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# Hydraulic Network Modeling to Analyze Stream Flow Effectiveness on Heat Transfer Performance of Shell and Tube Heat Exchangers

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*ABSTRACT*

In this article, stream flow effectivness is based on hydraulic network studied in the shell-side of a shell and tube heat exchange as a case study. For an appropriate heat exchangers rating design to meet a specified duty, it's better to consider each stream flow separately. Using the hydraulic network principals, a set of the correlations for calculating different stream flow rates in the cross and window area, leakage from tube-bundle and shell-baffle bypass are suggested. By the presented correlations, the actual flow direction and different stream flow rates of shell-side fluid for calculating of shell-side heat transfer and pressure drop in different regions between adjacent baffles has been taken into account. Also, the effects of each stream flow in each baffle section on the overall heat transfer coefficient (HTC) and pressure drop could be investigated. The comparison results of using these correlations and results of published values, like Bell-Delaware method and Kern correlations, is reasonable, which can be used in the optimum design of shell and tube heat exchangers with segmental baffles. Also, according to the results, the cross flow stream show much better heat transfer performance with lower pressure drop behavior than window stream at the same mass flow rates. Average heat transfer performance of window-section is almost 7-12% of overall heat transfer performance for studied case study.

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**NOMENCLATURE**

| | | | |
|---|---|---|---|
| B | Central baffle spacing (m) | $\dot{m}_s$ | Shell-side mass flow rate (kg/s) |
| $B_c$ | Baffle cut (%) | Pr | Prandtl number, $Pr_s = \frac{C_{p,s}\mu_s}{K_s}$ |
| $B_t$ | Baffle thickness (m) | $P_t$ | Tube pitch (m) |
| $C_p$ | Fluid specific heat (W/kg.K) | $Re_j$ | Stream flows Reynolds number, $Re_j = \frac{d_e G_j}{\mu}$ |
| $d_e$ | Equivalent diameter (m) | $r_s$ | Shell and tube fraction area, $r_s = \frac{S_{sb}}{S_{sb}+S_{tb}}$ |
| $d_0$ | Outer tube diameter (m) | $r_{lm}$ | Leakage fraction area, $\frac{S_{sb}+S_{tb}}{S_m}$ |
| $D_{otl}$ | Outer tube limit diameter (m) | $S_m$ | Cross-flow area ($m^2$) |
| $D_s$ | Inner diameter of shell (m) | $S_{tb}$ | Tube-to-baffle leakage area ($m^2$) |
| $f_j$ | Fanning factor, $f_j = f_j$ ($Re_j$, geometry) | $S_{sb}$ | Shell-to-baffle leakage area ($m^2$) |
| $F_{sbp}$ | Bypass flow fraction factor, $F_{sbp} = \frac{S_b}{S_m}$ | $S_b$ | Bundle bypass flow area ($m^2$) |
| $f_{id}$ | Ideal tube bank friction factor | $S_w$ | Window flow area ($m^2$) |
| $G_j$ | Mass velocity, $G_j = \frac{\dot{m}_j}{S_j}$ (kg/s.m$^2$) | **Greek symbols** | |
| $h_j$ | Heat Transfer Coefficient, HTC (W/m$^2$.K) | $\mu$ | Fluid viscosity (N.s/m$^2$) |
| $J_j$ | Colburn factor | $\mu_w$ | Viscosity in wall temperature (N.s/m$^2$) |
| $J_\mu$ | Viscosity ratio, $J_\mu = (\frac{\mu}{\mu_w})^{0.25}$ | $\rho$ | Density (kg/m$^3$) |

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| | | | |
|---|---|---|---|
| $K_s$ | Fluid thermal conductivity (W/m.k) | $\Delta p$ | Pressure drop (kpa) |
| l | Tube effective length (m) | $\delta tb$ | Tube-baffle clearance |
| $m^{\cdot}_j$ | Different stream flow rate (kg/s) | $\delta p$ | Pass partition clearance |
| $n_b$ | Baffle numbers | $\delta sb$ | Shell-baffle clearance |

## 1. INTRODUCTION

In industry and engineering applications, a shell and tube heat exchanger has played a vital role, since it is used in industry such as: power plant, process industry, chemical and nuclear reactors, petrochemical industry, air-conditioning units, etc. Shell and tube models can be classified according to the discretization details used by different types of models such as: one zone; two zones, finite element method and hydraulic network model. The model used in this research is based on hydraulic network model principles. Many handbooks covering the design of shell and tube heat exchangers are available.

Taborek [1], Hewitt [2], Shah and Sekulic [3] and Serth and Lestina [4] were the first to give a physical description of thermal analysis of shell and tube heat exchangers. Shell-side flows of shell and tube heat exchanger are particularly complex, because of many geometrical factors involved and the different behavior of the stream flows which flow across the tube bundle and leakage and bypass areas. Tinker [5] was the first to give a physical description of this process which these were further developed by others. Critical review of this method is developed by Bell-Delaware [6] and Kern [7]. In the Bell-Delaware method [6], empirical correlations is used for calculating the shell-side heat transfer coefficient and pressure drop of shell and tube heat exchangers. In this method, they assume that all the shell-side fluid flowing across the tube bundle without leakage, and then a correction factors are applied to accounts for the various leakage and bypass streams. In the Kern method correlations [7, 8], authors have assumed a model which all fluid flow rates in shell-side stream is perpendicular to the tube bundle. However, in a baffled shell-tube heat exchanger a different fraction of fluid flows in each baffled section. So, it is imperative to account for each stream effect on heat transfer performance and pressure drop on each regions shell-side individually.

Wills and Johnston [9] published a simplified set of correlation for the flow resistance coefficient to calculate shell-side pressure drop as a solution of the hydraulic equations. A great effort to use new type of baffles, like helical and rod baffles are done. More attentions of Tahery et al. [10] were paid on technique to improved shortcomings of the conventional segmental baffles using NTW shell and tube heat exchanger. Their present method is extended to the pressure drop and heat transfer performance of the cases with no tubes in the window region. Azar et al. [11] has modified the existing heat transfer and pressure drop correction factors of the modified Bell-Delaware method used for heat exchangers with segmental baffles, taking into consideration the helical baffles geometry. The results of their comparison show that the proposed method is accurate and can be used by designers confidently. Parikhshit et al. [12] have used the concept of Finite Element Method (FEM) to calculate pressure drop on the shell-side of a shell and tube heat exchangers. In their model, the shell-side region is discredited into a number of elements and by taking into account the effect of flow pattern, the pressure drop on the shell-side of a shell-tube heat exchanger is determined. Baghban et al. [13] used experimental and theoretical methods for thermal analysis of shell and tube heat exchangers. In this paper, the effect of major geometric parameters like baffle cut and baffle spacing by a new approach which including entrance and exit regions have been considered. The results show that these parameters have important role in heat transfer rate, velocity and temperature field of shell-side flow of investigated shell and tube heat exchanger. Besides the improvement of structure, the modified heating and cooling medium are used to improve the performance of heat exchanger systems. Nandan and Singh [14] experimentally investigated the use of air bubble injection technique. Based on the results, injection air bubbles throughout the tube enhances the heat transfer rate by 25-40% at different Reynolds number by increasing the turbulence of the flowing fluid.

Shell-side flow over tube bundles in different sectional area are particularly complex, because of the many geometrical factors and the many possible fluid flow paths involved. It is imperative to account for each stream effect on heat transfer performance and pressure drop on each regions shell-side individually by extrapolation from hydraulic network concepts. In the present work, an attempt has been made to develop the concept of the stream analysis in hydraulic network model to predict the different stream flow rates, pressure drop and heat transfer coefficient of each stream flow at different regions of a shell-and-tube heat exchangers with segmental baffles. By this method, designers considered more fundamental principles in hydraulic networks and didn’t need correction factors for the effects of deviation from the ideal tube bank flow.

## 2. GOVERNING EQUATIONS

### 2. 1. Definition of Stream Flow Areas and Stream Flow Rates

Because of tube-baffle holes and shell-baffle clearance, a fraction of fluid flow across each baffle section can become bypass or leakage through each gap respectively, which affect the window and cross stream. It is necessary to analyze them individually in the different section to see their effectiveness on shell-side heat transfer performance and pressure drop. The shell-side flow is divided into individual streams: cross-flow stream, tube-baffle leakage stream, shell and tube leakage stream, bundle-shell bypass stream, pass partition bypass stream and stream W as window-section stream. Also, to account for non-uniform flow rates, this model requires the shell-side of heat exchanger to be divided into three main flow-sections; window-section, cross-section and tube-baffle clearance. The expressions to calculate other geometrical characteristics are given in the following by Equation (1) to (4), which can also be found in the literature [1-3]. Figure 1 shows the stream flow and different baffle-section regions of shell and tube heat exchanger. In addition, the shell-side equivalent hydraulic network for different stream flow is shown in Figure 2.

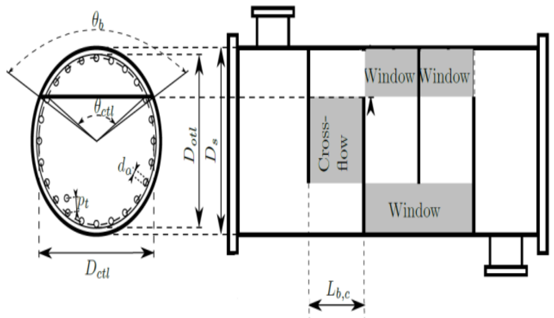


**Figure 1.** Schematic view of shell and tube heat exchanger with segmental baffles

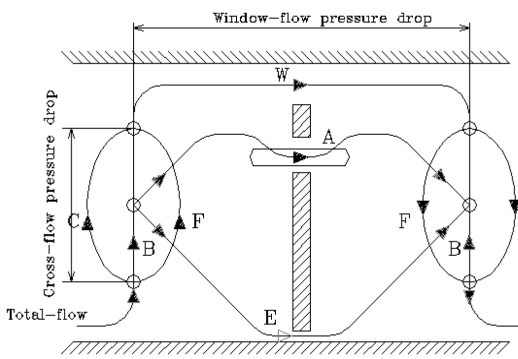


**Figure 2.** Equivalent hydraulic network for shell-side flow [1, 4]

The cross; bypass and leakage flow areas in the shell-side can be obtained as Equations (1)-(4), respectively: [1-3]

$$S_m = B[(D_s - D_{otl}) + \left(\frac{D_{otl}-d_0}{P_t}\right)(P_t - d_0)] \quad (1)$$

$$S_{bp} = B[\,(D_s - D_{otl}) + N_p\delta_p] \quad (2)$$

$$S_{tb} = 0.5\pi d_0 \delta_{tb} N_t (1 + F_c) \quad (3)$$

$$S_{sb} = \pi D_s \left(\frac{\delta_{sb}}{2}\right)\left(\frac{360-\theta_{ds}}{360}\right) \quad (4)$$

Using balanced pressure drop and mass conservation correlation leads to the following correlations of cross flow mass velocity [10]:

$$G_{CB} = \left(\frac{(d_e/\mu)^{0.168} f_{id} G_s^2 \varphi \exp(-\varepsilon F_{sbp})}{0.324}\right)^{0.546} \quad (5)$$

where, the correction factor, ε, various from 0.8 for very large $N_c$ to 1.8 at small $N_c$. Also, φ is defined from the following equation [10]:

$$\varphi = \exp\left[-1.33(1 + r_s) r_{lm}^{[-0.15(1+r_s)+0.8]}\right] \quad (6)$$

Now, the cross-stream flow rate is obtained from the following equation:

$$\dot{m}_{CB} = G_{CB} S_m \quad (7)$$

Correlation to calculate tube-baffle leakage mass velocity is defined as Equation (8) [10]:

$$G_A = \left(\frac{10.2 G_s^{1.83} D_s \delta_{tb} (d_e/\mu)^{0.1}}{N_c B_t F_c n_b (P_t - d_0)}\right)^{0.58} \quad (8)$$

Then, the tube-baffle leakage flow rate is as following equatressed as the following equation:

$$\dot{m}_A = G_A S_{tb} \quad (9)$$

Using continuity and compatibility principles for the total cross stream flow the bypass, effective cross stream flow rate and shell-baffle leakage through the clearance between the edge of a baffle and the shell are as following equations:

$$\dot{m}_{CF} = \left(\frac{S_{bp}}{S_m}\right) \dot{m}_{CB} \quad (10)$$

$$\dot{m}_B = \dot{m}_{CB} - \dot{m}_{CF} \quad (11)$$

$$\dot{m}_E = \dot{m}_S - (\dot{m}_B + \dot{m}_{CF} + \dot{m}_A) \quad (12)$$

Values of different flow rates can be used individually for each stream to calculate Reynolds number, pressure drop and shell-side heat transfer coefficient in each area-section. Using this method provides a good and complete representation of the real situation without using many correction factors.

**2. 2. Calculation of Heat Transfer Coefficient (HTC) and Pressure Drop** The present investigation calculates $h_x$, $h_w$, $h_A$ from the ideal tube bank correlation using appropriate Reynolds number for each stream flow and the fraction of each section area occupied by the tubes. Correlation to obtain the total heat transfer coefficient is shown as Equation (13): [1, 3, 10]

$$h_{0,tot} = F_c h_c + 2F_w h_w + F_c h_A \tag{13}$$

which, $F_c$ , $F_w$ are as fraction factors and is expressed as follow [1, 3]:

$$F_c = 1 + \frac{1}{\pi}(\theta_{ctl} - \sin\theta_{ctl}),\ F_w = 0.5(1 - F_c) \tag{14}$$

Correlations for evaluating shell-side cross and window stream heat transfer coefficient, $h_c$, $h_w$, suggested by Shah and Sekulic, which is expressed as below: [1, 3, 10]

$$h_j = J_j c_{p,s} \dot{m}_j Pr^{-\frac{2}{3}} j_\mu{}^m \tag{15}$$

The Colburn factor of the ith stream in each shell-side section is obtained as follows: [1, 3]

$$J_j = a_1\left(\frac{1.33}{p_t/d_0}\right)^a (Re_j)^{a_2} \tag{16}$$

$$a = \frac{a_3}{1+0.14(Re_i)^{a_2}} \tag{17}$$

where, the empirical constants values of $a_1$ , $a_2$ , $a_3$ and $a_4$ are listed in Table 1 [3].
Heat transfer coefficient for flow which passes through tube-baffle holes is expressed as Equation (18) [10]:

$$h_A = 0.029\frac{K}{de} Re_A^{0.76} Pr^{-\frac{2}{3}} \tag{18}$$

For a shell and tube heat exchanger, the pressure drop is equal to the sum of the cross flow pressure drops $\Delta p_C$, the window pressure drops $\Delta p_w$ and the inlets and outlet baffle zones $\Delta p_n$, which is expressed as follows: [1, 3]

$$\Delta P_{s,tot} = 2\Delta P_c n_b + 2\Delta P_w(n_b - 1) + 2\Delta P_n \tag{19}$$

where, $n_b$, defined as the total number of baffle of the shell-side.

Equation for evaluating $\Delta p_c$ is suggested by Shah and Sekulic, which is expressed as Equation (20) [3, 10]:

$$\Delta p_c = \frac{2f_{j,c}G_c{}^2 N_c}{\rho_c\left(\frac{\mu_{s,w}}{\mu_s}\right)^{-0.25}} \tag{20}$$

In addition, correlation which is used to evaluate the window-section pressure drop, $\Delta p_w$ , expressed as Equation (21) [10]:

$$\Delta p_w = \frac{(\alpha + 2\beta f_{j,w} N_{cw}) G_w{}^2}{\rho_c\left(\frac{\mu_{s,w}}{\mu_s}\right)^{-0.25}} \tag{21}$$

The correction factor, α, when tube diameter is 1 " varies from 0.7 to 1.5 and from 0.5 to 1.2 when tube diameter is 3/4 " for very large to small baffle spacing, respectively. Also, the correction factor, β, varies from 0.5 for small baffles cuts/baffle spacing to unity at very large baffles cuts/baffle spacing [10].

Equation to account for differences in baffle spacing, the flow rate and flow distribution in inlet and outlet spaces of shell-side is suggested as Equation (22) [3, 10].

$$\Delta p_n = \frac{2f_{j,n}G_n{}^2 N_c}{\rho_c\left(\frac{\mu_{s,w}}{\mu_s}\right)^{-0.25}} \tag{22}$$

where, $G_n$, in the present article is approximately expressed as the following equation [10]:

$$G_n = \frac{(\dot{m}_0 - \dot{m}_A - \dot{m}_E)}{S_n} \tag{23}$$

The Fanning friction factor of each stream flow are expressed by Equation (24) [2, 3]:

$$f_j = b_1\left(\frac{1.33}{p_t/d.}\right)^b (Re_j)^{b_2} \tag{24}$$

$$b = \frac{b_3}{1+0.14(Re_j)^{b_4}} \tag{25}$$

where, the empirical constants values of $b_1$ , $b_2$ , $b_3$ and $b_4$ are listed in Table 1 [3].

## 3. PHYSICAL PROPERTIES AND MODEL VALIDATION

The configuration and geometric features of the tested heat exchanger as a case study is AES type shell and tube heat exchanger with single pass and copper tubes are given in Table 2. The Kerosene is taken as working fluid for the shell-side, which thermo physical properties of the fluid are listed in Table 3.

**TABLE 1.** Empirical constants coefficient for calculation ideal friction factor $f_j$ and Colburn factor $J_j$ [3]

| Layout angle | Reynolds number | $a_1$ | $a_2$ | $a_3$ | $a_4$ | $b_1$ | $b_2$ | $b_3$ | $b_4$ |
|---|---|---|---|---|---|---|---|---|---|
| 30° | $10^5$–$10^4$ | 0.321 | −0.388 | 1.450 | 0.519 | 0.372 | −0.123 | 7.00 | 0.500 |
| | $10^4$–$10^3$ | 0.321 | −0.388 | | | 0.486 | −0.152 | | |
| | $10^3$–$10^2$ | 0.593 | −0.477 | | | 4.570 | −0.476 | | |
| | $10^2$–10 | 1.360 | −0.657 | | | 45.100 | −0.973 | | |
| | <10 | 1.400 | −0.667 | | | 48.000 | −1.000 | | |
| 45° | $10^5$–$10^4$ | 0.370 | −0.396 | 1.930 | 0.500 | 0.303 | −0.126 | 6.59 | 0.520 |
| | $10^4$–$10^3$ | 0.370 | −0.396 | | | 0.333 | −0.136 | | |
| | $10^3$–$10^2$ | 0.730 | −0.500 | | | 3.500 | −0.476 | | |
| | $10^2$–10 | 0.498 | −0.656 | | | 26.200 | −0.913 | | |
| | <10 | 1.550 | −0.667 | | | 32.000 | −1.000 | | |
| 90° | $10^5$–$10^4$ | 0.370 | −0.395 | 1.187 | 0.370 | 0.391 | −0.148 | 6.30 | 0.378 |
| | $10^4$–$10^3$ | 0.107 | −0.266 | | | 0.0815 | +0.022 | | |
| | $10^3$–$10^2$ | 0.408 | −0.460 | | | 6.0900 | −0.602 | | |
| | $10^2$–10 | 0.900 | −0.631 | | | 32.1000 | −0.963 | | |
| | 10 | 0.970 | −0.667 | | | 35.0000 | −1.000 | | |

**TABLE 2.** Geometry specifications of shell and tube heat exchangers

| Item | Dimensions and description | |
|---|---|---|
| **Shell-side parameters** | D0/Di/mm | 500/488 |
| | Material | Copper |
| **Tube parameters** | d0/di/mm | 25.4/24.2 |
| | Effective length/mm | 4250 |
| | Number | 140 |
| | Layout pattern | Square |
| | Tube pitch/mm | 32 |
| | Material | 0Cr18Ni9 |
| **Baffle parameters** | **Baffle pitch/mm** | **98** |
| | **Thickness/mm** | **4.57** |

**TABLE 3.** Thermo physical properties of fluids [10, 11]

| Shell-side | Kerosene |
|---|---|
| **Density (kg/m$^3$)** | 785 |
| **Specific heat (kj/kg.K)** | 2.47 |
| **Kinematic viscosity (kg/ m.s)** | 0.000401 |
| **Thermal conductivity(W/m. K)** | 0.133 |

Following assumptions are made; the fluid flow and heat transfer performance are turbulent and in steady-state, the tube wall temperature is kept constant, shell-side wall insulation from the environment is well done and heat losses are neglected.

A typical design procedure of the present method is summarized with the flowchart of Figure 3. This figure showing steps for thermal and mechanical design involved in studying the affect of different stream flow on heat transfer coefficient and pressure drop at different baffle areas.

Validation between presented method and experimental data show that the deviation of mass flow rate between the proposed method and experimental data derived from references [8] are between the ranges -8% to 15% at different baffle areas. Moreover, as a validation, two different method; the Bell-Delaware method [6] and Kern correlations [7], are used for calculation the shell-side overall heat transfer coefficient ,$h_s$, total pressure drop, $\Delta P_s$, and the ration, $h_s/\Delta P_s$, in the described shell and tube heat exchanger.

These validation results are reported in reference [10]. The results show that the difference between the predicted and published values is quite reasonable. So, this method can be used with confidence in heat exchangers shell-side calculation [10].

## 4. RESULTS AND DISCUSSION

After having establisehed the accuracy of the hydrualic method employed in the present investigation, the detailed results in the terms of each steram flow rate, pressure drop and the heat transfer performance distribution is discussed. Different stream flow rates at different baffle section of defined heat exchanger are presented in Figure 4. Based on the results, the enhancement of mass flow rate has strong influence on different stream flow rates. This variation trend is because by the increase of mass flow rate the shell-side velocity increase, thus the stream flow rate is enhanced because of the velocity increase. Each of the streams has a certain flow fraction of the total flow, so each stream have different influence on heat transfer performance and pressure drop in the shell-side. As indicated in this figure, at the same stream flow rate the window flow rate is bigger than the other streams flow rate. Hence, the Reynolds number of window flow is bigger than the other streams, leading to a significant increase in pressure drop in the window-section at the same mass flow rate.

HTC, pressure drop and the heat transfer coefficient per pressure drop of each stream flow were shown in Figures 5-7, respectively. The obtained results show that the cross stream flow has the most important role in the shell-side heat transfer intensity, because the must number of tubes stand in this section of baffles.

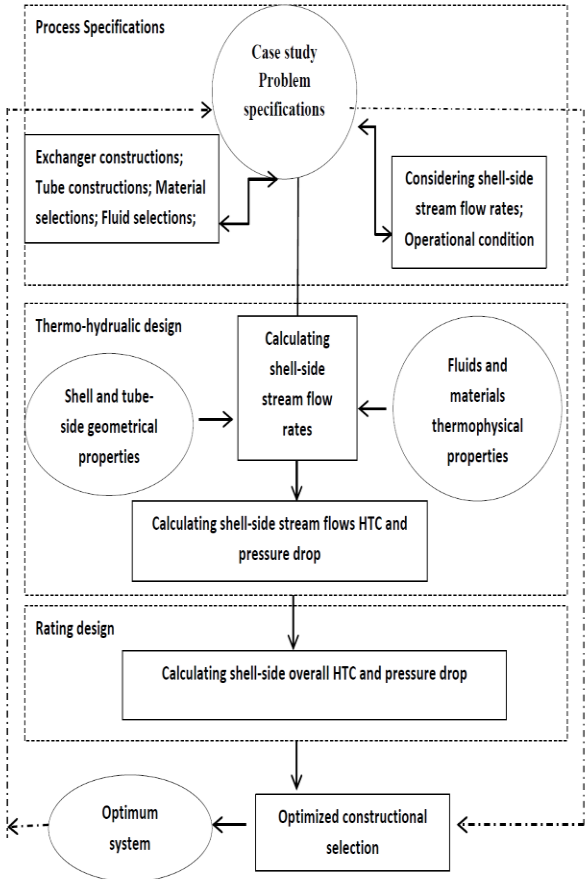


**Figure 3.** Investigated shell and tube heat exchanger design methodology

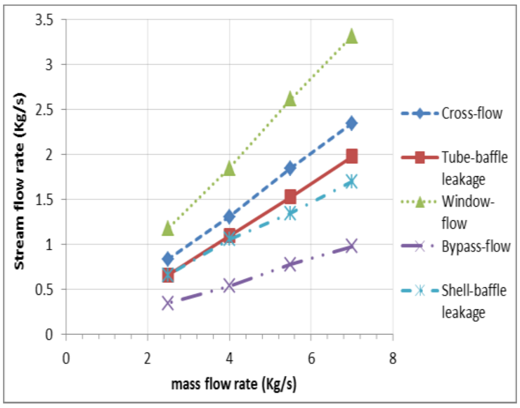


**Figure 4.** Stream flow rates at differ baffle-sections

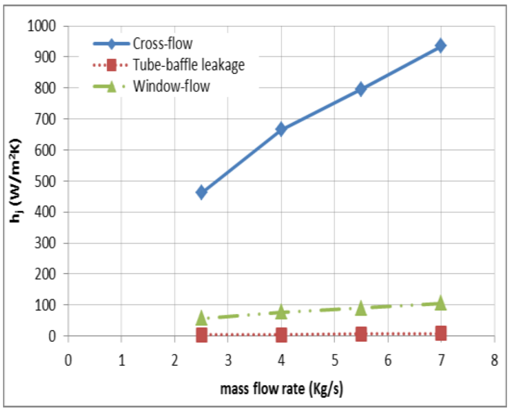


**Figure 5.** Comparisons of the stream flow effectivness on mean convective heat transfer coefficient, $h_j$ (W/m$^2$K)

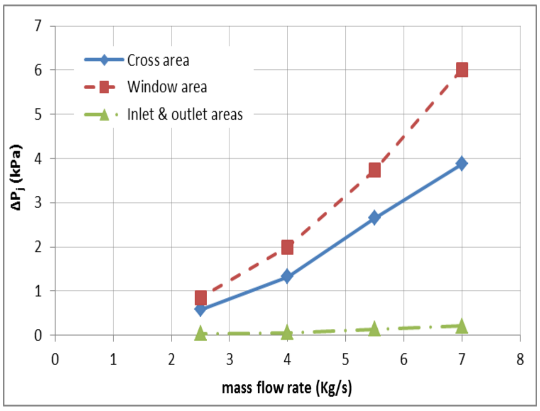


**Figure 6.** Comparisons of the stream flow effectivness on the total pressure drop, $\Delta P_j$ (kPa)

However, since the bypass-bundle area in the cross-section has a lower resistance than through the bundle, therefore the prime area where the flow can bypass is the area between the shell wall and the tube bundle, which decrease the effective cross stream flow rate. By decreasing the cross flow rate, the efficiency of the heat transfer performance and the pressure drop in the cross section will be decreased.

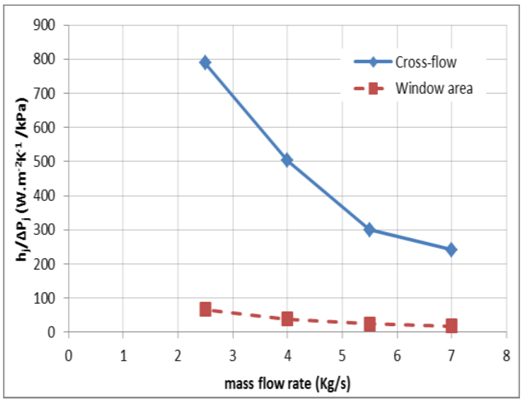


**Figure 7.** Comparisons of the stream flow effectivness on the overall heat transfer performance index, $h_j/\Delta P_j$ (W.m$^2$K$^{-1}$/kPa).

Though, two fluids in the tubes and passing from tube-baffle clearance are at different temperature and also separated by solid wall take acts in causing heat transfer and pressure loses. However, the tube-baffle holes are only partially effective in heat transfer performance because litte tube surface, so this stream flow usually has a relatively small effect on heat transfer, Since it's not great concern if its flow fraction is decreased. The results show that shell-side flow cross a tube bank has better heat transfer performance than the flow parallel to a tube bank at the same velocity. Also, the difference between heat transfer coefficient and pressure drop between the cross and window section is decreased by increasing mass flow rate. For the fixed thermal load and allowed pressure drop condition, the heat transfer coefficient per pressure drop is the most important and also meaningful comparison criterion to evaluate the heat transfer performance of heat exchangers.

Enhancement of flow rate causes a significant increase in cross flow heat transfer coefficient, which causes a significant enhancement in window flow pressure drop. Since the cross-flow stream is effective from heat transfer point of view, it is better if its flow fraction is large. The bigger $h_j/\Delta P_j$ means that the heat exchanger has better heat transfer performance. Based on the obtained results from proposed method, enhancement the mass flow rate causes a significant enhancement in pressure drop/HTC ratio in baffle-window section. So, the cross flow stream at the same mass flow rates process better performance than window flow stream, which increasing mass flow rate causes a significant deterioration in the shell-side heat transfer performance. Since decreasing mass flow rate may cause the sell-side velocity in the cross and the window section decrease, so the cross section fluid flows in a smoother manner which in turn leads to fewer disturbances in the both cross and window-section. Furthermore, in the window-section stream flow area is small relative to the stream flow rate, so increasing mass flow rate caused higher pressure drop.

## 4. CONCLUSION

In the present study, an analytical method based on the concepts of hydrualic network method by new correlations was developed for the shell-dide flow and heat transfer performance of a shell and tube heat exchanger with segmental baffles. Based on the analysis, the following conclusions are obtained:

1. Using proposed correlation is applicable to obtain HTC and pressure drop separately at different baffle sections. The results show that the difference between the predicted and published values, like Bell-Delaware and Kern correlation, is quite reasonable. So, this method can be used in heat exchangers shell-side calculation.
2. Investigations were done for different mass flow rates at different baffle section. The results using proposed correlations show that the increase of window-section pressure drop is almost 35% bigger than the pressure drop in cross-section.
3. Shell-baffle leakage stream is the least effective for heat transfer point of view, approximately 1-3% of overall heat transfer coefficient. Because it may not be in contact with any tube.
4. Based on data at the same mass flow rate, the window-flow stream is partially effective in shell-side heat transfer coefficient, because it contact with a little fraction of tubes.
5. The bypass flow rates cause the pressure drop to decrease but has no effect in heat transfer performance. The design of the tube bundle should be such that it minimize this flow fraction.
6. At the same stream flow rate, the window flow rate is approximately between 20-40 % which is bigger than the cross-section streams flow rate. It Leads to a significant increase in pressure drop in the window-section.

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# Hydraulic Network Modeling to Analyze Stream Flow Effectiveness on Heat Transfer Performance of Shell and Tube Heat Exchangers

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**چکیده**

در این مقاله، براساس اصول شبکه هیدرولیک جریانی تاثیر جریان سمت پوسته مبدلهای حرارتی پوسته و لوله ای برای تحلیل نمونه مطالعاتی مبدل حرارتی پوسته و لوله مورد بررسی قرار گرفت. برای طراحی مناسب Rating مبدلهای حرارتی و دستیابی به قابلیت حرارتی معین بایستی تاثیر این جریانها بصورت مجزا مورد بررسی قرار گیرند. بعنوان یک راه حل با استفاده از اصول شبکه هیدرولیک جریانی، مجموعه معادلاتی برای محاسبه دبی های جریان مقاطع عرضی و پنجره جریان، نشتی لوله–باندل و جریان بای پس پوسته–لوله پیشنهاد گردید. با بکارگیری معادلات پیشنهادی جهت واقعی جریان و دبی های جریان سیال سمت پوسته برای محاسبه انتقال حرارت و افت فشار در مناطق مختلف بین بافل های متناظر مدنظر قرار می گیرد. همچنین با این روش تاثیر جریان مقاطع مختلف بافل بر روی ضریب انتقال حرارت و افت فشار کل سمت پوسته این مبدلها را می توان تعیین کرد. مقایسه نتایج بدست آمده از این معادلات و نتایج روشهای دیگری، همچون روش Bell-Delaware و معادلات Kern، قابل قبول بوده و از این روش می توان در طراحی بهینه مبدل های حرارتی پوسته و لوله ای استفاده کرد. همچنین براساس نتایج، جریان مقطع عرضی عملکرد حرارتی بهتر و افت فشار کمتری نسبت به جریان مقطع پنجره در دبی های جریان جرمی یکسان دارد. میانگین عملکرد حرارتی مقطع پنجره تقریبا ۷–۱۱٪ از عملکرد حرارت کل نمونه مطالعاتی می باشد.

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