validation of the software was performed by comparison of the software results with the values calculated in an Excel sheet for each of the selected methods. After its validation the software was applied on two industrial plate heat exchangers. Process parameters, geometry of the plates and thermophysical properties of working fluids were taken from the project documentation.

The most important geometrical data and process parameters for the first tested plate heat exchanger are given in Table 2. Process fluid, light vacuum gas oil (LVGO), was going through the temperature change from 184 to 104 °C. As a cooling medium, water with inlet temperature of 29 °C was used.

Summary of the software results calculated by four literature methods, Martin [4], VDI [5], Kumar [6] and Coulson and Richardson [7], as well as the values obtained from the Excel sheet by using the experimental correlation (Eq. (1)) for calculation of the heat transfer coefficient, is given in Table 3. The experimental correlation was not included in the software since its applicability was never properly tested under conditions different from the ones under which it was originally established.

From Table 3 it can be concluded that in the first investigated case, the greatest disagreement is between the Martin [4] and VDI [5] methods, which were, interestingly, both established by the same author. These two methods give the upper and lower limits of possible values for the heat transfer coefficient and consequently the area required for heat transfer. As expected, values of the heat transfer coefficient for hot fluid are much lower than for the cold fluid due to the difference in thermophysical properties (especially viscosity) between LVGO and water. Also, the velocity

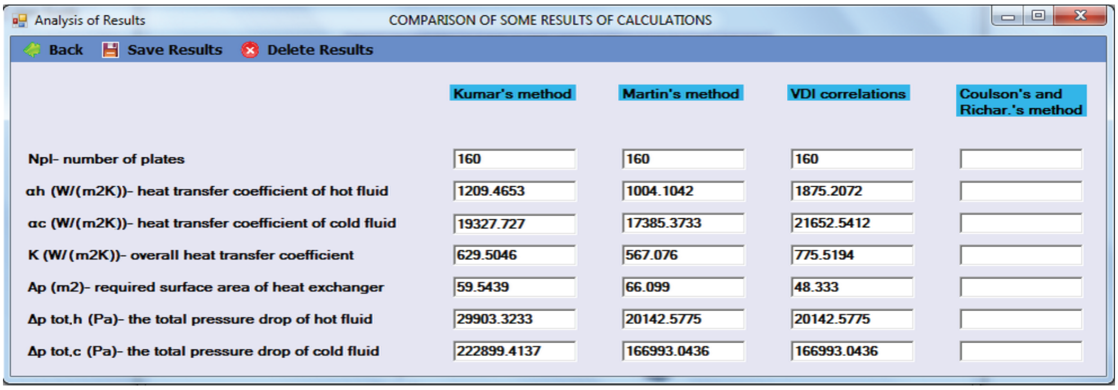

Figure 7. Analysis of results window.

Table 2. Process parameters and geometry of the plates for the first tested plate heat exchanger

Geometrical data	Value	Process parameter	Value
Length, L_p / m	0.872	Hot fluid	LVGO
Width, W / m	0.486	Cold fluid	water
Corrugation angle, α_{pl} / °	60	Hot fluid mass flow rate m_h / kg s^{-1}	21
Plate thickness, δ_{pl} / m	0.0006	Hot fluid inlet temperature, t_{hi} / °C	184
Plate enhancement factor, μ_p	1.16	Hot fluid outlet temperature, t_{ho} / °C	104
Amplitude of corrugation, 2a	0.0032	Cold fluid inlet temperature, t_{ci} / °C	29
Flow configuration	1–1	Cold fluid outlet temperature, t_{co} / °C	40
Area of heat exchanger, A_s / m^2	67.4	Hot fluid density, ρ_h / kg m^{-3}	825
(available for heat transfer)		Hot fluid viscosity, μ_h / Pa s	0.00138
		Hot fluid thermal conductivity, λ_h / W m^{-1} K^{-1}	0.106
		Hot fluid specific heat, $C_{p,h}$ / kJ kg^{-1} K^{-1}	2.36
		Hot fluid fouling factor, R_{ph} / m^2 K W^{-1}	0.0005
		Cold fluid fouling factor, R_{pc} / m^2 K W^{-1}	0.00017

Table 3. Calculation results for the first tested plate heat exchanger; Exp. values calculated from the Excel sheet

Parameter	Martin	VDI	Kumar	Coulson and Richardson	Exp.
Heat exchanger duty, Q (kW)	3964.8	3964.8	3964.8	3964.8	3964.8
Hot fluid heat transfer coefficient, α_h / W m^{-2} K^{-1}	1005.9	1878.3	1212.0	1049.7	1318.9[a]
Cold fluid heat transfer coefficient, α_c / W m^{-2} K^{-1}	17422.7	21695.0	19370.4	13754.7	15032.0[a]
Overall heat transfer coefficient, K / W m^{-2} K^{-1}	567.7	776.1	630.2	576.3	651.6
Required area for heat transfer, A_p / m^2	66.0	48.3	59.5	65.0	57.5
Reserve in heat transfer area, ΔA / %	2.0	28.3	11.7	3.6	14.7
Hot fluid friction pressure drop, $\Delta p_{k,h}$ / Pa	7488	7488	17317	1546	–
Cold fluid friction pressure drop, $\Delta p_{k,c}$ / Pa	82136	82136	138443	11456	–

[a]Values calculated from the Excel sheet using the empirical correlation (1) and Eq. (2)

of hot fluid through one channel is more than three times lower than the velocity of cold fluid (0.21 in comparison to 0.69 m s^{-1}), which results in much higher pressure drop on the cold fluid side. Values of the friction pressure drop, calculated by using the investigated methods, are very different, the upper and the lower limits being defined by Kumar [6] and Coulson & Richardson [7] methods.

The Muley and Manglik [8] method, although not included in the software, was also tested by means of the Excel sheet. According to this method, demands for heat transfer area are much higher than the available area for heat transfer, while all the other literature correlations showed the existence of some reserve in the heat transfer area, although the numbers varied from 2.0 to 28.3%.

As an illustration, another example of the results obtained by the plate heat exchanger design software is shown here. In this case hot and cold fluids had similar thermophysical properties: demineralized water was heated from 40 to 60 °C while process water was used as a heating medium. The most important geometrical data and process parameters are presented in Table 4 and calculation results are compared in Table 5.

From the results presented in Table 5 it can be concluded that the upper limit for heat transfer coefficient and consequently the largest reserve in heat transfer area was again obtained by using the VDI [5] correlation, and the lower limit is, in this case, predicted by the Coulson and Richardson [7] method. Values of the friction pressure drop are, as in the previous case, very different, with the upper and lower limits obtained by Kumar [6] and Coulson and Richardson [7] methods.

Table 4. Process parameters and geometry of the plates for the second tested plate heat exchanger

Geometrical data	Value	Process parameter	Value
Length, L_p / m	0.472	Hot fluid	water
Width, W / m	0.286	Cold fluid	Demineralized water
Corrugation angle, α_{pl} / °	60	Cold fluid mass flow rate m_h / kg s^{-1}	10
Plate thickness, δ_{pl} / m	1-1	Hot fluid inlet temperature, t_{hi} / °C	100
Plate enhancement factor, μ_p	6.61	Hot fluid outlet temperature, t_{ho} / °C	90
Amplitude of corrugation, 2a	0.472	Cold fluid inlet temperature, t_{ci} / °C	40
Flow configuration	0.286	Cold fluid outlet temperature, t_{co} / °C	60
Area of heat exchanger, A_s / m^2	60	Hot fluid fouling factor, R_{ph} / m^2 K W^{-1}	0.00017
(available for heat transfer)		Cold fluid fouling factor, R_{pc} / m^2 K W^{-1}	0.00017

Table 5. Calculation results for the second tested plate heat exchanger; Exp. values calculated from the Excel sheet

Parameter	Martin	VDI	Kumar	Coulson and Richardson	Exp.
Heat exchanger duty, Q / kW	839.4	839.4	839.4	839.4	839.4
Overall heat transfer coefficient, K / W m^{-2} K^{-1}	3212.0	3411.3	3302.3	2888.9	3020.1[a]
Required area for heat transfer, A_p / m^2	5.83	5.49	5.67	6.48	6.20
Reserve in heat transfer area, ΔA / %	11.8	17.0	14.2	1.9	6.2
Hot fluid friction pressure drop, $\Delta p_{k,h}$ / Pa	70790	70790	111453	7752	–
Cold fluid friction pressure drop, $\Delta p_{k,c}$ / Pa	19608	19608	32688	2768	–

[a]Values calculated from the Excel sheet using the empirical correlation (1) and Eqs. (2) and (10)

CONCLUSION

Software PHeatEx Designer 1.0 for the thermal and hydraulic design of plate heat exchangers based on four literature methods: Kumar, Martin, VDI and Coulson and Richardson, has been developed in the Sharp Develop programming environment. The selection of methods to be incorporated into the application was based on comparison of the results obtained by literature methods with experimental values. The application offers to the user a number of options regarding the calculation method, saving results, review and representation of the results, comparative analysis of the results obtained by different calculation methods as well as the explanation of plate heat exchanger components and working principle in two video clips. After its validation, the application has been tested on heat exchangers from industrial practice and the results have been compared and analyzed.

It is hoped that this software will help to bridge the gap between teaching the fundamentals and industrial design and application of plate heat exchangers.

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IZVOD

PROGRAMSKI PAKET ZA PROJEKTOVANJE PLOČASTOG RAZMENJIVAČA TOPLOTE ZA INDUSTRIJSKU I EDUKATIVNU PRIMENU

Nikola R. Zlatković, Divna M. Majstorović, Mirjana Lj. Kijevčanin, Emila M. Živković

Tehnološko-metalurški fakultet, Univerzitet u Beogradu, Karnegijeva 4, 11120 Beograd, Srbija

(Stručni rad)

Pločasti razmenjivači toplote su tip aparata koji danas nalazi veoma široku oblast primene, od prehrambene i hemijske industriji do elektroenergetskih postrojenja. Osnovni konstruktivni element ovih aparata je slog orebrenih metalnih ploča. Orebrenja na pločama su tako dizajnirana da omogućavaju postizanje turbulentnog strujanja duž cele površine za toplotnu razmenu, čime se postižu najviši mogući koeficijenti prenosa toplote pri relativno maloj razlici temperatura fluida. Ovo dovodi do manje potrebne površine za toplotnu razmenu, manjih dimenzija aparata i, u nekim slučajevima, manjeg broja razmenjivača toplote. Pored odličnih karakteristika u pogledu prenosa toplote, imaju i brojne druge prednosti u odnosu na najčešće korišćene razmenjivače toplote sa cevnim snopom i omotačem: lako se sastavljaju i rastavljaju radi čišćenja i popravki, površina za toplotnu razmenu se lako može menjati promenom broja ploča u slogu, mogu raditi pri vrlo malim razlikama temperature fluida koji učestvuju u toplotnoj razmeni, a stvaranje onečišćenja je manje izraženo zbog većih brzina strujanja i bolje distribucije toka fluida. Osnovni problem pri termohidrauličkom proračunu pločastih razmenjivača toplote je ograničen broj literaturnih korelacija, posebno onih koje bi imale širi opseg primenljivosti, što je i razumljivo obzirom na različite geometrije i tipove profila ploča koji se javljaju kod ovih aparata. U ovom radu je testirano nekoliko literaturnih metoda za proračun koeficijenta prelaza toplote i pada pritiska u pločastom razmenjivaču toplote sa orebrenjem tipa "riblja kost" i dobijeni rezultati su upoređeni sa eksperimentalno određenim vrednostima za koeficijent prelaza toplote i pad pritiska. Izabrane su četiri metode koje su ugrađene u programski paket za termički i hidraulički proračun pločastog razmenjivača toplote razvijenog u okruženju Sharp Develop. Ovaj programski paket za termički i hidraulički proračun pločastog razmenjivača toplote koristi sledeće četiri metode proračuna: Martinov metod, VDI metod, Kumarov metod i metod Kulsona i Ričardsona (Coulson–Richardson). Razvijeni softver je vizuelno prikazan preko prozora za unos ulaznih podataka, prozora za prikaz rezultata proračuna dobijenih svakom od metoda, prozora na kome je prikazano poređenje najbitnijih rezultata proračuna za sve korišćene metode i pomoćnog prozora za prikaz principa rada pločastog razmenjivača toplote. Nakon izbora metoda, razvoja i verifikacije softvera pristupilo se njegovoj primeni na dva industrijska razmenjivača toplote.

Ključne reči: Pločasti razmenjivač toplote • Prenos toplote • Pad pritiska • Programski paket