Calculation Methods for Gasket Plate Heat Exchangers Used in Vegetable Oil Manufacture

Comparative study

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Gasket plate heat exchangers are extensively used in the vegetable oil manufacture, due to their robust construction and effective heat transfer. Knowledge of heat transfer coefficients is useful in dimensioning of new industrial plants or in technological analysis of the actual installations. The aim of this work was to find the best mathematical equations in literature for the calculation of heat transfer coefficients in the gasket plate heat exchanger equipped with 30° chevron angle plates. For this, experimental coefficients were determined in an industrial plant processing sunflower oil and then compared with values obtained by calculation with mathematical equations from literature. The best fitting was with Kumar model.

Keywords: gasket plate heat exchanger, heat transfer coefficient, Nusselt number, Reynolds number, Prandtl number

The gasket plate heat exchangers have applications in the refining vegetable oil process. By refining, the crude oil is converted into marketable oil both in terms of sensorial quality and storage stability. The main refining operations are degumming, neutralization, bleaching, winterizing, deodorization and polishing [1-3]. The gasket plate heat exchanger is used in these refining units. It consists of a pack of thin corrugated metal plates with portholes for the passage of the fluids. Each plate contains a bordering gasket, which seals the channels formed when the plate pack is compressed and mounted on a frame. The space between two adjacent plates forms flow channels, and the gaskets are arranged so that the two fluids flow alternately in the formed channels. In the four corners of the plates, circular orifices are made to allow the passing of the fluids to the channels [4]. This geometry ensures higher heat transfer coefficients than the tubular heat exchagers. Also, gasket plate heat exchangers type chevron were introduced in the food industry in 1930 because of their robust construction allowing easy cleaning of fouling.Over the years, the development of these types of heat exchangers has continued at higher working capacities, high working temperatures and pressures [5].

In literature, a number of experiments were conducted on gasket plate heat exchangers chevron type.

Muley and Manglik [6] used in their experimental investigations the water as working fluid. They studied the heat transfer and pressure drop in gasket plate heat exchangers with different plates chevron angles: 30°,60° and mixed 30°/ 60°. The Reynolds number range was 600 to 10^4. Both the Nusselt number and friction factor increased with increasing chevron angle, however friction factor experienced faster growth compared Nusselt number. Based on their experimental data, they reported correlations on heat transfer and pressure drop.

Heavner and collaborators [7] investigated effects of mixed chevron angles for a wide range of plates. They conducted experiments for $\beta = 23^0/23^0, 23^0/45^0, 45^0/45^0, 23^0/90^0$ and $45^0/90^0$ and $400 < Re < 10{,}000$. Heat transfer and pressure drop were found to increase with increasing chevron angle. Their results are quite different from other researchers.

Pinto and Gut [8] developed an optimization method to determine the best configuration of gasket plate heat exchangers. The main objective was to select the configuration with minimum heat transfer area, taking also into account constraints for the number of channels, the pressure drop, the fluid velocity and the heat exchange efficiency. The configuration of plate heat exchanger is defined by six parameters: the number of channels, the number of passes on each side, the fluid distribution location, the feed position and the type of flow in the channels.

Warnakulasuriya *et al.* [9], investigated the heat transfer and pressure drop for viscous salt solutions in a heat exchanger. The main purpose of this experiment was to establish the equations for calculating the heat transfer coefficient and pressure drop, to optimize the operating parameters of the experiment. Reynolds number ranged between 250 and 1100, Prandtl number varied between 82 and 174 and the overall heat transfer coefficient ranged between 970 - 2270 (W / m^2 . °C).

Naik and Matawala [10] carried out experimental investigation on a type chevron gasket plate heat exchanger. In this case, the plates had different corrugation angles (30, 45, 60°) and fluids used were oil and water. Reynolds number varied between 50 and 10 000 and Prandtl number ranged from 3 - 75. Based on the experimental data, a correlation was worked out for Nusselt number as a function of Reynolds number, Prandtl number and chevron angle. The best results have been obtained for a plate heat exchanger whose corrugation angle was 60°. The relationship proposed in this experiment was compared to other relationships in the literature, but only for the plates crimping angle of 60°.

In the present work, the experimental heat transfer coefficients obtained in an industrial plant processing vegetable oil were compared with values proceeding from calculations with different equations, in order to select the best fitting correlation to be used in heat exchangers design.

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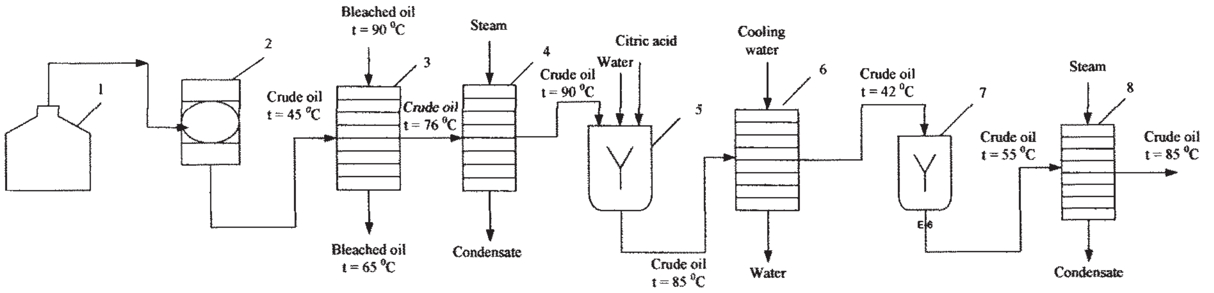

Fig. 1. Schematic diagram of the experimental setup

Experimental part

Experimental set-up

A commercially gasket plate heat exchangers type has been used in this study, from a local company refining the sunflower oil. Experiments are made for every heat exchanger in the process, at different flow rates of the oil.

The experimental setup is shown in figure 1. It respresents the degumming unit of the vegetable oil manufacture. The crude oil (CO) is pumped from the buffer tank (1) through the filter (2) for a raw filtration, to a gasket plate heat exchanger (3) where it is heated by oil coming from the bleaching unit. If the oil temperature after (3) did not reach 90°C, it would be warmed up to this temperature in a second heat exchanger (4), using 3 bar saturated steam. After reaching 90°C, the oil is mixed with the citric acid solution in a mixing-reaction unit (5) where the phosphatides are precipitated. The oil is then cooled to 42 °C with water in the heat exchager (6) and sent to the mixer (7) where the residence time is 60 min and the phoshpatides are hydrated. During this process, the temperature increases to 55°C. The oil is then warmed up to 85°C with 3 bar saturated steam in the heat exchager (8), in view of drying and phosphatides removing from the oil.

The plate heat exchanger PHE #1 has a two pass configuration and U-arrangement, since thet exchangers PHE #2, PHE #3 and PHE #4 are in one pass configuration with countercurrent flow.

The basic geometry of the heat exchangers is shown in figure 2. Chevron plates of the heat exchanger are made of stainless steel (AISI 316).

Im portant structural data of chevron plates used in the study are shown in table 1.

Experimental data collection and processing

Experimental data were collected for each apparatus, this concerning flow rates and temperature on both fluids. There were four experimental periods, corresponding to four different flow rates of the crude oil: 1.736 kg/s; 2.049 kg/s; 2.475 kg/s; 2.713 kg/s. This allowed us to collect more data for the intended comparison between experimental and literature data.

The physical properties of the fluids were experimentally determined or calculated, as follows. The density and the

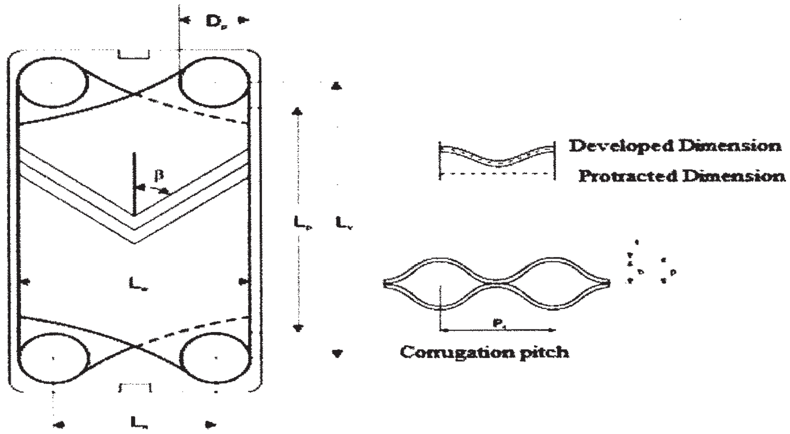

Fig. 2. Basic geometric characteristics of chevron plate [3]

Table 1
STRUCTURAL DATA OF CHEVRON PLATES

Geometric characteristic	PHE #1	PHE #2	PHE #3	PHE #4
Vertical distance between centers of ports, L_v (mm)	1070	620	1070	381
Port-to-port channel length, L_p (mm)	858	436	858	253
Plate width, L_w (mm)	450	334	450	198
Horizontal distance between centers of ports, L_h (mm)	238	150	238	70
Port diameter, D_p (mm)	212	184	212	128
Plate pack dimension, L_c (mm)	176	86	111	65
Plate thickness, δ (mm)	0.6	0.6	0.6	0.6
Corrugation pitch, p (mm)	3.08	3.185	3.171	2.708
Mean channel spacing, b (mm)	2.48	2.585	2.571	2.108
Hydraulic diameter, d_h (mm) $d_h = 2b/\varphi$	4.24	4.418	4.395	3.604
Plate surface enlargement factor, φ	1.17	1.17	1.17	1.17
Effective area of plate heat exchanger, A_e (m^2)	18.2	3.7	11.2	3.3
Chevron angle, degrees	30°	30°	30°	30°

dynamic viscosity of vegetable oil have been determined in laboratory, in the range temperature of 20 – 80 °C. The density and the dynamic viscosity were measured with an Anton-Paar viscometer SVM 3000 type. The results are presented in [11]. The thermal conductivity and the specific heat were calculated with the relationships developed by Choi and Okos [12].

Heat transfer coefficients

For the calculation of heat transfer coefficients, the well known theory was applied [13,14]. The termophysical properties of the fluids were considered at the mean temperatures, as follows:

$$t_{c,b} = \frac{t_{c,in}+t_{c,out}}{2} \qquad [°C] \qquad (1)$$

$$t_{h,b} = \frac{t_{h,in}+t_{h,out}}{2} \qquad [°C] \qquad (2)$$

$$t_{w} = \frac{t_{c,b}+t_{h,b}}{2} \qquad [°C] \qquad (3)$$

where, $t_{c,b}$ represents the bulk temperature of the cold side (^{0}C), $t_{c,in}$ - inlet temperature of the cold side (^{0}C), $t_{c\ out}$ – outlet temperature of the could side (^{0}C), $t_{h,b}$ - bulk temperature of the hot side (^{0}C), $t_{h,in}$ – inlet temperature of the hot fluid (^{0}C), $t_{h,out}$ – outlet temperature of the hot fluid (^{0}C), t_w - wall temperature (^{0}C) .

The experimental overall heat transfer coefficient, k_{exp}, is determined from equation (4):

$$Q = k_{exp} \cdot A \cdot \Delta t_{LMTD} \qquad \left[\frac{W}{m^2 \cdot K}\right] \qquad (4)$$

where, Q represents the heat transfer rate (W, A – effective heat transfer area (m^2) from table 1 , Δt_{LMTD} – log-mean temperature difference (°C) between the hot and the cold fluids.

The heat transfer rate in equation (4) can be calculated on the cold fluid side , with equation (5):

$$Q_c = m_c^{\circ} \cdot c_{p,c} \cdot (t_{c,in} - t_{c,out}) \qquad (5)$$

where, Q_c represents the heat transfer rate on cold side (W), m_c^{o} - mass flow rate on cold side (kg . s^{-1}), $c_{p,c}$ - specific heat capacity of the cold fluid (J . kg^{-} K^{-1}), $t_{c,in}$ - inlet temperature of the cold fluid (^{0}C), $t_{c,\ out}$ – outlet temperature of the cold fluid (^{0}C).

Also, the heat transfer rate can be calculated on the hot fluid side, with equation (6):

$$Q_h = m_h^{\circ} \cdot c_{p,h} \cdot (t_{h,in} - t_{h,out}) \qquad (6)$$

where, Q_c represents the heat transfer rate on hot side (W), m_h^{o} - mass flow rate on hot side (kg . s^{-1}), $c_{p,h}$ - specific heat capacity (J . kg$\cdot$ K^{-1}), $t_{h,in}$ - inlet temperature of the hot side (^{0}C), $t_{h,\ out}$ – outlet temperature of the hot fluid (^{0}C).

The overall heat transfer coeficient can also be calculated , with equation (7):

$$k_{calc} = \frac{1}{\frac{1}{\alpha_c}+\frac{1}{\alpha_h}+\frac{\delta}{\lambda}} \qquad \left[\frac{W}{m^2 \cdot K}\right] \qquad (7)$$

where, k_{calc} – represents overall heat transfer coefficient (W/m^2 . K), α_c – convective heat transfer coefficient for the cold side (W/m^2 . K), α_h - convective heat transfer coefficient for the hot side (W/m^2 . K), δ – plate tickness (m), λ – thermal conductivity for stainless steel (W/m . K).

For fouling taken into account, the overallheat transfer coefficient is obtained from equation (8):

$$k_f = \frac{1}{\frac{1}{\alpha_c}+\frac{1}{\alpha_h}+\frac{\delta}{\lambda}+R_{f,c}+R_{f,h}} \qquad \left[\frac{W}{m^2 \cdot K}\right] \qquad (8)$$

where k_f represents overall heat transfer coefficient in fouling conditions (W/m^2 . K), $R_{f,c}$ -total fouling resistance on the cold side (m^2 . K/W), $R_{f,h}$ - total foling resistance on the hot side (m^2 . K/W).

The log-mean temperature difference is calcuated with equation (9):

$$t_{LMTD} = \left\{ \frac{(t_{h,in}-t_{c,out})-(t_{h,out}-t_{c,in})}{ln\left[\frac{(t_{h,in}-t_{c,out})}{(t_{h,out}-t_{c,in})}\right]} \right\} \qquad (9)$$

The Reynolds number, which is a non-dimensional number based on channel mass velocity, hydraulic diameter of the channl and dynamic viscosity is defined as:

$$Re = \frac{G_{ch} \cdot d_h}{\mu} \qquad (10)$$

where Re represents the Rey . s^{-1}), d_h - hydraulic diameter (m), - dynamic viscosity (Pa’”s).

The mass velocity in the channel, G_{ch} , is given by equation (11):

$$G_{ch} = \frac{\dot{m}}{N_{cp} \cdot b \cdot L_w} \qquad \left[\frac{kg}{m^2 \cdot s}\right] \qquad (11)$$

where $\dot{m}$ represents the total mass flow rate in the opening port (kg . s^{-1}), N_{cp} - number of channels per pass, b - mean channel spacing (m), L_w - plate width (m).

In the calculations of convective heat transfer coefficient, the Nusselt number is a function of Reynolds number, Prandtl number and the ratio of dynamic viscosities at bulk and the wall temperature, as in equation (12):

$$Nu = C \cdot Re^a \cdot Pr^b \cdot \left(\frac{\mu}{\mu_w}\right)^c \qquad (12)$$

where Nu represent Nusselt number, C, a, b and c - coefficients

In heat transfer within a fluid film, the Nusselt number is the ratio of convective to conductive heat transfer across the boundary. Nusselt number is then defined by equation (13):

$$Nu = \frac{\alpha_{c,h} \cdot d_h}{\lambda_f} \qquad (13)$$

where Nu represents Nusselt number, $\alpha_{c,h}$ - convective heat transfer coefficient for the cold/hot side (W/m^2 . K), λ_f - thermal conductivity of the fluid (W/m . K).

In order to calculate the convective heat transfer coefficients, particular forms of the equation (12) for Nusselt number are used. In table 2, the following correlations found in literature are presented: the Kumar correlation [14], Muley et al [14], Bond [15], Martin [16], and Buonopane and Troupe [17].

Results and discussions

The experimental overall heat transfer coefficients were calculated with equation (4). Then, the overall heat transfer coefficients were calculated with Eq. (7), after and using the criterial equation (12) to calculate convective coefficients on both sides of the transfer area, in the particular forms presented in table 2.

The experimental values were compared with the calculated values obtained by applying the correlations

Reference	Chevron angle, (°)	Correlation	
		Nusselt number	Friction factor
Kumar	30	$Nu = 0.348 \cdot Re^{0.663} \cdot Pr^{1/3} \cdot \left(\frac{\mu}{\mu_w}\right)^{0.17}$	$f = \frac{19.400}{Re^{0.589}}$
Muley et all	30	$Nu = 0.44 \cdot \left(\frac{\beta}{30}\right)^{0.38} \cdot Re^{0.5} \cdot Pr^{1/3} \cdot \left(\frac{\mu}{\mu_w}\right)^{0.14}$	$f = \left(\frac{\beta}{30}\right)^{0.83} \cdot \left[\left(\frac{30.2}{Re}\right)^{5} + \left(\frac{6.28}{Re^{0.5}}\right)^{5}\right]^{0.2}$
Bond	30	$Nu = 0.329 \cdot Re^{0.529} \cdot Pr^{0.33} \cdot \left(\frac{\mu}{\mu_w}\right)^{0.17}$	$f = 3{,}01 \cdot Re^{-0.457}$
Bogaert and Bolcs	30	$Nu = B_1 \cdot Re^{B_2} \cdot Pr^{1/3 \cdot e^{\left(\frac{6.4}{Pr+30}\right)}} \cdot \left(\frac{\mu}{\mu_w}\right)^{\frac{0.3}{(Re+6)^{0.125}}}$ $20 < Re < 50, B_1 = 0.0875, B_2 = 1$ $50 < Re < 80, B_1 = 0.4223, B_2 = 0.6012$	-
Muley	30	$Nu = 0.44 \cdot \left(\frac{6 \cdot \beta}{\pi}\right)^{0.38} \cdot Re^{0.5} \cdot Pr^{1/3} \cdot \left(\frac{\mu}{\mu_p}\right)$	-
Martin	30	$Nu = 0.122 \cdot Pr^{1/3} \cdot \left(\frac{\mu}{\mu_w}\right)^{1/6} \cdot (f \cdot Re^2 \cdot \sin\beta)^{0.374}$	$\frac{1}{\sqrt{f}} = \frac{\cos\beta}{\left(0.18 \cdot \tan\beta + 0.36 \cdot \sin\beta + \frac{f_0}{\cos\beta}\right)^{1/2}} + \frac{1-\cos\beta}{\sqrt{3.8 \cdot f_1}}$ $Re \leq 2000, f_0 = \frac{64}{Re}, f_1 = \frac{597}{Re} + 3.85$
Buonopane and Troupe	30	$Nu = 0.45\left(Re \cdot Pr \cdot \frac{d_h}{L_v}\right)^{0.333} \cdot \left(\frac{\mu}{\mu_w}\right)^{0.14}$	$f = \frac{2.5}{Re^{0.3}}$

Table 2
CORRELATIONS FOR NUSSELT NUMBER AND FRICTION FACTOR FROM LITERATURE

Heat exchanger	Mass flow rate on oil side, kg s^{-1}	k_{exp} W $m^{-2}K^{-1}$	k_{calc} Kumar Model and rel. err.(%)	k_{calc} Mulley Model and rel. err.(%)	k_{calc} Bond Model and rel. err.(%)	k_{calc} Martin Model and rel. err.(%)	k_{calc} Buonapane& Troupe Model and rel. err.(%)
PHE#1	1.736	244	280	217	185	210	198
			+14.8	-11.1	-24.2	-13.9	-18.9
	2.049	422	302	231	199	222	207
			-28.4	-45.3	-52.8	-47.4	-50.9
	2.475	506	326	247	214	235	207
			-15.8	-51.2	-57.7	-53.6	-57.1
	2.713	559	340	256	222	243	222
			-39.2	-54.2	-60.3	-56.5	-60.3
PHE#2	1.736	439	618	487	408	471	390
			+29.0	+10.9	-7.1	+7.3	-11.2
	2.049	520	656	499	417	485	397
			+26.2	-4.0	-19.8	-6.7	-23.7
	2.475	628	741	512	472	500	405
			+18.0	-18.5	-24.8	-20.4	-35.5
	2.713	687	767	518	432	508	409
			+10.4	-24.6	-59.0	-26.1	-40.5
PHE#3	1.736	558	399	221	271	314	285
			-28.5	-60.4	-51.3	-43.7	-48.9
	2.049	623	429	231	289	323	296
			-31.1	-62.9	-53.6	-48.2	-52.5
	2.475	746	463	240	327	347	307
			-37.9	-67.8	-56.2	-53.5	-58.8
	2.713	824	482	246	339	357	314
			-41.5	-70.1	-58.9	-56.7	-61.7
	1.736	479	618.4	339	299	336	346
			+29.1	-29.2	-37.6	-29.9	-27.8

Table 3
COMPARISON BETWEEN THE EXPERIMENTAL VALUES OF THE OVERALL HEAT TRANSFER COEFFICIENT (k_{exp}) AND THE VALUES CALCULATED WITH DIFFERENT MODELS (k_{calc})

PHE#4	2.049	565	656	347	306	317	353
			+16.1	-38.6	-45.8	-43.9	-37.5
	2.475	678	741	354	313	331	358
			+8.5	-47.8	-53.8	-51.2	-47.2
	2.713	748	767	358	316	334	361
			+2.5	-52.1	-57.8	-55.3	-51.7

Table 3 (continuated)

produced with Kumar [14], Muley et al [14], Bond [15], Martin [16] and Buonopane and Troupe [17].

For comparison, the relative error (RE) was calculated with the equation (18):

$$RE = \frac{k_{exp}-k_{calc}}{k_{exp}} \cdot 100 \quad [\%] \qquad (18)$$

where:

k_{exp} - the experimental value of the overall heat transfer coefficient calculated with Kumar correlation;

k_{calc} - calculated value of the overall heat transfer coefficient with models proposed in the literature.

The overall transfer coefficients and the relative errors are presented in table 3.

As seen in table 3, the experimental values of the overall heat transfer coefficient increases with the mass flowrate of the crude oil, since the main target of the heat transfer was to mentain a certain temperature difference on each circuit in the apparatus. So, the flowrate in the second circuit increased as well and Re number with it, and by consequence the heat transfer improves, illustrated by the values of the overall heat transfer coefficient.

Calculated values of the overall heat transfer coefficient with different models from literature are pretty far from the experimental ones with Kumar model giving better results. Kumar model gives relative errors both positive and negative thus indicating a distribution of k_{calc} around the k_{exp}. The other models give almost all negative errors and $k_{calc} < k_{exp}$. This indicates that the Kumar model can be a starting point for finding a good model for the calculation of the overall heat transfer coefficients.

Conclusions

Knowledge of heat transfer coefficients in gasket plate heat exchangers is important for the design of new equipment or for the analysis of the existing one. In this work, the heat transfer coefficients were experimentally determined in industrial conditions and then calculated with mathematical equations proceeding from scientific works. This allowed us a comparison in order to find the best fitting model in the literature.

The evaluated models give big errors so they are unreliable for the prediction of the overall heat transfer coefficients in case of gasket plate heat exch angers used in vegetable oil refining. This is probably due to the fact that mathematical models were worked out, in most cases, in water-water heat exchange.

The Kumar model fits better and it could be a starting point for the development of a new and accurate model. For this, the experimental base has to be extended to different fluids (crude vegetable oil of different origins, cooling water, steam and condensate).

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