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Heat Transfer and Pressure Drop in Cross-Flow Welded Plate Heat Exchanger for Ammonia Synthesis Column

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Plate Heat Exchanger (PHE) is one of the modern types of compact heat transfer equipment, which can significantly enhance the heat recuperation and improve efficiency of energy usage in many industrial applications. The construction of welded PHE (WPHE) can significantly widen the range of its application on temperature and pressure. The results of experimental study of heat transfer and pressure drop in a model of WPHE specially designed for work in ammonia synthesis columns are presented. The model consists of the round plates with special form of corrugations. The plates are welded together forming a pack with channels for two heat exchanging streams. The streams are in a cross flow that correspond to one pass of the multi-pass WPHE with overall counter-current flow. It is shown that for calculation of heat transfer effectiveness can be used the cross flow model where fluid in one channel is mixed, but in another unmixed. The Equations for calculation of film heat transfer coefficients and pressure drop are presented, which can be used in earlier developed method of WPHE design. The validity of the Equations and developed method of WPHE design is confirmed by the data of tests with WPHE installed in ammonia synthesis column at temperature up to 520 °C and pressure 32 MPa.

1. Introduction

Efficient heat recuperation is of primary importance in resolving the problem of efficient energy usage and consequent reduction of fuel consumption and greenhouse gas emissions, as is discussed by Klemes and Varbanov (2013). Heat transfer enhancement is substantially facilitate the solution of the problem (Gough et al., 2013). Plate Heat Exchanger (PHE) is one of the modern efficient types of compact intensified heat transfer equipment. The principles of the construction and design for different types of PHEs are sufficiently well described in the literature, e.g. (Klemes et al., 2015). The conventional type is plate-and-frame PHE, which was initially developed for the food industry and later proved efficient in many other applications. It is confirmed by a number of researchers, as e.g. Hajabdollahi et al. (2016) have found in their study case of water to water HE that the comparison of the optimum results for plate-and-frame PHE shown 13 % improvement in the total cost compared with shell-and-tube heat exchanger at the same operating conditions. Their flexibility allows finding economically viable solutions in different processes of waste heat utilisation, asshown by e.g. Arsenyeva et al. (2016). However, due to elastomer gaskets the range of plate-and-frame PHE application is limited to pressures up to 25 bar and temperatures up to 180 °C and working fluids friendly to gaskets material. Besides, the cost of the gaskets, especially for severe working conditions, can dramatically increase the cost of heat exchanger as a whole unit. To widen the PHE application range by excluding elastomer gaskets the brazed (BPHE) and welded (WPHE) types of PHE were developed. In the construction of welded PHE the gaskets between plates are eliminated, that allows to widen significantly the range of its application on temperatures and pressures.

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Nowadays there are a number of different by construction principles types of welded PHEs produced by contemporary PHE manufacturers. The comparison of two most widely used types, which are Plate-and-Block (Compabloc) HE and Plate-and-Shell HE (PSHE), is presented by Arsenyeva et al. (2016). According to analysis published by Andersson et al.(2009) before the year 2009 it was installed more than 750 Compabloc HEs only in oil refining industry on different positions worldwide.

A different design of welded PHE is developed for work at ammonia synthesis process under high temperature up to 520 °C and pressure up to 32 MPa. Here the results of experimental research on the heat transfer and hydraulic resistance in channels formed by plates of such WPHE are presented.

The WPHE developed for work in high pressure shell of ammonia synthesis column consist of a stack of round plates with special form of corrugations, depicted in Figure 1. The plates are welded together to form a number of channels for cold and hot streams exchanging heat. The welded collectors of special design are organizing multi pass movement of both streams with overall counter current flow. The movement of two streams in one pass block is cross flow. Compare to the flow of streams in conventional plate-and-frame PHE there is significant difference. From hydraulic point of view the stream is entering the channel through almost full cross section, while in channels of plate-and-frame PHE it is entering from distribution collector of small diameter compare to channel width. It causes much smaller local hydraulic resistance at the port zone of WPHE and ensures even flow distribution across channel width. But for the heat transfer cross flow of streams causes the reduction of mean temperature difference compare to counter flow in one pass of plate-and-frame PHE. The overall counter flow in WPHE is making this loss in mean temperature difference smaller, but still this cross flow feature for individual passes should be accounted in correct PHE design. The available literature data are not directly applicable, as the level of fluid mixing across PHE channel is not known.

2. Experimental model and test rig

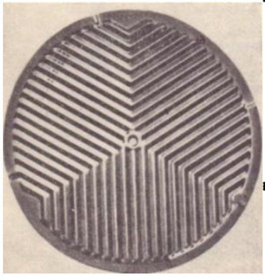
The heat transfer and pressure drop in WPHE were investigated experimentally using the model consisted of 15 plates representing the block of plates in one pass of WPHE. The plates are forming 14 channels, in 7 of which the hot stream is directed and in other 7 channels is directed the cold stream. The geometrical forms of the channels for hot and cold fluids are different. For the cold stream the channel is formed by one plate with most of the corrugations (at 2/3 plate area) directed along the main flow direction. The adjacent plate has herringbone corrugation direction with angle 60 degrees to main flow direction. Resulting average corrugation angle in this area is 30 º. At the remaining 1/3 of plate area the angle of corrugations to flow direction is 60 º on both adjacent plates. The average angle of corrugations to flow direction is β_2=40 º. Such form of corrugations is made to facilitate the discharge of possible dust in the synthesis gas stream after catalyser, which can appear with catalyser aging. The average angle of corrugations in another channel for hot stream is equal to β_1=50 º. The main geometrical parameters of the tested model, plate corrugations and inter-plate channels are presented in Table 1.

Table 1: Table title (Style: CET-table-title)

Total heat transfer surface area, F_a, m^2	4.2
Number of plates, N_p	15
Heat transfer surface area of one plate, F_p,m^2	0.32
Cross section area of one channel, f_{ch}, m^2	0.0022
Plate outside diameter, D_o, m	0.626
Plate thickness, δ_w, m	0.001
Plate metal	AISI 304
Heat conductivity of the wall, λ_w, W/(m K)	16
Corrugations height, *b*, m	0.004
Corrugations pitch, S, m	0.018
Average channel width, W_{ch}, m	0.55
Equivalent diameter of channel, d_e, m	0.008
Average corrugations angle to main flow direction: For	
hot stream, β_1, degrees	50
For cold stream, β_2,degrees	40
The width of channel entrance (exit), W_{enx}, m	0.4

The thermal and hydraulic characteristics of the WPHE model were measured at specially developed test rig with water as test fluid for both streams, schematically shown in Figure 2. The testing fluid is distilled water. It is pumped from water tank to the tubular heat exchanger, where is heated by steam to required temperature

t_{11} of hot water inlet to the model. After being cooled in the model to outlet temperature t_{12}, the water is entering the PHE, where it is cooled by cooling water from the central cooling water circuit with cooling tower. It is cooled to the required temperature t_{21} of cold stream inlet to the WPHE model. Heated in the model to the cold stream outlet temperature t_{22}, water is returning back to the water tank. The flowrate of the coming to the model cold water is measured by calibrated orifice meter. The temperatures at heat exchanging streams inlet and outlet at the WPHE model were measured with calibrated copper-constantan thermocouples with accuracy ±0.05 °C. Two differential diaphragm-pressure gauges were used to measure pressure differences between the inlet and outlet ports of the hot and cold water. The tests were made in stabilized regime when the parameters of the process were not changed in consecutive 2 min. The inlet temperature of hot water was varied in the range 55 – 85 °C, inlet temperature of cold water 30 – 45 °C. The mass flow rate of water was varied in the range from 0.8 kg/s to 4.5 kg/s.



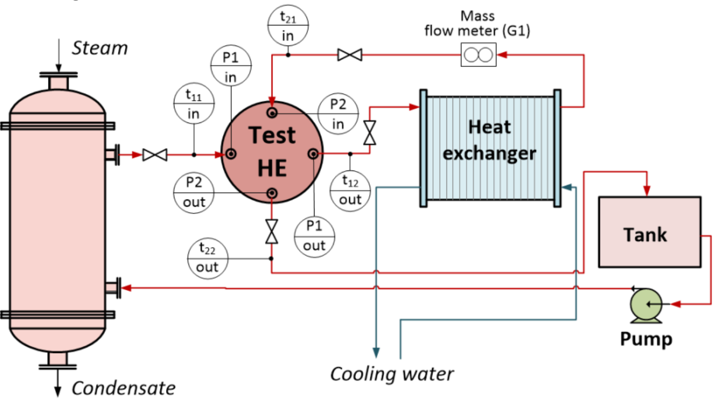


Figure 1: The plate of WPHEFigure 2: Schematic drawing of the test rig for ammonia synthesis column

3. Results and discussion

The results of the tests measurements allowed calculate heat load of the test model. The heat load Q for further calculations was taken as average between the values calculated for hot Q_1 and cold Q_2 streams, W:

$$Q_1 = G \cdot c_{p1} \cdot (t_{11} - t_{12}); \_\_ Q_2 = G \cdot c_{p2} \cdot (t_{22} - t_{21}); \_\_ Q = (Q_1 + Q_2)/2, \tag{1}$$

where G is water mass flow rate, kg/s; c_{p1} and c_{p2} are specific heat capacities of hot and cold water, J/(kg K). The mass flow rates of both streams in the model are equal and heat capacity differ not more than 0.3%, hence the mean temperature difference is calculated as:

$$\Delta t_m = \left[(t_{11} - t_{22}) + (t_{12} - t_{21}) \right] / 2 \tag{2}$$

The experimental value of overall heat transfer coefficient, W/(m^2 K):

$$U_{ex} = \frac{Q}{\Delta t_m \cdot F_a} \tag{3}$$

The results of experiments were compared with predictions by Equations proposed by Arsenyeva et al. (2012). For calculating the friction factor in the channels for hot (index *i*=1) and cold (index *i*=2) sides by following Equation:

$$\zeta_i = 8 \cdot \left[\left(\frac{12 + p_{2i}}{\mathrm{Re}_i} \right)^{12} + \frac{1}{(A_i + B_i)^{\frac{3}{2}}} \right]^{\frac{1}{12}},$$

$$A_i = \left[p_{4i} \cdot \ln \left(\frac{p_{5i}}{\left(\frac{7 \cdot p_{3i}}{\mathrm{Re}_i} \right)^{0.9} + 0.27 \cdot 10^{-5}} \right) \right]^{16}; \quad B_i = \left(\frac{37530 \cdot p_{1i}}{\mathrm{Re}_i} \right)^{16} \tag{4}$$

where p_{1i}, p_{2i}, p_{3i}, p_{4i}, $p_{5i}$are the parameters defined by channel corrugation form.

$$p_{1i} = \exp\left(-0.15705 \cdot \beta_i\right); \quad p_{2i} = \frac{\pi \cdot \beta \cdot \gamma_i^2}{3}; \quad p_{3i} = \exp\left(-\pi \cdot \frac{\beta_i}{180} \cdot \frac{1}{\gamma_i^2}\right);$$

$$p_{4i} = \left(0.061 + \left(0.69 + tg\left(\beta_i\right)\right)^{-2.63}\right) \cdot \left(1 + \left(1 - \gamma_i\right) \cdot 0.9 \cdot \beta_i^{0.01}\right); \quad p_{5i} = 1 + \frac{\beta_i}{10}$$

$\gamma_i = 2 \cdot b / S$ is the corrugation doubled plate spacing b to corrugation pitch S ratio; β_i is the corrugations inclination angle, degrees; $\mathrm{Re}_i = w_i \cdot d_e \cdot \rho_i / \mu_i$is the Reynolds number; d_e is the equivalent diameter of the channel, $d_e = 2 \cdot b$, m; w_i is the stream velocity in the channel, m/s; μ_i is the dynamic viscosity of fluid, Pa·s; $\rho_i$is the density of the fluid, kg/m^3.
The film heat transfer coefficients are estimated using for hot (1) and cold (2) sides the following relation:

$$Nu_i = 0.065 \cdot \mathrm{Re}_i^{6/7} \cdot \left(\psi_i \cdot \zeta_i / F_X\right)^{3/7} \cdot \mathrm{Pr}_i^c \cdot \left(\mu_i / \mu_{wi}\right)^{0.14} \tag{5}$$

Where μ_i and μ_{wi} are the dynamic viscosities for the stream and wall temperatures, Pa·s; $\mathrm{Nu} = h_i \cdot d_e / \lambda_i$ is the Nusselt number; $\lambda_i$is the thermal conductivity of the fluid, W/(m·K); h_i is the film heat transfer coefficient, W/(m^2·K); Pr_i is the Prandtl number; ζ_i is the friction factor accounting for total pressure losses in the channel, calculated by Equation (2); ψ_i is the share of pressure loss due to friction on the wall in total loss of pressure; F_x is the coefficient of surface area enlargement due to corrugation.
The value of ψ_i is estimated according to relation:

$$A_{1i} = 380 / \left[tg(\beta_i)\right]^{1.75}; \left[\begin{array}{l} \psi_i = \left(\mathrm{Re}_i / A_{1i}\right)^{-0.15 \cdot \sin(\beta_i)} \quad at\ \mathrm{Re}_i > A_{1i} \\ \psi_i = 1 \quad at\ \mathrm{Re}_i \le A_{1i} \end{array}\right. \tag{6}$$

The calculated value of overall heat transfer coefficient is determined as:

$$\mathrm{U_{cl}} = \left(\frac{1}{\mathrm{h_1}} + \frac{1}{\mathrm{h_2}} + \frac{\delta_\mathrm{w}}{\lambda_\mathrm{w}}\right)^{-1} \tag{7}$$

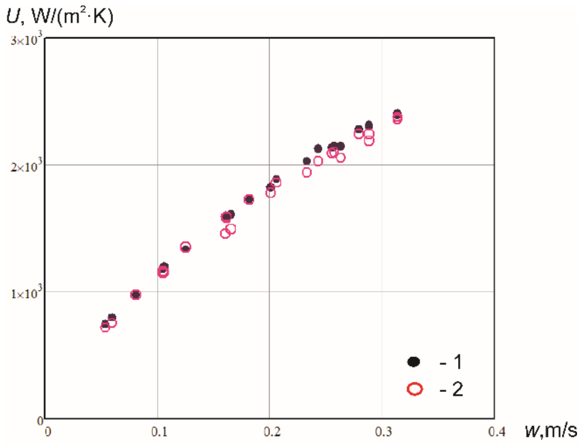
The influence of fluid velocity on overall heat transfer coefficient is shown on graph in Figure 3. There presented experimental and calculated values. The calculated values are somewhat higher than experimental ones, but the discrepances are not exciding +8 %. Beside some experimental error it can be explained by reduction of heat transfer effectiveness due to cross flow. In Figure 4 is presented the dependance of the WPHE model heat transfereffectivness $\varepsilon = (t_{11} - t_{12})/(t_{11} - t_{21})$ from the number of heat transfer units (NTU), expressed through calculated overall heat transfer coefficient U_{cl} by following Equation:

$$\mathrm{NTU} = \frac{\mathrm{F_a} \cdot \mathrm{U_{cl}}}{\mathrm{G} \cdot \mathrm{c_{p1}}} \tag{8}$$

The ε-NTU experimental values are situated below ε-NTU values calculated by Equation for counter-current flow, but can be approximated on a safe side by ε-NTU relation for cross flow with one fluid mixed another unmixed (see Shah and Sekulic, 2003):

$$\varepsilon = 1 - \exp\left[\frac{-1 + \exp\left(-\mathrm{R} \cdot \mathrm{NTU}\right)}{\mathrm{R}}\right] \tag{9}$$

where $R = G_1 \cdot c_{p1} / (G_2 \cdot c_{p2})$. In our case R=1, but it can be approximated also for other cases with R<1. Using this relation (Eq.9) the ε-NTU method proposed by Arsenyeva et al. (2016) can be used for calculation of considered multipass WPHEs.



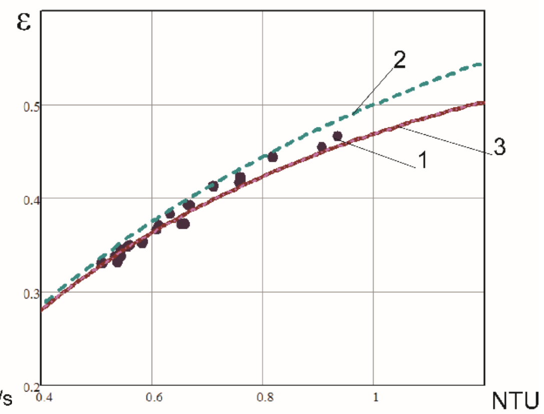
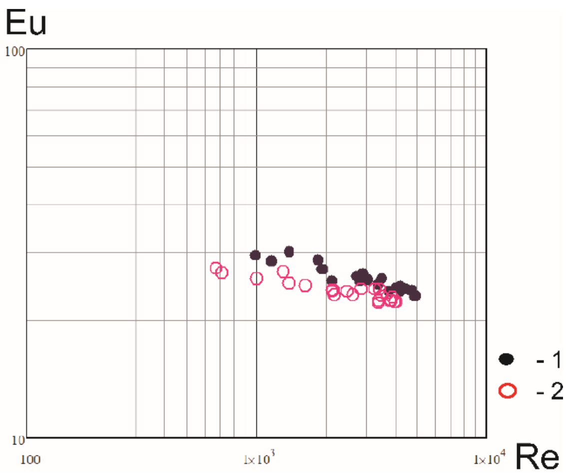


Figure 3: The influence of flow welocity on overall heat transfer coefficient: 1 – prediction; 2 – experiment

Figure 4: The ε-NTU relation for WPHE model: 1- experiment; 2 – counter-current; 3 – cross flow mixed – unmixed



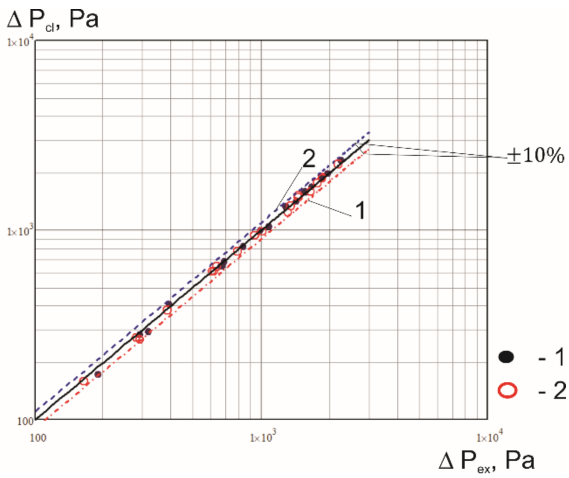


Figure 5: The experimental data for pressure 1 – hot stream; 2 – cold stream

Figure 6: The comparison of experimental losses: pressure drop with predicted by Eq.(10): 1 – hot stream; 2 – cold stream

For correct design of the WPHE the data for pressure drop calculation is required. In Figure 5 are shown the experimental data on $Eu_i = \Delta P_i/(\rho_i \cdot w_i^2)$ number for both hot and cold streams in WPHE model. The data for channel with higher corrugation angle β_1=50° are demonstrating higher hydraulic losses than for channel with β_2=40°. For calculation of the pressure losses the division of the channel on main corrugation field and distribution zones at the inlet and outlet of the inter-plate channel is used. Arsenyeva et al. (2013) introduced the coefficient of local hydraulic resistance in those zones ζ_{DZi}, with which the pressure drop inone pass of WPHE, represented by the model under investigation, can be calculated by Eq.(10):

$$\Delta p_i = \zeta_i \cdot \frac{L}{d_e} \cdot \frac{\rho_i \cdot w_i^2}{2} + \zeta_{DZi} \cdot \frac{\rho_i \cdot w_{enx.i}{}^2}{2} \tag{10}$$

where $w_{enx.i}$ is the velocity at channel entrance/exit.

The comparison with experimental data gave following values for coefficient of local hydraulic resistance in distribution zones ζ_{DZ1}=11 and ζ_{DZ2}=17. The comparison of calculation by Eq.(10) with experimental data is shown by graph in Fig.6. The error not exceed 10 %.

4. Conclusions

The experimental study of heat transfer and pressure drop in the model of WPHE has confirmed the validity of Equations proposed for PHE channels of different geometry in case of cross flow. It is also estimated the dependence of PHE heat transfer effectiveness (ε) from the number of heat transfer units (NTU) in one pass of WPHE with cross flow of streams. These Equations can be used in the method for calculation of WPHE with overall counter flow and cross flow inside separate passes.

The validity of the proposed Equations and developed method for WPHE design was confirmed by comparison with the data of tests on WPHE installed in ammonia synthesis column at industrial enterprise of ammonia production. WPHE was installed in existing synthesis column of ammonia unit instead of shell-and-tube heat exchanger. The use of WPHE instead shell-and-tube unit enable to cut down the volume occupied by heat exchanger in high pressure shell of ammonia synthesis column and allows increase of the volume of catalyst. It leads to 15% rise of ammonia output.

Acknowledgments

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