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Thermal performance analysis of triple heat exchangers via the application of an innovative, simplified methodology

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ABSTRACT

Despite the double tube heat exchangers, in the triple tube heat exchangers, there are three fluids, and the methodology based on the assessment of the logarithmic mean temperature difference is no longer applicable. Moreover, in triple tube heat exchangers, there are two overall heat transfer coefficients dependent on each other. As such, it is necessary to solve them simultaneously, thus making the evaluation of the thermal performance of triple tube heat exchangers more complex compared to double tube heat exchangers. Among the proposed approaches in the literature to solve this issue, one of the most powerful and commonly adopted in several engineering applications is the parameter estimation procedure. Nevertheless, for the specific implementation examined in our analysis, a thorough numerical model of the triple tube heat exchanger was required to apply the inverse procedure properly. Furthermore, it is mandatory to measure the temperature of the three fluids at the inlet and outlet sections. In so doing, the inverse procedure can be successfully applied to the characterisation of triple tube heat exchangers tested in well-equipped research labs; however, its application to heat transfer devices operating in industrial facilities can be difficult. In order to overcome this limitation, an innovative parameter estimation technique that enables the evaluation of the thermal performance of this type of heat transfer devices is presented. The suggested methodology is based on a simple model of the triple tube heat exchanger in which an equivalent double tube heat exchanger is considered, thus requiring only four temperature measurements. The results obtained by applying this simplified methodology are numerically validated and compared to those obtained using a comprehensive model.

1. INTRODUCTION

The role of heat exchangers is crucial in several thermal processes (e.g. food pasteurisation and sterilisation, cooling electronic devices and refrigeration). Although many experimental and numerical investigations of heat exchangers have been carried out, improving the performance of these heat transfer devices represents one of the major technological challenges due to the rising cost of energy and raw materials. Among the heat transfer enhancement techniques that can be adopted to improve the performance of heat exchangers, increasing the heat transfer area is one the most used [1-3]. Triple concentric-tube heat exchangers (TTHEs) represent an improvement of double tube heat exchangers (DTHEs) since they are characterised by a supplementary section that increases the heat transfer and provides a bigger heat transfer surface per unit length.

Although TTHEs are widely used in the food and pharmaceutical industries, and they present many advantages, in the literature, few studies focusing on the heat transfer phenomena analysis of these kinds of devices can be found [4]. It must be pointed out that since in a TTHE, there are two overall heat transfer coefficients (at both sides of the annulus) that are dependent on each other, it is necessary to solve them simultaneously. Thus, the computation of overall heat transfer coefficients in a TTHE is more complex compared with double tube heat exchangers.

Closed-form expressions for the effectiveness-NTU relations for both the counter-flow and parallel-flow configurations were proposed by Ünal et al. [5,6]. Additionally, a computational algorithm that enables the evaluation of the overall heat transfer coefficients and axial distributions of the temperature in a TTHE was presented by Batmaz and Sandeep [7] and Radulescu et al. [8].

A new definition of the logarithmic mean temperature difference was introduced in [9] and [10], the so-called average log-mean temperature difference, which was evaluated as the average value of the two log-mean temperature differences (i.e. the logarithmic mean temperature difference between the inner and outer annular sections and the logarithmic mean temperature difference between inner annular section and inner tube).

Recently, an Artificial Neural Network (ANN) model for assessing the thermal behaviour of a TTHE was developed by Moya-Rico [11]. Nevertheless, due to the specificity of each thermal process, for instance, in terms of treated product and geometry, it is often difficult to apply the proposed heat transfer correlations to other processes.

More recently, Malavasi et al. [12] proposed an innovative approach to characterise TTHEs, which was validated using both synthetic and experimental data. The proposed methodology was based on an inverse technique (i.e. the parameter estimation procedure), which represents a powerful tool for designing and optimising heat transfer devices that are customised for specific purposes, as often occurs with TTHEs [13-17]. This novel procedure enables the estimation of the heat transfer correlation for the product-side Nusselt number, thus allowing the limitations of the Wilson plot technique to be overcome [18].

Nevertheless, to properly characterise TTHEs by applying this innovative methodology, a comprehensive numerical model of the investigated heat exchanger is necessary. Moreover, this inverse procedure requires six temperature measurements (i.e. the inlet and outlet temperatures for the three fluids flowing through the TTHE).

It must be highlighted that in most of TTHEs, the service fluids flowing in the inner tube and outer annular section are the same. Accordingly, only two temperatures for the service side (i.e. one at the inlet and one at the outlet) can be easily measured, possibly leaving the others unknown.

Although the inverse procedure proposed in [12] can be successfully applied to the TTHE characterisation of heat exchangers tested in well-equipped research labs, its application to heat exchangers operated in industrial facilities can be difficult.

In order to overcome this limitation, a new parameter estimation method that allows the evaluation of the performance of TTHEs was described by Vocale et al. [16]. The proposed methodology, which was validated using synthetic data, was based on a simplified TTHE model in which an equivalent DTHE was introduced. For this novel approach, only four temperature measurements were required. The present work is an extension of the study carried out in [19].

2. HEAT EXCHANGER MODEL

The triple concentric-tube heat exchanger investigated here operates in a counter-flow configuration that is the most used arrangement in industrial problems because it guarantees the higher efficiency. The product flows into section 2, and the service fluids flow into sections 1 and 3, as shown in Figure 1.

Therefore, the simplified model of the TTHE, i.e. the equivalent Double Tube Heat Exchanger (DTHE), operates in counter-flow configuration, as outlined in Figure 2: the subscripts s and p indicate the service fluid and the product, respectively.

The geometric characteristics of the equivalent DTHE are calculated as follows: the diameter of the service side is the average between the hydraulic diameters of the two service sections of the TTHE (i.e. sections 1 and 3 in Figure 1); the diameter of the product side has the same value as the hydraulic diameter of the inner annulus of the TTHE (i.e. section 2 in Figure 1); the heat transfer areas are the same.

For any double tube heat exchanger in a steady-state condition, in which the heat loss to the surroundings and the heat conduction in the flow direction are negligible, it is possible to evaluate the average overall heat transfer coefficient U for the inner heat transfer surface area A_i as follows:

$$U = \frac{Q}{A_i \Delta T_{ml}} \tag{1}$$

being ΔT_{ml} the logarithmic mean temperature difference and Q the heat transfer rate.

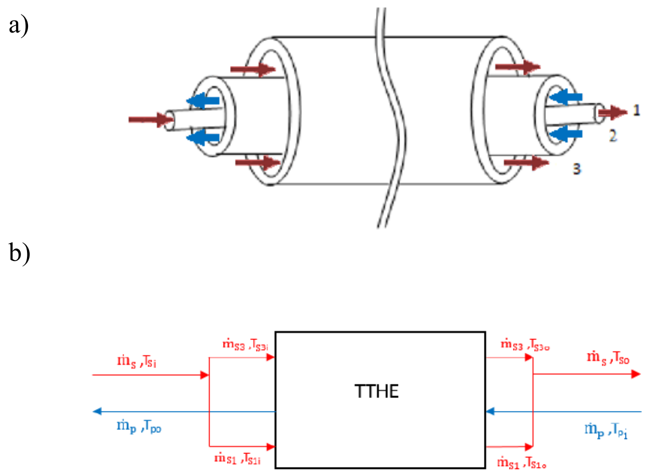

Figure 1. TTHE under investigation. a) 3D representation; b) system schema.

Both the fluids are single-phase and presented constant thermal conductivity k, density ρ and specific heat c_p. The temperatures at the inlet of both the tube and shell sides are assumed to be known as well as the two mass flow rates and the exchanged heat flux. Therefore, it is possible to write:

$$Q = \dot{m}_p\, c_{pp}\, (T_{p,out} - T_{p,in}) \tag{2}$$

$$Q = \dot{m}_s\, c_{ps}\, (T_{s,in} - T_{s,out}) \tag{3}$$

where $\dot{m}$ and Q are the mass flow rate and the exchanged power, respectively. The subscripts *in* and *out* refer to the inlet and outlet conditions.

The overall heat transfer coefficient can be evaluated by considering the product and service fluid convective heat transfer coefficients as follows:

$$\frac{1}{UA_i} = \frac{1}{h_p A_i} + R_{w,eq} + \frac{1}{h_s A_e} \tag{4}$$

where h_p and h_s are the product and service fluid convective heat transfer coefficients, respectively, A_e the external heat exchanger surface area and $R_{w,eq}$ the wall thermal resistance.

Figure 2. System schema of the equivalent DTHE.

To evaluate the wall thermal resistance for the equivalent DTHE, it has to be considered that in a TTHE, there are two thermal resistances (i.e. one due to the wall between sections 1 and 2 and one due to the wall between sections 2 and 3) in a parallel arrangement. Thus, the wall thermal resistance for the equivalent DTHE can be estimated as follows:

$$R_{w,eq} = \frac{R_{w12} \cdot R_{w23}}{R_{w12}+R_{w23}} \tag{5}$$

where the wall thermal resistance for each wall, which depends on the internal and external diameter of the pipe, on the wall thermal conductivity and on the pipe length, can be approximated as follows [20]:

$$R_w = \frac{ln\left({D_o}/{D_i}\right)}{2\pi\lambda_{wL}} \tag{6}$$

Assuming that the hydraulic diameter for both the product and service sides is known, as well as the fluid thermal conductivity, it is possible to define the Nusselt numbers:

$$Nu_p = \frac{h_p\, D_{hp}}{\lambda_p} \tag{7}$$

$$Nu_s = \frac{h_s\, D_{hs}}{\lambda_s} \tag{8}$$

For fully developed flows, the Nusselt numbers may be obtained from [20]:

$$Nu_p = C_p Re_p^{\alpha_p} Pr_p^{\beta_p} \tag{9}$$

$$Nu_s = C_s Re_s^{\alpha_s} Pr_s^{\beta_s} \tag{10}$$

being *Re* and *Pr* the Reynolds and the Prandtl numbers, respectively.

Therefore, the overall heat transfer coefficient *U* can be obtained from:

$$U = \frac{1}{A_i}\left(\frac{D_{hp}}{A_i\lambda_p C_p Re_p^{\alpha_p} Pr_p^{\beta_p}} + R_w + \frac{D_{hs}}{A_e\lambda_s C_s Re_s^{\alpha_s} Pr_s^{\beta_s}}\right)^{-1} \tag{11}$$

The log mean temperature difference, ΔT_{ml} (Eq. 12) is adopted to determine the product and service fluids outlet temperatures [20].

$$\Delta T_{ml} = \frac{\Delta T_2 - \Delta T_1}{ln\left({\Delta T_2}/{\Delta T_1}\right)} \tag{12}$$

where ΔT_1 and ΔT_2 are obtained from:

$$\Delta T_1 = T_{s,in} - T_{p,out}, \qquad \Delta T_2 = T_{s,out} - T_{p,in} \tag{13}$$

Replacing Eqs. (2,3,13) into Eq. (1), the exchanged heat flow rate can be expressed as:

$$Q = (B-1)\left[\frac{T_{si} - T_{pi}}{\left(\frac{B}{\dot{m}_p\, c_{pp}} - \frac{1}{\dot{m}_s\, c_{ps}}\right)}\right] \tag{14}$$

with *B* equal to [14]:

$$B = exp\left[UA_i \frac{(\dot{m}_s\, c_{ps} - \dot{m}_p\, c_{pp})}{\dot{m}_s\, c_{ps} \cdot \dot{m}_p\, c_{pp}}\right] \tag{15}$$

The outlet temperatures, $T_{p,out}$ and $T_{s,out}$, are obtained from:

$$T_{p,out} = T_{p,in} + \frac{Q}{\dot{m}_p\, c_{pp}} \tag{16}$$

$$T_{s,out} = T_{s,in} - \frac{Q}{\dot{m}_s\, c_{ps}} \tag{17}$$

Equations (16) and (17) represent the solution to the direct problem, which allows the evaluation of the outlet temperatures for both the product and service fluids when the coefficients *C*, α, and β are known.

3. INVERSE PROBLEM

Usually, the coefficients *C*, α, and β appearing in Eqs. (9) and (10) are unknown and since they cannot be directly measured, their estimation is required. The inverse problem enables, indeed, the estimation of the coefficients *C*, α, and β for the product and service sides, knowing four temperature measurements. In particular, the predicted outlet temperatures, computed by Eqs. (16) and (17) (i.e. direct problem solution) are constrained to fit the experimental temperatures by changing *C*, α, β.

Therefore, by adopting a least-square method, the objective function reads as follows:

$$S(\boldsymbol{P}) = \sum_{j=1}^{N}\left[\boldsymbol{T}_{exp,j} - \boldsymbol{T}_{pred,j}\right]^2 \tag{18}$$

being $\boldsymbol{P}$ the vector of the parameters that must be assessed, $\boldsymbol{T}_{exp}$ the vector of the experimental temperatures, $\boldsymbol{T}_{pred}$ the vector of the estimated temperatures, and *N* the number of experimental values.

$\boldsymbol{T}_{exp}$ consists of the product and service sides outlet temperatures obtained in the *N* measurements; $\boldsymbol{T}_{pred}$ contained the solutions of the direct problem [12].

In a general industrial application, different fluids can flow in a DTHE; therefore, for the equivalent DTHE, a six-parameter optimisation procedure is required to estimate the coefficients *C*, α and β for both the tube and shell sides.

Since the problem is non-linear with respect to the unknown variables, a non-linear optimisation algorithm is required. Among the techniques that could be used for parameter estimation, an algorithm commonly used is the non-linear fit based on the Iterative Reweighted Least Squares method [17].

The reliability of the parameter assessments can be evaluated by considering the 95% confidence interval, $CI^{95\%}$, and the coefficient of variation, *CV*, which are defined as follows [21]:

$$CI_{P_i}^{95\%} = (P_i - 1.96\sigma_{Pi}, P_i + 1.96\sigma_{Pi}) \tag{19}$$

$$CV_{Pi} = \frac{\sigma_{Pi}}{P_i} \tag{20}$$

4. RESULTS AND DISCUSSION

The new proposed methodology was tested within the Matlab® environment by adopting synthetic data. The geometric features of the TTHE and the equivalent DTHE are presented in Tables 1 and 2, respectively.

Synthetic data were obtained by solving the full direct problem presented in [12] and by considering a highly viscous fluid food (i.e. fruit purees or concentrated juices) as product flowing into section 2 (Figure 1) and water as service fluid flowing in both sections 1 and 3.

Table 1. TTHE geometrical features

Parameter	Value
D_{h1} (m)	0.0409
D_{h2} (m)	0.0186
D_{h3} (m)	0.0108
L (m)	10.10
A_{i1} (m^2)	1.2990
A_{i2} (m^2)	2.1237
A_{e1} (m^2)	1.5326
A_{e2} (m^2)	2.3172

Table 2. DTHE geometrical features

Parameter	Value
D_{hs} (m)	0.0259
D_{hp} (m)	0.0186
L (m)	10.10
A_i (m^2)	3.4227
A_e (m^2)	3.8498

The fluids properties were: ρ_s= 1'000 kg·m^{-3}, λ_s= 0.6 W·m^{-1}·K^{-1}, μ_P= 1·10^{-3} Pa·s, c_{ps}= 4'180 J kg^{-1}K^{-1}, ρ_p = 1'054 kg·m^{-3}, μ_P= 2.6·10-1 Pa·s, λ_p = 5.9 10-1 W·m^{-1}·K^{-1} and c_{ps}= 3'852 J kg^{-1}K^{-1}. A turbulent flow regime was supposed for the service fluid (18·10^3 < Re_s < 65·10^3), whereas a laminar flow regime (5 < Re_p < 500) was assumed for the product. In these, Reynolds ranges 225 operating conditions were considered.

To account for the experimental noise, the synthetic outlet temperatures were deliberately spoiled by using random noise [9] and then used as input data for the inverse procedure based on the simplified model presented.

Since the working fluid was supposed to be the same under all operating conditions, there was no sensibility concerning the effect of the Prandtl number on the heat transfer mechanism, and consequently, the estimations of β_p and β_s were unfeasible using the inverse problem methodology. As such, only C_p, α_p, C_s and α_s were assessed, while β_p and β_s were assumed as established.

For the assessed quantities, the 95% confidence intervals and coefficients of variation for noise level equal to 0.05 K (i.e. a typical noise level for the specific application and experimental setup here evaluated) are presented in Table 3.

Table 3. Results of the simplified model

Unknow parameter	Estimated parameter	$CI^{95\%}$		CV
C_p	0.025	0.024	0.025	0.60%
α_p	0.807	0.804	0.811	0.24%
C_s	0.006	0.005	0.006	4.71%
α_s	0.788	0.779	0.798	0.60%

The numerical outcomes revealed that the *CI* were very narrow, thus highlighting that the novel estimation procedure was able to estimate the unknown coefficients with good accuracy. Moreover, the coefficients of variation were very small: the highest *CV* value was 4.7%, thus verifying the effectiveness of the simplified model for the specific case here considered.

The restored outlet temperature values of the product fluid for 0.05 K of noise level versus the correct ones are shown in Figure 3.

The comparison confirms that the simplified model enables the estimation of the product fluid's outlet temperature with good accuracy.

With the aim of providing a better insight into the efficacy assessment of the novel model, a residual analysis was carried out by defining the average estimation error between the restored and exact values of the heat transfer rate:

$$E_Q = \frac{\|Q_{restored} - Q_{exact}\|_2}{\|Q_{exact}\|_2} \tag{21}$$

The exact values of the heat transfer rate were evaluated by applying the full model presented in [12], whereas the restored values were computed by adopting the novel parameter estimation procedure proposed here.

The average estimation error of the heat transfer rate, evaluated via Eq. (21), was lower than 1%.

The comparison between the estimated and exact restored values of the heat transfer rate is presented in Figure 4, together with the *CI* bands. These intervals were estimated by computing the E_Q standard deviation:

$$e_Q = \frac{Q_{restored} - Q_{exact}}{Q_{exact}} \tag{22}$$

A good agreement between the exact heat transfer rate and the values obtained by considering the outlet temperatures computed using the simplified model (Eqs.(16) and (17)) was determined, as shown in Figure 4.

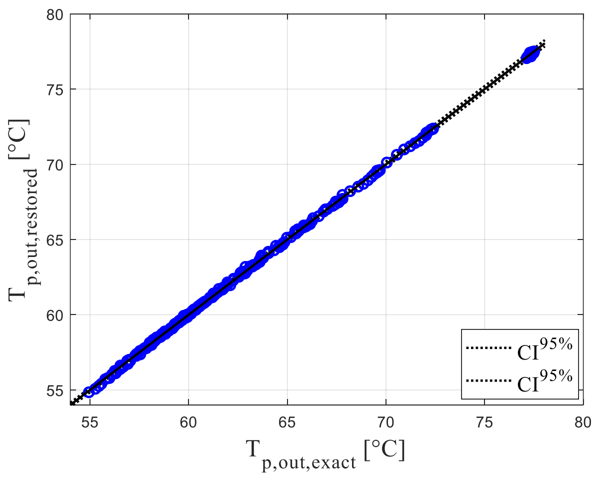

Figure 3. Correlation between the estimated and exact outlet temperatures of the product fluid

It can be noticed that the diversity between the estimated and correct heat transfer rate varied with the product *Re* to the service *Re* ratio. Particularly, the maximum variation could be observed in correspondence of the lowest Re_s whereas the smallest disparity was detected for the greatest Re_s.

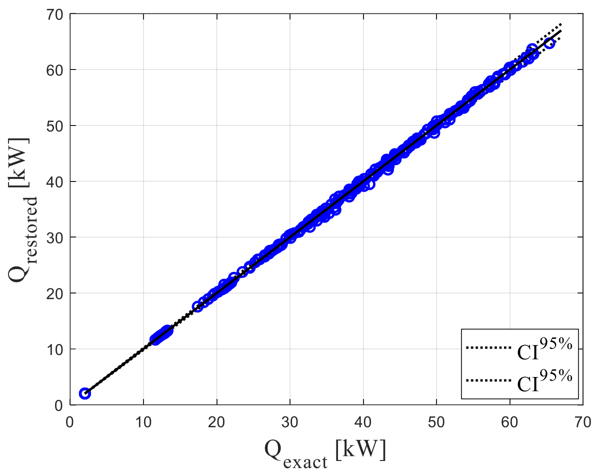

Figure 4. Correlation between the estimated and exact heat transfer rates

These outcomes are corroborated by Figures 5 and 6, in which the estimated Q for a noise level equal to 0.05 K is shown against the correct ones for a single value of Re_s (i.e. the highest and the lowest here studied).

The influence of the product Re to the service fluid Re ratio on the estimation of the heat transfer rate was also investigated in terms of estimation error. To this aim, Eq. (21) was applied to each value of Re_s analysed in the current study.

The results, presented in Table 4, reveal that the estimation error value of the heat transfer rate was dependent on the value of the Reynolds number of the service fluid; notably, the maximum estimation error value was obtained for the smallest value of Re_s deemed in this investigation. However, the maximum error was about 2%, thus confirming the accuracy of the novel estimation procedure.

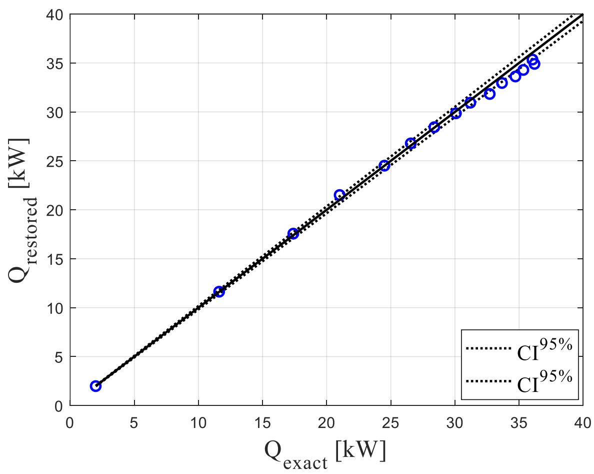

Figure 5. Correlation between the estimated and exact heat transfer rates for Re_s= 18’370

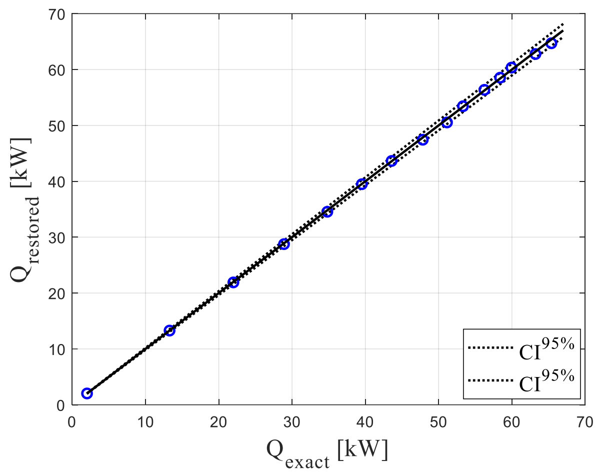

Figure 6. Correlation between the estimated and exact heat transfer rates for Re_s= 64’293

In order to assess the practical applicability of the novel estimation procedure, reduced datasets in the same ranges of Re_p and Re_s were investigated. Practically, it can be very difficult to carry out too numerous experiments; as such, 49 and 25 operating conditions were considered.

Table 4. Estimation error of the heat transfer rate for different values of the Reynolds number of the service fluid

Re_s	E_Q
18’370	2.09%
22’043	1.43%
25’717	1.26%
29’391	0.71%
33’065	0.91%
36’739	0.77%
40’413	0.63%
44’087	0.51%
47’761	0.58%
51’435	0.42%
55’109	0.25%
58’783	0.40%
60’619	0.33%
62’456	0.60%
64’293	0.54%

As expected, by decreasing the number of measurements, the outcomes obtained by applying the novel parameter estimation procedure become less accurate, as shown in Tables 5 and 6, in which the estimated parameters for the reduced datasets are reported, including a noise level equal to 0.05 K.

Table 5. Estimation results of the simplified model (49 operating conditions)

Unknow parameter	Estimated parameter	IC		CV
C_p	0.025	0.024	0.025	0.96%
α_p	0.804	0.797	0.810	0.44%
C_s	0.006	0.005	0.007	10.03%
α_s	0.785	0.765	0.805	1.30%

Table 6. Estimation results of the simplified model (25 operating conditions)

Unknow parameter	Estimated parameter	IC		CV
C_p	0.025	0.024	0.025	1.10%
α_p	0.803	0.794	0.811	0.55%
C_s	0.006	0.005	0.008	12.53%
α_s	0.785	0.760	0.810	1.64%

Based on the results presented in Tables 5 and 6, it can be deduced that the *CI*s were wider and the *CV*s higher than those of the complete dataset. Notably, the highest values of *CV* were approximately 10% and 13% for the 49 and 25 operating conditions, respectively.

However, a good agreement between the values of the exact and restored heat transfer rates was found even for the reduced datasets, as shown in Figures 7–8 and Tables 7–8. Notably, the maximum estimation error of the heat transfer rate was less than 1% (even for the reduced datasets), thus confirming that the heat transfer rate could be accurately evaluated even with a limited number of measurements.

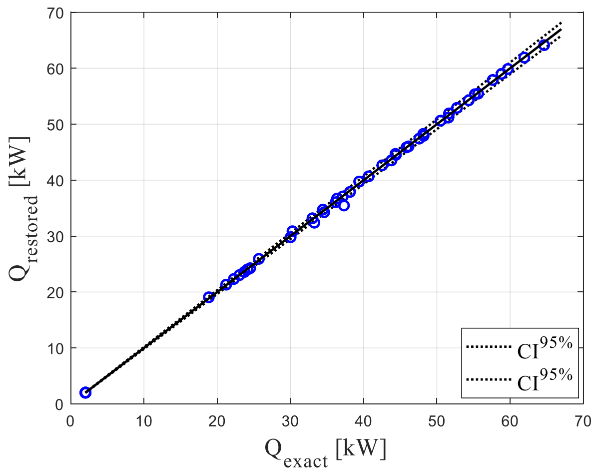

Figure 7. Correlation between the estimated and exact heat transfer rates (49 operating conditions)

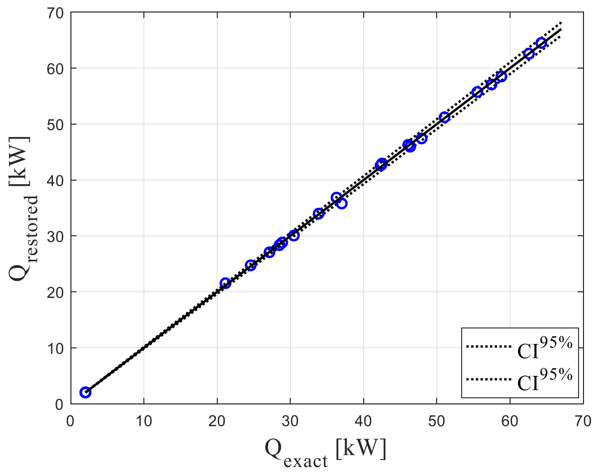

Figure 8. Correlation between the estimated and exact heat transfer rates (25 operating conditions)

Table 7. Estimation error of the exchanged heat transfer rate (49 operating conditions)

Re_s	E_Q
18’370	2.81%
29’391	0.80%
36’739	0.67%
44’087	0.49%
51’435	0.31%
58’783	0.31%
64’293	0.54%

Table 8. Estimation error of the exchanged heat transfer rate (25 operating conditions)

Re_s	E_Q
18’370	2.17%
29’391	1.08%
44’087	0.43%
58’783	0.46%
64’293	0.61%

The numerical outcomes for the reduced datasets reveal that the novel estimation procedure enabled the estimation of the heat transfer rate with good accuracy; however, the heat exchanger model was simpler than the full model and required a limited number of fluid temperatures.

5. CONCLUSIONS

In the present work, a novel parameter estimation procedure for evaluating the thermal performance of triple concentric-tube heat exchangers is proposed. In order to reduce the complexity of the full model available in the literature, our methodology presents a simple model for the triple tube heat exchanger, based on an equivalent double tube heat exchanger.

The validation of this new procedure highlights the simplified model’s capability of estimating the heat transfer rate with good accuracy, even when the number of the measurements is limited. Accordingly, the proposed simplified model can be successfully applied to heat exchangers operating in industrial facilities.

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NOMENCLATURE

A	Heat transfer surface area, m^2
C	Multiplicative constant (Eq. (9))
$CI^{95\%}$	Confidence interval
CV	Coefficient of variation
c_p	Specific heat, J kg^{-1} K^{-1}
D	Diameter, m
h	Convective heat transfer coefficient, W m^{-2} K^{-1}
L	Heat exchanger's length, m
$\dot{m}$	Mass flowrate, kg s^{-1}
Nu	Nusselt number
P_i	Generic unknown parameter
Pr	Prandtl number
Q	Heat transfer rate, W
Re	Reynolds number
R_w	Wall thermal resistance, W K^{-1}
$R_{w,eq}$	Equivalent wall thermal resistance, W K^{-1}
S	Target function
T	Temperature, K
U	Overall heat transfer coefficient, W m^{-2} K^{-1}

Greek symbols

α	Reynolds number exponent (Eq. (9))
λ	Fluid thermal conductivity, W m^{-1} K^{-1}
λ_w	Wall thermal conductivity, W m^{-1} K^{-1}
σ	Standard deviation

Subscripts

in	Inlet section
out	Outlet section
p	Product
s	Service