Model Validation for the Heat Transfer in Gasket Plate Heat Exchangers Working with Vegetable Oils

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Abstract: Many models for accurately predicting the performance of gasket plate heat exchangers were developed in the last decades, grouped in three categories: empirical, semi-analytical or theoretical/numerical, with the view to saving materials and energy through correct design of industrial equipment. This work addresses one such model, namely Lévêque correlation modified by Martin and by Dović, which is promising due to the correct assumption of the flow in sine duct channels and the consideration of energy losses caused by flow reversal at plate edges and the flow path changing when entering the chevron angle. This model was validated by our own experimental data under industrial conditions for vegetable oils processing, both in laminar flow ($Re = 8–42$) and fully developed turbulent flow ($Re = 446–1137$). Moreover, in this study, particular values for constants/parameters of the model were determined for the corrugation inclination angle relative to vertical direction equal to $30^\circ$. Through statistical analysis, this study demonstrates that this particularized form of the generalized Lévêque correlation can be used with confidence.

Keywords: vegetable oil; chevron plate heat exchanger; heat transfer; semi-analytical model validation

1. Introduction

In technological installations, heat exchangers are parts of the process flow; they ensure the heating, cooling or heat recovery from hot fluxes, as the technology requires. The use of plate heat exchangers (PHE) is desirable since they have a simple compact construction, are reliable, and very efficient at small temperature difference between fluids. Additionally, they have low fabrication and maintenance costs. This type of heat exchanger is a good choice in vegetable oils refining where the high viscosity of the fluids represents a high resistance to flowing and heat transfer.

Gasket plate heat exchangers were introduced in vegetable oil processing due to their specific geometry [1], with small hydraulic diameter and corrugations, therefore inducing enhanced Reynolds numbers and enlarging the heat transfer area, resulting in the intensification of heat transfer [2].

Numerous studies have been carried out to explore the influence of geometric characteristics on the heat transfer [3–8]. The results show that geometric parameters such as chevron/herringbone angle, surface enlargement factor, and the corrugation profile are important factors influencing the thermal-hydraulic behavior. Another influencing factor is the flow maldistribution [9–11]. S. Al-Zahrani, et al. [12] showed that the maldistribution of the fluids in ports contributes to the performance of the apparatus and their position should be rethought by introducing a longitudinal baffle, so that the cross sectional area changes radically. The simulations proved that friction factors for the modified PHE are 4.5–7 fold higher than those of conventional corrugate plate apparatus with effect on the overall thermal performance.

The experiments on fluid hydrodynamics and thermal transfer in this type of heat exchanger allowed for the development of correlations in accordance with the similitude...
theory (theorem π). In older [13–15] or in more recent studies [16,17], empirical correlations are included to estimate Nusselt number (heat transfer) as a function of Reynolds number (flow regime), taking also into account the physical properties of the fluids (Prandtl number). The friction factors, $f$, are strongly correlated with the pressure drop; they are decisive for the flow regime and heat transfer, so their accurate estimation is of great importance. Usually, they are calculated as functions of Reynolds number, some relationships adding the chevron angle $\beta$ and surface enhancement factor $\varphi$ [14,18].

The analytical (numerical) models also contributed to predict the performance of this type of exchangers. In general, analytical models apply the mass, momentum and energy balance to an element of fluid flowing in the channel and consist in a set of differential equations solved by general CFD or dedicated software such as, gProms or free version of Alfa Laval soft. Thus, Gut and Pinto [11] developed a model including six constructive parameters of PHEs, and their simulations results are: the temperature profile and the distribution of the overall heat transfer coefficients in the apparatus, the thermal effectiveness, the pressure drop. Zhong et al. [19] established the governing equation for the flow in channels, developed the correlation for $f$ as function of $Re$, at $Re < 50$ and validated the model with their own experimental data.

Another approach involves the use the thermal transfer unit notion in order to determine the heat exchange effectiveness ($\varepsilon$) and the overall heat transfer coefficient ($U$). Arsenyeva et al. [20] developed such a model regarding the PHE viewed as a system of one-pass block of plates with equal conditions for all channels.

Semi-analytical models improve the criterial correlations $Nu = f(Re, Pr)$ by including the influence of some geometric parameters. For example, Martin [21] included the chevron angle $\beta$, the corrugation depth $b$ and the corrugation wavelength $l$. Martin’s model modified the generalized Lévêque correlation for $Nu$ by taking into account the superposition of two flow components in the channel, the longitudinal component moving in the main direction of the flow and the furrow component following the corrugation path, but calculated the friction factor with relations developed for a circular tube. Then, Dovič et al. [22] introduced empirical correction factors to the classical Lévêque correlation for $Nu$ calculation, counting for energy losses due to flow reversal at plate edges and flow path changing when entering the chevron angle. Piper et al. [23] designed new equations for heat transfer and pressure drop, considering two zones in the flow pattern characteristic for pillow plates channels, with validity for $1000 \leq Re \leq 8000$ and $1 \leq Pr \leq 150$.

The validation of all these models must be performed on experimental data, if possible, on equipment at full scale. For convenience, in many studies, the experimental fluid is hot water/cold water. Rarely, other fluids are considered, e.g., high viscosity refrigeration liquids [19,22], salt solutions [15] or unspecified (mineral or vegetable) oil [16].

In the present work, the model of Dovič et al. [22] will be validated on experimental data obtained in industrial conditions, for vegetable oil processing.

2. Experimental Setup and Procedure

2.1. Equipment and Data Collection Procedure

For the development of a mathematical model describing the heat transfer in chevron plate heat exchangers, four gasket chevron plate exchangers (HE) were tested, with characteristics described in Table 1 and Figure 1. The flow of both fluids in the pre-heater HE#1 is in counter-current with two passes, and in the other three HE is one-pass counter-current.
Table 1. The technological role, fluids and geometrical characteristics of the HE.

| Geometrical Characteristics of Chevron Plates | Symbol | Heat Exchanger #1 | Heat Exchanger #2 | Heat Exchanger #3 | Heat Exchanger #4 |
|--------------------------------------------|--------|-------------------|-------------------|-------------------|-------------------|
| Technological role                          | RO—raw oil | Oil pre-heating   | Cooler            | Cooler            | Cooler            |
| Fluids                                      | BO—bleached oil | BO/water         | BO/water         | WO/water         |
| Vertical distance between centres of ports  | \( L_v \) | 1070 (mm)         | 1070 (mm)         | 1070 (mm)         | 1070 (mm)         |
| Plate length between ports                  | \( L_p \) | 858 (mm)          | 858 (mm)          | 858 (mm)          | 858 (mm)          |
| Plate width                                 | \( L_w \) | 450 (mm)          | 450 (mm)          | 450 (mm)          | 450 (mm)          |
| Horizontal length between centres of ports  | \( L_h \) | 238 (mm)          | 238 (mm)          | 238 (mm)          | 238 (mm)          |
| Port diameter                               | \( D_p \) | 212 (mm)          | 212 (mm)          | 212 (mm)          | 212 (mm)          |
| Plate thickness                             | \( \delta \) | 0.6 (mm)          | 0.6 (mm)          | 0.6 (mm)          | 0.6 (mm)          |
| Plate pitch                                 | \( p \) | 3.08 (mm)         | 3.17 (mm)         | 3.14 (mm)         | 3.14 (mm)         |
| Corrugation depth (amplitude of sinusoidal duct) | \( b \) | 2.48 (mm)         | 2.57 (mm)         | 2.54 (mm)         | 2.55 (mm)         |
| Surface enlargement factor                  | \( \varphi \) | 1.17             | 1.17             | 1.17             | 1.17             |
| Hydraulic diameter \((-2 b/\varphi)\)       | \( d_h \) | 4.24 (mm)         | 4.396 (mm)        | 4.34 (mm)         | 4.5 (mm)          |
| Channel cross-sectional free flow area       | \( A_{ch} \) | \( 1.16 \times 10^{-3} (m^2) \) | \( 1.16 \times 10^{-3} (m^2) \) | \( 1.144 \times 10^{-3} (m^2) \) | \( 1.145 \times 10^{-3} (m^2) \) |
| Heat transfer area for a plate               | \( A_1 \) | 0.4517 (m²)       | 0.4517 (m²)       | 0.4517 (m²)       | 0.4517 (m²)       |
| Heat transfer total area                    | \( A_e \) | 18.2 (m²)         | 11.2 (m²)         | 9.2 (m²)          | 19.7 (m²)         |
| Total number of plates                      | \( N_l \) | 57                | 35                | 28                | 63                |
| Effective heat transfer number of plates    | \( N_p \) | 55                | 53                | 26                | 60                |
| Number of fluid passes                     | \( N_{fp} \) | 2                | 1                | 1                | 1                |
| Number of channels for one pass            | \( N_{fp} \) | 14               | 17               | 13.5              | 31                |
| Corrugation inclination angle relative to vertical direction | \( \beta \) | 30°              | 30°              | 30°              | 30°              |

Figure 1. Main geometrical characteristics of a chevron corrugated plate.
The advantage to use large scale equipment for modeling is that models proceeding from laboratory scale experiments can be validated on it, on one hand, and newly developed models are more accurate, on the other hand.

Data collection was performed in a vegetable oil processing installation. It is a fully automated process. Temperatures are controlled at constant values by the cooling water flowrate in case of HE#2, HE#3, HE#4 and by the bleached oil flowrate in case of HE#1. Measured temperatures are shown in the third column of Table 2. The average temperature at the wall is calculated by considering no temperature gradient in it, so the temperature should be the same on both sides of the wall.

Table 2. The temperature of fluids.

| HE # | Fluid  | Temperature in (1) and out (2) of Hot (h) and Cold (c) Fluid, °C | Average Temperature of the Fluid, °C | Average Temperature at the Wall, °C |
|------|--------|---------------------------------------------------------------|--------------------------------------|------------------------------------|
| 1    | RO     | \( t_{c1} = 45; t_{c2} = 76 \)                             | 60.5                                 | 69                                 |
|      | BO     | \( t_{h1} = 90; t_{h2} = 65 \)                             | 77.5                                 |                                    |
| 2    | RO     | \( t_{h1} = 85; t_{h2} = 42 \)                             | 63.5                                 | 48.5                               |
|      | Water  | \( t_{c1} = 30; t_{c2} = 37 \)                             | 33.5                                 |                                    |
| 3    | BO     | \( t_{h1} = 60; t_{h2} = 45 \)                             | 52.5                                 | 42.7                               |
|      | Water  | \( t_{c1} = 30; t_{c2} = 35.6 \)                           | 32.8                                 |                                    |
| 4    | WO     | \( t_{h1} = 110; t_{h2} = 40 \)                            | 75                                   | 55                                 |
|      | Water  | \( t_{c1} = 30; t_{c2} = 40 \)                             | 35                                   |                                    |

In each campaign, the measurements were performed during the steady-state regime which lasts for hours and days. In the case of first two campaigns, there were four such regimes when flowrates and temperatures remained constant. In the third campaign, the installation worked in only one steady-state regime.

2.2. Fluids

It is of great importance to determine the values of the physical properties of fluids with precision, either experimentally or by calculating with reliable mathematical correlations. Therefore, we determined, experimentally, the variation with temperature of density and dynamic viscosity of all three types of oils (raw, bleached, and winterised), covering the whole range in functioning (20–110 °C). The analyses were made with the apparatus Anton Paar, model SVM 3000 (Anton Paar OptoTech GmbH, Seelze-Letter, Germany), using the standard method ASTM D445/ISO 121852.

There were three processing campaigns in the factory, working with three crudes as follows: sunflower oil 1 (SO1), sunflower oil 2 (SO2) and rapeseed oil (RO).

By processing these data with the least squares’ method, the correlations allowing the calculation of oil density and viscosity at the working temperature were obtained. These correlations are given in Tables 3 and 4.
Table 3. Mathematical correlation of vegetable oils density (g × cm$^{-3}$) with temperature (°C).

| Campaign/Oil | Oil Type | Equation                  | Coefficient of Determination, $R^2$ |
|--------------|----------|---------------------------|-------------------------------------|
| I/SO1        | RO       | $y = -0.0128 \times x + 0.9308$ | 0.9917                              |
|              | BO       | $y = -0.0125 \times x + 0.9296$ | 0.9925                              |
|              | WO       | $y = -0.0125 \times x + 0.9296$ | 0.9935                              |
| II/SO2       | RO       | $y = -0.0122 \times x + 0.9290$ | 0.9905                              |
|              | BO       | $y = -0.0126 \times x + 0.9292$ | 0.9935                              |
|              | WO       | $y = -0.0124 \times x + 0.9293$ | 0.9903                              |
| III/RO       | RO       | $y = -0.0125 \times x + 0.9289$ | 0.9937                              |
|              | BO       | $y = -0.0124 \times x + 0.9281$ | 0.9907                              |
|              | WO       | $y = -0.0126 \times x + 0.9289$ | 0.9929                              |

Table 4. Mathematical correlation of vegetable oils viscosity (mPa × s) with temperature (°C).

| Campaign/Oil | Oil Type | Equation                  | Coefficient of Determination, $R^2$ |
|--------------|----------|---------------------------|-------------------------------------|
| I/SO1        | RO       | $y = 68.283 \times x^{-1.307}$ | 0.9959                              |
|              | BO       | $y = 66.491 \times x^{-1.277}$ | 0.9966                              |
|              | WO       | $y = 68.338 \times x^{-1.312}$ | 0.9958                              |
| II/SO2       | RO       | $y = 65.051 \times x^{-1.359}$ | 0.9991                              |
|              | BO       | $y = 59.649 \times x^{-1.226}$ | 0.9995                              |
|              | WO       | $y = 66.409 \times x^{-1.298}$ | 0.9958                              |
| III/RO       | RO       | $y = 71.710 \times x^{-1.325}$ | 0.9959                              |
|              | BO       | $y = 68.285 \times x^{-1.229}$ | 0.9964                              |
|              | WO       | $y = 71.061 \times x^{-1.322}$ | 0.9952                              |

Even though the equations parameters seem very close for different type of oils, small differences can lead to bigger differences in similitude criteria values, especially for Prandtl numbers, with an effect on the accuracy of the mathematical model predicting the global heat transfer coefficients.

The specific heat capacity ($c_p$) and the thermal conductivity ($\lambda$) were estimated by the Equations (1) and (2) of Choi and Okos [24], developed for the vegetable oils:

$$c_p = 1984.2 + 1473.3 \times 10^{-3} \times t - 4800.8 \times 10^{-6} \times t^2 \ [\text{J/kg × °C}]$$  \hfill (1)

$$\lambda = 0.18701 - 2.7604 \times 10^{-4} \times t - 1.7749 \times 10^{-7} \times t^2 \ [\text{W/m × °C}]$$  \hfill (2)

For cooling water, the physical properties at different temperatures (density, viscosity, specific heat capacity and thermal conductivity) are found in reference [25].

2.3. Modeling

The predominant heat transfer mechanism involved in this type of heat exchanger is the forced convection in stationary regime. Most correlations that are used for modeling the heat transfer describe the phenomenon in terms of similitude criteria, following the similitude theory, and the most frequent correlation is exponential (Equation (3)):

$$Nu = a \times Re^b \times Pr^c$$  \hfill (3)
where: $Nu$, $Re$ and $Pr$ are three similitude criteria: Nusselt, Reynolds and Prandtl and coefficients $a$, $b$ and $c$ are to be determined for different flow regimes, for fluids with and without change in phase, for different geometries.

Reynolds number is calculated in relation to mass velocity in a channel $G_{ch}$ [kg m$^{-2}$ s$^{-1}$], hydraulic diameter $d_h$ and dynamic viscosity $\mu$ (Equation (4)).

$$Re = \frac{G_{ch} \times d_h}{\mu} \quad (4)$$

Prandtl number is calculated from physical properties of the fluids with Equation (5).

$$Pr = \frac{c_p \times \mu}{\lambda} \quad (5)$$

$Nu$ number contains the heat transfer coefficient $\alpha$ [W m$^{-2}$ °C$^{-1}$] which can be determined from Equation (6), after calculating $Nu$ with Equation (3):

$$Nu = \frac{\alpha \times d_h}{\lambda} \quad (6)$$

For the forced convection in stationary regime, the coefficients $a$, $b$, $c$ in Equation (3) were determined in many experimental studies but, the Kumar correlation (Equation (7)) is mostly recommended [18,26] to calculate $Nu$ number when starting the development of a new and more accurate model.

$$Nu = 0.348 \times Re^{0.663} \times Pr^{1/3} \times \left(\frac{\mu}{\mu_w}\right)^{0.17} \quad (7)$$

where $\mu_w$ is the dynamic viscosity of the fluid at the wall temperature.

The Equation (7) is valid when referring to the cross-sectional free area.

Lévéque gave another solution for the heat transfer during the laminar flow in the gap between two plates of length $L$ and constant spacing $H$ (Equation (8)), the so-called “1/3 power law”, confirmed by the numerical results of Holzbecher’s simulations [27]:

$$Nu = 0.40377 \times \left(Re \times Pr \times \frac{H}{L}\right)^{1/3} \quad (8)$$

The original Equation (8) was modified for micro-channels with sine ducts, after Martin [21] showed that the heat coefficient depends on pressure drop which is directly proportional to $f_{app} \times Re^2$, where $f_{app}$ is the apparent friction factor (Equation (9)):

$$Nu_{sine} = 0.40377 \times \left(4 \times f_{app} Re_{sine}^2 \times Pr \times \frac{d_{h,sine}}{L_{furr}}\right)^{1/3} \quad (9)$$

Later, Dović and co-authors [22] adjusted this equation, taking into account the influence of factors ignored in the Equation (9) (swirling, real effective flow length), and in accordance with their own experimental data corroborated by other authors’ data [28,29] (Equation (10)):

$$Nu_{sine} = 0.38 \times 0.40377 \times \left(4 \times f_{app} Re_{sine}^2 \times Pr \times \frac{d_{h,sine}}{L_{furr}}\right)^{0.375} \times Pr^{1/3} \times \left(\frac{\mu}{\mu_w}\right)^{0.14} \quad (10)$$

In plate heat exchangers with chevron corrugations, the high heat transfer is due to the flow in small hydraulic diameter sections and interactions between streams flowing and generating swirl secondary flows. Dović and co-authors [22] describe the flow pattern, following an experiment with aqueous glycerol (62% wt.), with $Re$ in range 0.1 to 250.
This experiment allowed us to visualize two flow components: the furrow component (Equation (11)) and the longitudinal component (Equation (12)), and their partial mixing.

\[ L_{furr} = \frac{l}{\sin(2\beta)} \]  
\[ L_{long} = p = \frac{l}{\sin \beta} \]

where \( l \) is the corrugation wavelength.

In the classical correlation (Equation (3)), \( Nu, Re \) and average velocity \( u \ [m/s] \) are related to the total chevron channel cross sectional area transverse to the main flow direction. Taking into account the sinusoidal flow, by means of analytical approach, the average velocity in the cell’s sine duct in furrow direction \( u_{sine} \) is recalculated with Equation (13):

\[ u_{sine} = \frac{m_{ch}}{\rho \times A_{ch, sine}} \]

where \( m_{ch} \) is the mass flowrate in the channel \([kg/s]\), and \( A_{ch, sine} \) is the channel cross-section transverse to the furrow (Equation (14)):

\[ A_{ch, sine} = b \times L_W \times \cos \beta \]

\( L_W \) and \( \beta \) are geometrical characteristics from Table 1. Then, Reynolds sine is calculated with Equation (15).

\[ Re_{sine} = \frac{u_{sine}d_{h, sine}}{v} \]

where \( d_{h, sine} \) is the hydraulic diameter of a sine duct; it is related to the independent variable \( x \)—the ratio corrugation depth: corrugation wavelength \((x = b/l)\), in Equation (16) [30]:

\[ d_{h, sine} = 0.1429x^3 - 0.623x^2 + 1.087x - 0.0014 \]

\( Nu_{sine} \) referred to the cell’s sine duct is correlated with \( Nu \) related to the whole channel through Equation (17):

\[ \frac{Nu}{Nu_{sine}} = \frac{dh}{dh_{sine}} \]

\( L_{furr} \) in Equation (11) is the effective cell length \( L_{cell} \), in case of \( \beta < 60^\circ \). In case of \( \beta > 60^\circ \), \( L_{cell} = L_{long} \), following the prevailing pattern flow; \( f_{app} \) is the apparent friction coefficient which, for a given geometry, has the generalized form (Equation (18)):

\[ f_{app} = \frac{C}{Re_{sine}} + B \]

Constants \( B \) (Equation (19)) and \( C \) (Equation (20)) are functions of channel geometry, and can be calculated as a function of the aspect ratio \((x = b/l)\), by polynomial functions Equations (19)–(23), developed by Dović and co-authors [22] based on the work of Shah [30]:

\[ B = \frac{K_{ood}dh_{sine}}{4 \times L_{furr}} \]

\[ C = 2.6624x^4 - 10.586x^3 + 11.262x^2 - 0.1036x + 9.6 \]

\[ K_e(\infty) = -5.888x^4 + 9.46113x^3 - 4.248x^2 - 0.1333x + 2.648 \]

\[ K_d(\infty) = -1.7237x^4 + 2.7669x^3 - 1.2651x^2 - 0.0097x + 1.512 \]

\[ K(\infty) = 2[K_e(\infty) - K_d(\infty)] \]
where $K(\infty)$ represents the incremental pressure drop number, $K_e$ - kinetic energy correction factor, and $K_d$ is the momentum flux correction factor.

This model (Equations (10)–(23)) is intended to be validated by our own experimental data. The challenge is to find generalized correlations fitting to all flow regimes (laminar, intermediate and turbulent).

3. Results and Discussion

As mentioned before, there were three processing campaigns of vegetable oils. In two of them, the sunflower oils I and II were processed at four different flowrates, and in the third, the rapeseed oil was processed at one flowrate. In total, 72 sets of experimental data resulted. In Table 5, the total mass flowrate refers to the whole flowrate entering the circular port of the heat exchangers, since $Re$ is calculated taking into account the hydraulic diameter of the channel, $d_h$ (see also Table 1). The $Nu$ number is calculated by applying the Kumar correlation (Equation (7)) from [18].

Table 5. $Re$, $Nu$ and $Pr$ criteria for chevron 30° channel related to cross sectional area transverse to the main flow direction.

| Campaign | HE # | Fluids | Total Mass Flow Rate, kg/s | $Re$ | $Nu$ | $Pr$ |
|----------|------|--------|--------------------------|------|------|------|
| Sunflower oil I $\rho (20\,^\circ C) = 919.1\, kg/m^3$ $\mu (20\,^\circ C) = 0.0662\, Pa \times s$ | 1 | RO | 1.74/2.05/2.46/2.71 | 53/62/75/82 | 19.3/21.6/24.3/25.9 | 206.8 |
| | | BO | 2.14/2.52/3.03/3.34 | 27/32/39/43 | 24.5/27.2/30.6/32.9 | 125.1 |
| | | Water | | | | |
| | 2 | RO | 1.74/2.05/2.46/2.71 | 25/29/35/39 | 15.5/17.3/19.5/20.9 | 191.5 |
| | | Water | 5.25/6.20/7.43/8.21 | 1546/1782/2154/2386 | 74.2/85.7/98.1/106.5 | 5.2 |
| | 3 | RO | 1.94/2.19/2.49/2.78 | 30/34/39/43 | 17.7/19.3/20.9/22.4 | 260.6 |
| | | Water | 2.55/3.27/3.62 | 921/1038/1179/1305 | 58.4/63.3/68.9/73.7 | 5.3 |
| | 4 | RO | 1.74/2.05/2.46/2.71 | 18/21/26/28 | 11.7/12.8/14.4/15.3 | 135.8 |
| | | Water | 6.02/7.11/8.52/9.41 | 979/1154/1357/1528 | 63.8/71.1/80.1/85.7 | 5.0 |
| Sunflower oil II $\rho (20\,^\circ C) = 918.1\, kg/m^3$ $\mu (20\,^\circ C) = 0.0650\, Pa \times s$ | 1 | RO | 1.74/2.05/2.46/2.71 | 58/68/81/90 | 20.3/22.7/25.6/27.3 | 170.7 |
| | | BO | 2.14/2.52/3.03/3.34 | 33/39/47/52 | 24.2/27.0/30.5/32.6 | 136.6 |
| | | Water | | | | |
| | 2 | RO | 1.74/2.05/2.46/2.71 | 29/35/42/46 | 16.6/18.6/20.9/22.4 | 160.1 |
| | | Water | 5.25/6.20/7.43/8.21 | 1526/1801/2160/2385 | 74.5/86.0/98.4/107 | 5.2 |
| | 3 | RO | 1.94/2.19/2.49/2.78 | 25/28/32/35 | 19.1/20.7/22.6/24.1 | 213.6 |
| | | Water | 2.55/3.27/3.62 | 919/1103/1379/1504 | 58.4/63.3/68.9/73.7 | 5.3 |
| | 4 | RO | 1.74/2.05/2.46/2.71 | 19/23/27/30 | 11.6/12.9/14.6/15.6 | 144.6 |
| | | Water | 6.02/7.11/8.52/9.41 | 979/1156/1386/1528 | 63.8/71.2/80.3/85.9 | 5.0 |
| Rapeseed oil $\rho (20\,^\circ C) = 919.1\, kg/m^3$ $\mu (20\,^\circ C) = 0.0650\, Pa \times s$ | 1 | RO | 2.72 | 45 | 25.9 | 195.9 |
| | | BO | 3.35 | 72 | 31.2 | 156.2 |
| | | Water | 8.23 | 2391 | 112.6 | 5.2 |
| | 2 | RO | 2.72 | 40 | 21.4 | 184.4 |
| | | Water | 3.62 | 1306 | 73.7 | 5.3 |
| | 3 | RO | 2.72 | 27 | 15.3 | 149.6 |
| | | Water | 9.44 | 1533 | 85.9 | 5.0 |

As seen in Table 5, the experimental data covers a range of laminar flow regime ($Re = 18–90$, $Re < 100$) and another range of fully developed turbulent regime ($Re = 921–2391$, $Re > 250$). $Re$ and $Nu$ numbers, related to the chevron channel cross sectional area transverse to the main flow direction, served later to calculate the experimental values of $Re_{sine}$ and $Nu_{sine}$, related to cell’s sine duct in furrow direction.

Both Equations (9) and (10) consider that heat transfer depends on pressure drop, which is directly proportional to product $fRe^2$. Additionally, $Re_{sine}$, $dh_{sine}$, $s_{app}$, $L_{furrr}$, and constants $B$ and $C$ were determined in each experimental point, for the $Nu_{sine,calc}$ estimation, according to the model of Lévêque adjusted by Dović et al. [22].

The constants $B$ and $C$ were calculated with Equations (18)–(23), for each heat exchanger and the average values were $B = 0.19952 \pm 0.00006$ and $C = 12.4239 \pm 0.0121$. These
values are for the first time calculated in case of gasket plate heat exchangers with corrugation angles $30^\circ$ and the aspect ratio $b/l = 0.8$, since the model to be validated recommends application for $b/l < 0.5$.

The friction factor, $f_{app}$, determined in each experimental point with Equation (17), was plotted vs. $Re_{sine}$ (Figure 2) and compared with the model, on the whole range of $Re_{sine} = 8$–1137. The model fits almost perfectly at $Re_{sine} > 25$ with experimental data of Muley et al. [13] and Heavner [31] but has errors of 5–15% at $Re_{sine} = 9$–25. These errors can be explained by the different geometry (the aspect ratio $b/l > 0.5$ in contrast with $b/l < 0.5$ for all previous experiments), which can affect the friction coefficients at $Re_{sine} < 25$.

![Figure 2](image-url). Variation of apparent friction coefficients ($f_{app}$) with $Re_{sine}$ in the present experiment.

Since $Re$ numbers are between 18 and 2391, $Re_{sine}$ are between 8 and 1137. Moreover, $Nu$ numbers related to the main flow direction, calculated with Kumar correlation are between 6.1 and 112.6, since $Nu_{sine,exp}$ values related to the sine duct flow are between 4.5 and 42.2. These similitude criteria together with $Nu_{sine,calc}$, calculated with model Equations (9)–(23) are shown in Table 6. This table also includes the relative errors between the experimental values of $Nu_{sine,exp}$ (Equation (14)) and those computed with adjusted correlation Lévêque (Equation (10)).

For the whole set of data, the relative errors are between $-18.8\%$ and $+28.5\%$, both positive and negative values, demonstrating that they are not systematical errors. The medium error is $|\bar{\varepsilon}| = 9.56\%$.

The plot $Nu_{sine,calc}$ vs. $Nu_{sine,exp}$ (Figure 3), showing the dispersal of data around the bisector, uncovers that points spreading is minimal at lower values of $Nu_{sine}$, corresponding to the laminar flow, since at values corresponding to the turbulent flow, $Nu_{sine}$ are spread at a bigger distance; however, dispersal on both sides of the bisector for these values too, indicates that the model can be used up to $Re_{sine} = 1137$, with confidence.
Table 6. $Re_{sine}$, $Nu_{sine,exp}$ and $Nu_{sine,calc}$ related to the sine duct flow, for corrugation inclination angle $30^\circ$.

| HE # | Fluids | Total Massflow Rate, kg/s | $Re_{sine}$ | $Nu_{sine,exp}$ | $Nu_{sine,calc}$ | Relative Error, % |
|------|--------|--------------------------|-------------|----------------|----------------|------------------|
| 1    | RO     | 1.74/2.05/2.46/2.71      | 13/15/18/20 | 7.6/8.5/9.6/10.2 | 8.5/9.1/9.9/10.4 | 10.0/6.3/3.2/1.4 |
|      | BO     | 2.14/2.52/3.03/3.34      | 27/31/38/42 | 9.7/10.8/12.1/13.0 | 9.4/10.2/11.2/11.8 | −2.8/−5.3/−7.9/−10.1 |
| 2    | RO     | 1.74/2.05/2.46/2.71      | 11/13/16/18 | 6.1/6.8/7.7/8.2  | 7.1/7.7/8.3/8.7  | 14.4/11.3/7.8/5.5 |
|      | BO     | 1.94/2.19/2.49/2.78      | 11/13/15/15 | 7.0/7.6/7.9/8.3  | 6.7/7.1/7.4/7.4  | −5.0/−8.5/−5.2/−11.5 |
| 3    | Water  | 5.25/6.20/7.43/8.21      | 701/808/977/1082 | 29.3/33.8/38.7/42.0 | 27.5/30.5/35.0/37.7 | −6.3/−10.7/−10.4/−11.3 |
|      | RO     | 2.55/2.85/3.27/3.62      | 586/661/752/854 | 23.1/25.0/27.2/29.1 | 23.9/26.0/28.5/31.2 | 3.2/3.8/4.5/6.8 |
|      | BO     | 1.74/2.05/2.46/2.71      | 9/10/12/14   | 4.6/5.3/5.7/6.0  | 5.6/6.0/6.5/6.8  | 17.4/15.7/12.4/10.9 |
|      | Water  | 6.02/7.11/8.52/9.41      | 446/525/628/696 | 24.0/27.0/30.5/32.6 | 20.3/22.8/25.9/27.9 | −18.2/−18.1/−17.8/−17.1 |
| 4    | RO     | 1.74/2.05/2.46/2.71      | 15/18/21/24  | 7.6/8.5/10.1/10.8 | 8.4/9.1/9.9/10.4 | 4.7/1.0/−2.3/−4.2 |
|      | BO     | 2.14/2.52/3.03/3.34      | 24/28/34/38  | 9.6/10.7/12.0/12.9 | 9.3/10.0/11.0/11.6 | −3.3/−6.5/−9.8/−11.5 |
|      | Water  | 5.25/6.20/7.43/8.21      | 694/820/981/1085 | 29.6/33.9/38.3/42.2 | 27.2/30.7/34.9/37.0 | −8.0/−10.6/−11.0/−13.3 |
| 5    | RO     | 1.74/2.05/2.46/2.71      | 13/16/19/21  | 6.5/7.3/8.2/8.8  | 7.2/7.8/8.5/8.9  | 9.3/5.9/2.6/0.3  |
|      | BO     | 1.94/2.19/2.49/2.78      | 14/16/18/20  | 7.5/8.2/8.9/9.5  | 8.2/8.6/9.2/9.6  | 8.0/5.4/2.6/0.9 |
|      | Water  | 2.55/2.85/3.27/3.62      | 586/661/752/854 | 23.1/25.0/27.2/29.1 | 23.9/26.0/28.5/31.0 | 2.6/3.2/4.0/6.3 |
|      | RO     | 1.74/2.05/2.46/2.71      | 8/10/12/13   | 4.6/5.3/5.8/6.2  | 6.4/6.9/7.4/7.8  | 28.5/25.9/22.5/20.6 |
|      | BO     | 1.94/2.19/2.49/2.78      | 11/13/15/15  | 6.1/6.8/7.7/8.2  | 7.1/7.7/8.3/8.7  | 14.4/11.3/7.8/5.5 |
|      | Water  | 6.02/7.11/8.52/9.41      | 446/525/628/696 | 24.0/26.9/30.5/32.6 | 20.3/22.7/22.9/31.0 | −18.8/−18.3/−17.8/−17.5 |

Figure 3. Comparison between the experimental $Nu_{sine}$ values and those computed with the model (Equation (10)).

For the forced convection in stationary regime, it is customary to look for correlations between $Re$ and $Nu/Pt^{1/3}$ $(\mu/w)^{0.14}$. We used the facility Data analysis in Excel. The computed results for the similitude criteria taking into account the sine duct flow are shown in Table 7.
As seen in Table 7, both regression statistics and ANOVA analysis show that $Re_{sine}$ and $Nu_{sine}/Pr^{1/3} \left(\mu/\mu_w\right)^{0.14}$ are strongly correlated through linearity. The coefficient of determination is close to 1 ($R^2 = 0.9849$). The ANOVA variance test demonstrated that null hypothesis is false, so the mean squares for treatments MST is much larger than the mean squares for errors MSE (3770.1214 $\gg$ 0.8269); the MST/MSE ratio is $F = 4559.34$, a very high value, far from $F = 1$, the value for non-correlated series of data. In conclusion, the Fisher test confirms the strong correlation between the series of data. The linear correlation (Equation (24)):

$$Nu_{sine}/Pr \times \left(\mu/\mu_w\right)^{0.14} = 1.3305 + 0.0195 Re_{sine} \tag{24}$$

The $p$-values $< 0.05$ for the intercept and the slope of the line demonstrate that the model works with the confidence in the interval $\pm 95\%$. $Nu_{sine}$ includes the partial heat transfer coefficient $\alpha$ in the fluid on one side of the plate; the coefficient on the other side of the plate can be calculated with the corresponding value of $Nu_{sine}$. Taking also into account the conductivity of the plate material, the overall heat transfer coefficient can be found. It is very important to have an accurate model for the estimation of $Nu_{sine}$, because the errors spread in the values of the overall coefficient.

### 4. Conclusions

In this study, the semi-analytical model of Lévéque for heat transfer in gasket plate heat exchanger channels was validated in our experimental data obtained from industrial equipment for vegetable oil processing. The vegetable oils had different physical properties proceeding from three different raw materials in different stages of the process (raw, bleached, winterized). There were four exchangers with similar geometry but different in size. The flow regime was laminar ($Re < 100$) for oils and turbulent ($Re > 250$) for water, allowing us to check the model in a large range of $Re$ numbers, related to the cross-sectional free area of the channel.

The model of Lévéque takes into consideration the flow in cell’s sine duct in furrow direction. Consequently, the similitude criteria $Re_{sine}$ and $Nu_{sine}$ are calculated in new considered conditions. They are correlated with a more complex relationship, considering the construction details of the channels. In this work, the constants $C$ and $D$ of the model were determined, for corrugation inclination angle relative to vertical direction equal to $30^\circ$, for the first time. Thus, $B = 0.19952 \pm 0.00006$ and $C = 12.4239 \pm 0.0121$. As regards the constant $C_1$ of the model, that is the same as determined by Dović et al. [22] when adapting Lévéque correlation to the corrugation inclination angle $60^\circ$ ($C_1 = 0.1534$).
Both the analysis of relative errors and the statistical analysis concluded that Lévêque correlation adapted by Dović et al., with model parameters, determined here for corrugation angle 30°, can be used with confidence for predicting Nu criterium as a basis for heat transfer calculations in gasket plate exchangers.

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