Investigation of the Pressure Drop in the Shell Side of the Evaporator

Róbert Sánta[1], Igor Fürstner[2]

[1] University of Dunaújváros, Department of Mechanical Engineering and Material Sciences, Táncsics Mihály 1/a, 2400 Dunaújváros, Hungary
e-mail: santar@uniduna.hu

[2] Óbuda University, Bánki Donát Faculty of Mechanical and Safety Engineering,
e-mail: furstner.igor@bgk.uni-obuda

Abstract: This work focuses on the pressure drop of the cooled fluid in the shell side of the heat pump's evaporator since the structures and the results obtained for the process characteristics of the models available in the literature show a significant standard deviation. Hence, the mathematical modeling and description of the heat pump is rather uncertain due to the inaccuracy of the functions describing the pressure drop. This research paper presents a new formula developed on the basis of the measurements, which can be used to calculate the pressure drop in the shell side of the heat pump evaporator with greater accuracy compared to the correlations found in the literature. The maximum discrepancy – from measurement values – of the values yielded by the pressure drop model with the new proposed correlation as set up by the author is ε_{max}=5.72%, while the average discrepancy is only ε=2.58%. The presented new correlation was determined under the measuring conditions of 1-15 (m^3/h), Re = 478-7175 (-) and 13-15 (°C).

Keywords: heat pump; evaporator; shell side; pressure drop

1 Introduction

In addition to the compressor, the author also used pumps to operate the heat pump system: in the heat sinks, which are cooled systems connected to the evaporator and in the heat supply, the heating system connected to the condenser, the heat transfer fluids are circulated by pumps. An increase in the pressure drop in the cooled fluid, i.e., water, directly affects the economy of the heat pump circuit, as the higher energy consumption required to operate the pump reduces the economy of the entire heat pump system.

Thus, the pressure drop in the evaporator causes thermodynamic losses, which also degrade the cooling capacity and the COP value of the equipment. In the calculation of the COP of a compressor heat pump system, the performance of the circulating pumps must also be taken into account in the context of the real cycle:

$$COP = \left|\frac{Qo+W_{com}+W_{HC}}{W_{com}+W_{HC}+W_{CC}}\right| \rightarrow max! \tag{1}$$

By describing the stationary operation of the heat pump system with the algebraic equations [1] or describing the steady state operation of the heat pump system with a coupled system of differential equations [2] the problem lies is the auxiliary equations, because the results of the heat transfer and the pressure drop correlations show very large deviations, and the applicability and their validity and limitations are not always clear. Consequently, the results of the mathematical models describing the operation of the heat pump are highly uncertain and inaccurate.

The reason that it is challenging to predict the shell-side heat transfer coefficient and pressure drop with complete accuracy is the complex flow pattern on the shell-side, and the great number of variables involved. Before 1960 no one tried to account for the leakage and bypass streams through methods used for the design of exchangers. The basis of the correlations was the total stream flow, and empirical methods were implemented so as to explain the performance of real exchangers in comparison with that for cross flow over ideal tube banks. Kern [3] and Donohue [4] provided typical “bulk-flow” methods.

The first publication of a detailed stream-analysis method for predicting shell-side heat-transfer coefficients and pressure drop was connected to Tinker, his model becoming the basis of all subsequently developed methods [5]. Numerous new models were created later, for example that of Gnielinski and Gaddis [6]. The following procedure was used for the evaluation of the shell side pressure drop in shell-and-tube heat exchangers with segmental baffles. The basis of this procedure was the correlations for calculating the pressure drop in an ideal tube bank coupled with correction factors. These also considered the influence of leakage and bypass streams. Further, the basis were also the equations for calculating the pressure drop in a window section from the Delaware method [7]. The offered equations were examined: a comparison was performed of experimental measurements available in the literature with the theoretical predictions. The paper outlined the ranges of the geometrical and operational parameters, for which the deviations between the experimental measurements and the theoretical predictions were within ± 35%. Kottke [8] researched the local heat transfer and pressure drop on the shell side of shell-and-tube heat exchangers with segmental baffles, with a focus on different baffle spacings. Mass transfer measurements was used to determine and visualize the distributions of the local heat transfer coefficients on each tube surface within a fully developed baffle compartment were determined and visualized. The local values helped define the per-tube, per-row, and per-compartment average heat transfer coefficients. The local pressure measurements made it possible to determine the shell-side flow distributions. For the same Reynolds number, the pressure drop and average heat transfer are enhanced by a greater baffle spacing, which can be explained by a decreased leakage through the baffle-shell clearance. The authors compared the obtained results with values found in the relevant literature. Davide [9] both the pressure drop across a heat exchanger as well as the heat transfer capacity are vital important parameters. Indeed, the pressure losses define the actual the operating cost throughout the exchanger life cycle. Hence, predicting the pressure drop with a great degree of precision is just as crucial as predicting the heat transfer. A new data set of shell-side pressure drop measurements taken during isothermal flow of brines in shell and tube evaporators was collected in the Alfa Laval laboratory. Several different configurations of industrial shell and tube evaporators and a wide range of operating conditions, with the crossflow Reynolds number ranging from 170 to 33.000 are covered. Two predictive procedures available in the literature for computing shell-side pressure drop were used as a point of comparison for this database, demonstrating that there was no method which would be sufficiently precise for design purpose. This called for a new procedure, as an extension of the Wills and Johnston model, a novel and exact hand calculation method for shell side pressure drop and flow distribution. Uday [10] created a theoretical model for shell-side pressure drop. The model integrated the eff ect of pressure drop in both the inlet and outlet nozzles, as well as the losses in the segments created by baffles. The results of the model for Reynolds numbers ranging from 10^3 to 10^5 showed a greater correspondence with the experimental results available in the literature as compared to analytical models developed by other researchers for various configurations of heat exchangers. Hosseini [11] The authors of this work experimentally determined the heat transfer coefficient and pressure drop on the shell side of a shell-and-tube heat exchanger for three different types of copper tubes (smooth, corrugated and with micro-fins). Further, they performed a comparison of the experimental data with available theoretical data. Correlations were proposed for the pressure drop as well as the Nusselt number for the three tube types. They modelled and built a shell-and-tube heat exchanger of an oil cooler used in a power transformer for this experimental work so they could examine the effect of surface configuration on the shell side heat transfer along with the pressure drop of the three types of tube bundles. The bundles with identical geometry, configuration, number of baffles and length, yet with different external tube surfaces inside the same shell were implemented in the experiment. Corrugated and micro-fin tubes demonstrated degradation of performance at a Reynolds number below a certain value ($Re < 400$). At a greater Reynolds number, the performance of the heat exchanger significantly improved for micro-finned tubes.

The following models were investigated in this research: the Bell [12], Clark-Davidson [13], Jakob [14], Donohue [4], Chopey [15] models, which determined the pressure drop of the cooled fluid in the shell side of the evaporator. The results indicated an extreme degree of variation with respect to each other and with respect to the results of the laboratory measurement. Based on the measured values, the author presented a new correlation for the determination of the friction factor, one that enabled a more accurate description of the pressure drop of the cooled fluid -

water in the shell side of the evaporator over the range of the Reynolds number and under the operating conditions.

2 Mathematical Models of the Shell-Side Pressure Drop

The total pressure drop ΔP_{tot} on the shell side of the heat exchangers is composed of the pressure drop ΔP_w $\Delta p_w$the flow through the baffle windows, the pressure drop in the cross stream through the tubes ΔP_{cross} and the nozzle pressure drop in the inlet and outlet nozzle ΔP_n.

$$\Delta p_{tot} = \Delta p_w + \Delta p_{cross} + \Delta p_n \tag{2}$$

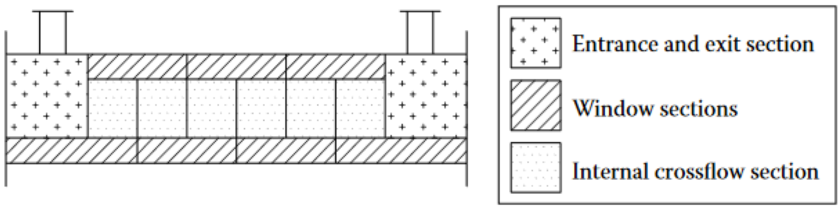

Figure 1
Elements of shell side pressure drop of TEMA E shell [16]

- The pressure drop ΔP_w in the flow through the baffle windows – Window section.

$$\Delta p_w = BF \cdot n_B \cdot \rho \cdot v_w^2 \tag{3}$$

The bypass factor BF comes from the assumption that because of the bypass streams, only 60% of the geometrically calculated flow velocity will be achieved.

Longitudinal flow velocity through the window:

$$v_w = \frac{\dot{V}}{a_w \cdot 3600} \tag{4}$$

- The nozzle pressure drop ΔP_n in the inlet and outlet nozzle – Entrance and exit section.

Pressure drop in the nozzle, $\Delta P_{n,i}$ in the terms of mass velocity through nozzle, G_n is given by

$$\Delta p_{n,i} = \frac{1.0 \cdot G_n^2}{2 \cdot g_c \cdot \rho} = 1.0 \ \cdot v_N^2 \cdot \frac{\rho}{2} \tag{5}$$

Pressure drop in the outlet nozzle, $\Delta P_{n,e,}$ is given by

$$\Delta p_{n,e} = \frac{0.5 \cdot G_n^2}{2 \cdot g_c \cdot \rho} = 0.5 \ \cdot v_N^2 \cdot \frac{\rho}{2} \tag{6}$$

Total pressure drop associated with the inlet and outlet nozzles, ΔP_n is given by

$$\Delta p_n = \frac{1.5 \cdot G_n^2}{2 \cdot g_c \cdot \rho} = 1.5 \cdot v_N^2 \cdot \frac{\rho}{2} \quad (7)$$

- The pressure drop ΔP_{cr} in the cross stream through the tubes – Internal crossflow section.

$$\Delta p_{cr} = BF \cdot (n_B + 1) \cdot f \cdot n_{cross} \cdot v_{cross}^2 \cdot \frac{\rho}{2} \quad (8)$$

The number of rows for cross stream:

$$n_{cross} = \frac{b \cdot D_i}{T} \quad (9)$$

The shell side stream crossflow velocity through the bundle can be determined using the following equations:

$$v_{cross} = \frac{\dot{V}}{3600 \cdot a_{cross}} = \frac{\dot{V}}{3600 \cdot D_i \cdot B \cdot \left(1 - \frac{d_o}{T}\right)} \quad (10)$$

The friction factor f is calculated according to different models Bell [12], Clark-Davidson [13], Jakob [14], Donohue [4] and Chopey [15] with the help of the Reynolds number.

$$Re = \frac{v_{cross} \cdot d_o}{\vartheta} \quad (11)$$

Table 1

Different friction factors in the shell side of the evaporator

Correlations	Equations	Number
Bell [12]	$f = 2.68 \cdot Re^{-0.182}$	(12)
Clark-Davidson [13]	$f = Re^{-0.2} \cdot \frac{3.12}{\sqrt{\frac{T}{d_o}}}$	(13)
Jakob [14]	$f = Re^{-0.2} \cdot \left(1 + \frac{0.47}{\left(\frac{T}{d_o} - 1\right)^{1.08}}\right)$	(14)
Donohue [4]	$f = Re^{-0.2} \cdot \frac{3}{\left(\frac{T - d_o}{d_o}\right)^{0.2}}$	(15)
Chopey [15]	$Re_{CH} = \frac{v_{cr} \cdot (T - d_o)}{\upsilon}$	(16)
	$f = 4 \cdot {Re_{CH}}^{-0.25}$	(17)
	Calculation of the bypass factor BF	
	$BF = R_1 \cdot R_2$	(18)

	$R_1 = 0.75 \cdot \sqrt{\frac{B}{D_i}}$	(19)
	$R_2 = 0.85 \cdot {D_i}^{0.08}$	(20)

3 Experimental Investigation

A measuring system was developed to determine the pressure drop of the cooled fluid in the shell side of the tube bundle evaporator. Considering the structure of the tested heat exchanger, it is horizontal, countercurrent, heat exchanger type 10, DN 500 with 164 tubes 25x2 and the baffle spacing of 100 mm type. The refrigerant R134a flows inside the tubes of the heat exchanger, in the tube bundle, while the working fluid, i.e., the heat dissipating fluid, which in this case is water, is on the outside of the tubes, namely, in the shell side of the heat exchanger. Thermocouples and transducers were used to measure the temperature and pressure of the cooled fluid at the established measuring points, while flow meters were implemented to measure the volumetric flow of the cooled fluid. The measurement layout is shown in Figure 2 below, whereas the accuracy of the measuring instruments is summarized in Table 2.

Table 2

Measured parameters and equipment uncertainty [1]

Measured parameters	Sensor	Uncertainty
Temperatures	K-type thermocouples DS18B20 90807A	$\pm 0.2\ K$
Pressures	Transducers TD220030 ELIWELL EWPA 030	± 1 %
Flow meters	Water flow sensors turbine flowmeters	± 0.2 %

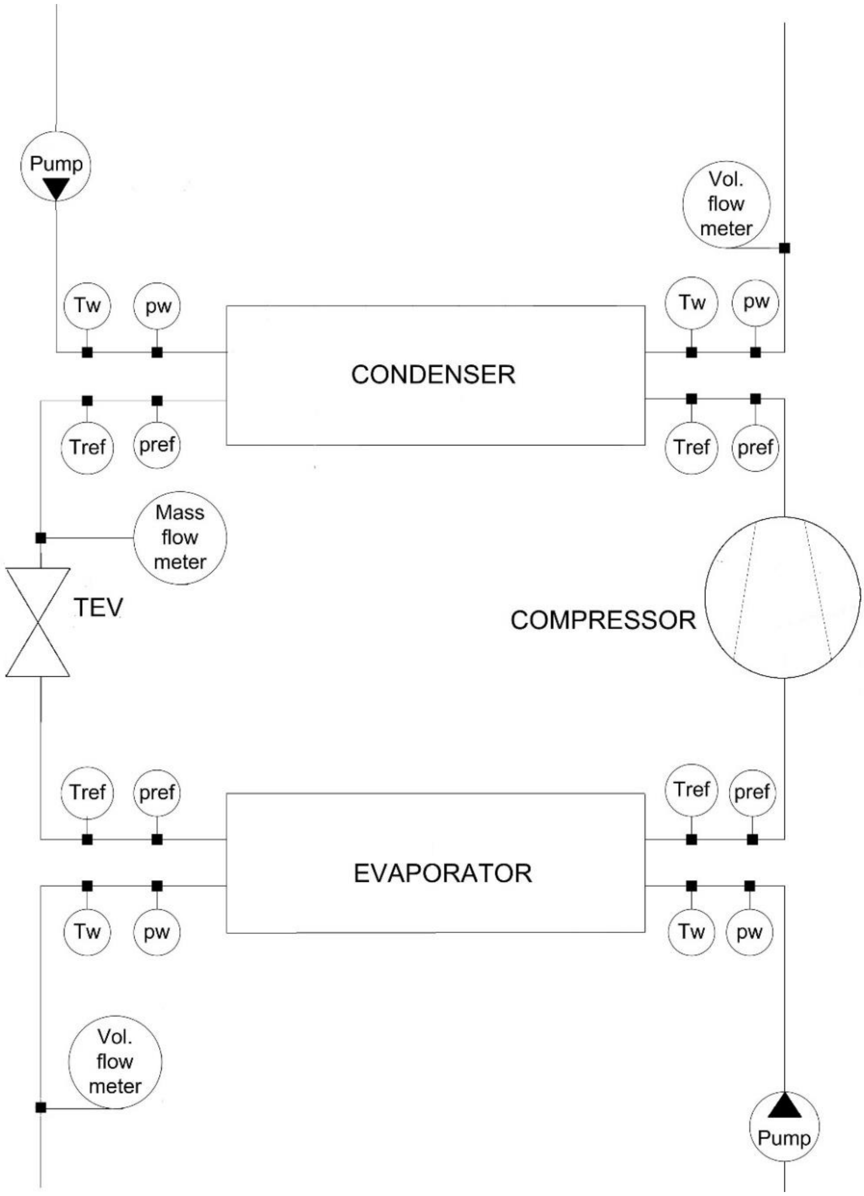

Figure 2
Installation and arrangement of measuring points

		$\pm 0.2\ K$
		$\pm 1\ \%$
		$\pm 0.2\ \%$

4 Result and Discussions

The measurement performed was intended to acquire the new proposed correlation equation (21) for the calculation of the pressure drop of the cooling water in the shell side of the evaporator, which are more accurate and more generally applicable than the results to be obtained by the existing correlations in the literature. Upon the evaluation of measurement results, the following correlation was produced for determining the friction factor in the shell side of the evaporator:

$$f = a \cdot Re^{b} \cdot \left(\frac{T}{d_o}\right). \tag{21}$$

Where: a = 2.7159, b = -0.2023, and T is the triangular pith, do is the tube outer diameter (m).

The presented equation (21) was defined in the following measurement conditions and subject to the following criteria:

- Fluid: *water,*
- Reynolds number: *Re=478 – 7175 (-),* $\mathrm{Re} = 478 - 7175\ [-]$
- Volumetric flow: *V=1-15 (m³/h),* $\dot{\mathrm{V}} = 1 - 15\ {}^{\mathrm{m}^3}/_{\mathrm{h}}$
- Temperature: *T=13 -15 °C* $\mathrm{T} = 13 - 15\ ℃.$

The new proposed correlation equation (21) obtained from the measurement results was compared with values defined from correlations for determining friction factors by Bell [12], Clark-Davidson [13], Jakob [14], Donohue [4], Chopey [15], and summarized in Figure 3 below.

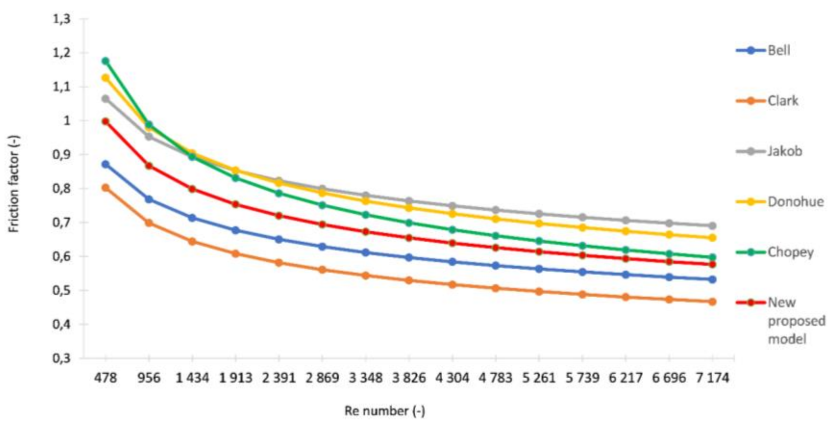

Figure 3

The friction factor as a function of the Reynolds number

The comparison results of the friction factors are presented in Table 3. It can be observed that the best values are yielded by the new proposed correlation equation (21).

Table 3
Comparison of the friction factor results of the different correlations in the shell side of the evaporator

Correlations	Average relative error ε %	Maximum discrepancy ε_{max}% e_{max}%
Bell [12]	10.36	21.64
Clark-Davidson [13]	23.84	34.66
Jakob [14]	13.30	22.17
Donohue [4]	11.74	20.3
Chopey [15]	7.62	20.35
New proposed equation (21)	5.24	9.28

In Figure 4, the values obtained from equation (2) - supplemented according to [12], [13], [14], [4], [15] were compared with the values of equation (21) - for determining the pressure drop of the cooled fluid in the shell side of the evaporator as a function of the volumetric flow.

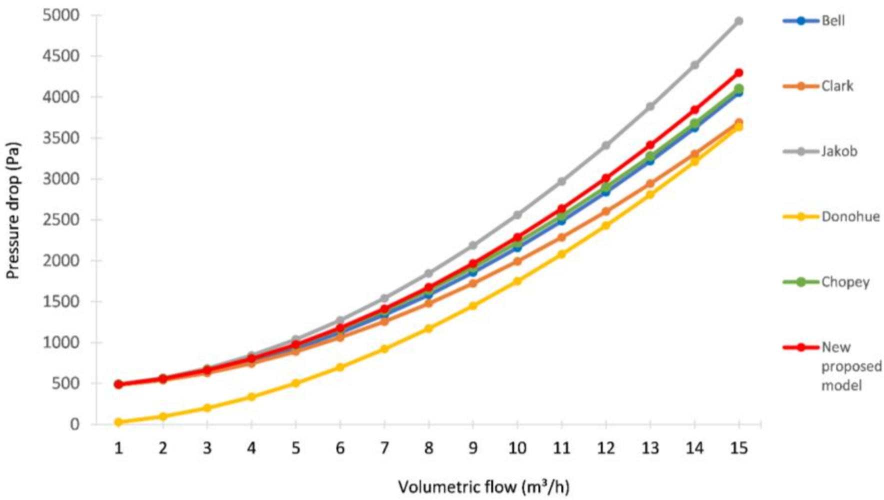

Figure 4
Pressure drop in the shell side of the evaporator

The comparison results of the pressure drop in the shell side of the evaporator are given in Table 4. It can be observed that the sum total pressure drop in the shell side, equation (2) according to the new proposed correlation equation (21) provided the best values.

Table 4

Comparison of the pressure drop results of the different methods in the shell side of the evaporator

Eq. 2 according to correlations:	Average relative error ε%	Maximum discrepancy ε_{max}%
Bell [12]	5.4	12.88
Clark-Davidson [13]	12.48	22.46
Jakob [14]	8.47	14.78
Donohue [4]	91.32	469
Chopey [15]	3.77	9.83
New proposed equation (21)	2.58	5.72

The average discrepancy of calculated and measured values was defined as follows:

$$e = \frac{1}{N} \cdot \sum_{i=a}^{N} \frac{x\,(i)_{pred} - x\,(i)_{exp}}{x\,(i)_{pred}} \tag{22}$$

Where x_{pred} is the calculated value, x_{exp} is experimental value, and N is the number of the data points.

Conclusions

In this research the correlations [12], [13], [14], [4], [15] were investigated for calculating the pressure drop in the cooling fluid in the shell side of the evaporator and it was found that they had a highly significant variance in their values, which is illustrated in Figures 3 and 4. The aim of the measurement was to obtain a more accurate and generally applicable calculation formula for the results gained with the examined correlations. The author refined the friction factor and introduced a new proposed correlation, equation (21) to determine the total pressure drop in the internal crossflow section equation (8), of the second term of equation (2). Thus, the new proposed models for determining the total pressure drop describe with greater accuracy the flowing processes of cooled water in the shell side of the evaporator than the investigated models.

It can be seen from Table 4 that the maximum discrepancy - from measurement values - of the values yielded by the new model for determination of the pressure drop in the shell side of the evaporator as set up by the author is ε_{max}=5.72%, while the average discrepancy is only ε=2.58%, which is the better value among the presented models.

References

[1] R. Santa, L. Garbai, I. Fürstner, "Optimization of heat pump system," Energy, Vol. 89, pp. 45-54, 2015, https://doi.org/10.1016/j.energy.2015.07.042

[2] Kassai M, "Development and experimental validation of a TRNSYS model for energy design of air-to-water heat pump system," Thermal Sciences, Vol. 24, No. 2, pp. 893-902, 2020, https://doi.org/10.2298/TSCI181206070K

[3] D. Q. Kern, “Process Heat Transfer,” McGraw-Hill, N.Y, 1950

[4] D. A. Donohue, “Heat Transfer and Pressure Drop in Heat Exchangers,” Industrial and engineering chemistry, Vol. 41, No. 11, pp. 2499-2511, 1949

[5] D. W. Green, R. H. Perry “Heat transfer Equipment,” Perry`s Chemical Engineers Handbook, 8th edition, chapter 12, pp. 670-671, 2007

[6] E. S. Gaddis, V. Gnielinski, “Pressure drop on the shell side of shell-and-tube heat exchangers with segmental baffles,” Chemical Engineering and Processing, Vol. 36, pp. 149-159, 1997, https://doi.org/10.1016/S0255-2701(96)04194-3

[7] K. J. Bell, “Delaware method for shell side design,“ S. Kakac, A. E., Bergles, F. and Mayinger, Heat Exchangers: Thermal Hydraulic Fundamentals and Design, Hemisphere/McGraw-Hill, Washington, DC, 1981

[8] H. Li, V. Kottke, “Effect of baffle spacing on pressure drop and local heat transfer in shell-and-tube heat exchangers for staggered tube arrangement,” International Journal of Heat and Mass Transfer, Vol. 41, No. 10, pp. 1303-1311, 1998, https://doi.org/10.1016/S0017-9310(97)00201-9

[9] D. Col, A. Muzzolon, P. Piubello, L. Rossetto, “Measurement and prediction of evaporator shell-side pressure drop,” International Journal of Refrigeration, Vol. 28, pp. 320-330, 2005, https://doi.org/10.1016/j.ijrefrig.2004.09.012

[10] U. C. Kapale, S. Chand, “Modeling for shell-side pressure drop for liquid flow in shell-and-tube heat exchanger,” International Journal of Heat and Mass Transfer, Vol. 49, pp. 601-610, 2006, https://doi.org/10.1016/j.ijheatmasstransfer.2005.08.022

[11] R. Hosseini, A. Hosseini-Ghaffar, M. Soltani, “Experimental determination of shell side heat transfer coefficient and pressure drop for an oil cooler shell-and-tube heat exchanger with three different tube bundles,” Applied Thermal Engineering, Vol. 27, pp. 1001-1008, 2007, https://doi.org/10.1016/j.applthermaleng.2006.07.023

[12] K. J. Bell, “Thermal design of heat transfer equipment,” Perry’s Chemical Engineers Handbook, R. H. Perry, C. H. Chilton, 5th ed, Section 10, pp. 10-22, 1973

[13] L. Clarke, R. L. Davidson, “Manual for Process Engineering Calculations, “ Second Edition, McGraw - Hill Company, New York, 1962

[14] M. Jakob, “Discussion of Heat Transfer and Flow Resistance in Cross Flow of Gases Over Tube Banks,“ Trans. A. S. M. E., Vol. 60, No. 4, May 1938

[15] N. P. Chopey, “Heat transfer,” Handbook of Chemical Engineering Calculations, McGraw-Hill, New York, 1993