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New models for conventional and heat exchangers enhanced with tube inserts for heat exchanger network retrofit

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HIGHLIGHTS

- Simple but reliable performance model of shell-and-tube heat exchanger is proposed.
- Performance correlations for twisted-tape and wired-coil inserts are introduced.
- The models are suitable for heat exchanger network retrofit optimization algorithms.

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ABSTRACT

The retrofit of heat exchanger networks requires detailed models of the heat exchangers for the detailed assessment of network performance. Network retrofit options include heat transfer enhancement. There is thus a requirement for detailed models of heat exchanger performance, including heat transfer enhancement, suitable for inclusion in network retrofit optimization algorithms. Such models must be robust, computationally efficient and accurate enough to reflect the heat transfer and pressure drop performance of actual equipment. This paper presents a simple but reliable method to model the performance of shell-and-tube heat exchangers, including heat transfer enhancement with tube inserts. Two commonly used tube inserts, twisted tapes and coiled wires, are investigated for tube-side heat transfer augmentation. A new set of consistent performance correlations covering all flow regions for twisted-tape and wired-coil inserts is introduced. The models can be conveniently used for existing exchanger simulation, rating and assessment of heat transfer enhancement. They can also be used for sizing new exchangers to be introduced in a network. A case study illustrates the application of these performance models.

1. Introduction

Although the commercial, proprietary computer software models available in the industry can address the complexity of shell-and-tube heat exchangers, these codes are not suitable for the optimisation of heat exchanger network retrofit. The published performance models for shell-and-tube heat exchangers [1–3] are not accurate enough for the detailed examination of networks, especially for the shell-side of shell-and-tube heat exchangers. Thus a compact but accurate and reliable method is required for the shell-and-tube heat exchanger performance calculation suitable for heat exchanger simulation and optimisation for use in network retrofit.

One of the major problems in heat exchanger network retrofit is that increasing heat recovery for the network requires additional heat transfer area in a number of locations. This has a detrimental effect on the economics of network retrofit, as the provision of such additional heat transfer area can be extremely expensive in an existing design. One way to overcome this problem is to enhance the performance of the existing heat exchangers. Intensified heat transfer techniques provide additional area indirectly by increasing the overall heat transfer coefficient. If the inside film heat transfer coefficient is controlling then tube-side heat transfer enhancement can be considered. Tube inserts are frequently used in such applications because inserts are cheap and easy to install in an existing heat exchanger. Numerous research investigations have been made into tube inserts [4–6]. Twisted tapes [7,8] and coiled wires [9–12] are two commonly used designs. Wire mesh inserts, for example, the hiTRAN inserts from Cal Gavin Ltd. have been successfully

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applied to enhance many laminar flow services. These inserts tend to yield much higher heat transfer to pressure drop augmentation ratios than the twisted-tape and coiled wire inserts. Unfortunately, no reliable published correlations are available for such mesh inserts and therefore cannot be included in the scope of this work. There are significant discrepancies among the existing correlations proposed for heat transfer and pressure drop prediction for twisted-tape and wire coil inserts. It is necessary to provide a set of reliable and consistent performance correlations covering the whole flow regions for these inserts. Then based on the detailed performance analysis of these enhancement inserts, it is possible to consider heat exchanger and heat exchanger network retrofit problems in a practical and consistent way, including multiple tube passes, shell passes and exchanger pressure drops.

This paper is intended to address these issues. A simple but reliable shell-and-tube heat exchanger performance model is developed to calculate single-phase heat transfer coefficients and pressure drops for a typical single segmental shell-and-tube heat exchanger. Consistent twisted-tape and wired-coil inserts enhancement models are presented, which can assess the performance of these tube inserts and allow rapid screening for the heat transfer enhancement technology selection.

2. Shell and tube heat exchanger model

Simple shell-and-tube heat exchanger models will be developed in which heat transfer coefficients and pressure drops are both related to velocity. This is especially useful in retrofit situations, where the allowable pressure drop is often highly constrained. The results from this method compare well with HTRI computer predictions [13,14]. The basic correlations used in the model [13] are suitable for the following conditions:

(i) single phase heat transfer in a shell-and-tube heat exchanger;
(ii) plain tubes;
(iii) single segmental baffles with 20–50% cut;
(iv) physical properties are assumed constant, based on an average between the inlet and outlet conditions for each heat exchanger.

2.1. Tube-side heat transfer coefficient

a. *for laminar flow* $Re \le 2100$ and $L \le 0.05 Re \cdot Pr \cdot d_I$

$$h_T = K_{hT1} v_T^{1/3} \tag{1}$$

where

$$K_{hT1} = 1.86 \frac{k}{d_I} \left[\left(\frac{\rho d_I}{\mu} \right) Pr \left(\frac{d_I}{L} \right) \right]^{1/3} \tag{2}$$

b. *for transition flow* $2100 < Re < 10^4$

$$h_T = K_{hT2} v_T^{2/3} - K_{hT3} \tag{3}$$

where

$$K_{hT2} = 0.116 \frac{k}{d_I} \left(\frac{\rho d_I}{\mu} \right)^{2/3} Pr^{1/3} \left[1 + \left(\frac{d_I}{L} \right)^{2/3} \right] \tag{4}$$

$$K_{hT3} = 14.5 \frac{k}{d_I} Pr^{1/3} \left[1 + \left(\frac{d_I}{L} \right)^{2/3} \right] \tag{5}$$

c. *for fully developed turbulent flow* $Re \ge 10^4$

$$h_T = K_{hT4} v_T^{0.8} \tag{6}$$

where

$$K_{hT4} = 0.023 \frac{k}{d_I} Pr^{1/3} \left(\frac{\rho d_I}{\mu} \right)^{0.8} \tag{7}$$

$$Re = \frac{\rho v_T d_I}{\mu}$$

$$Pr = \frac{C_P \mu}{k}$$

$$v_T = \frac{m_T (N_P / N_T)}{\rho \left(\pi d_I^2 / 4 \right)}$$

Fluid physical properties are taken at the average bulk fluid temperature between the tube-side flow inlet and outlet.

2.2. Tube-side pressure drop

The total tube-side pressure drop ΔP_T for a single shell comprises the pressure drop in the straight tubes (ΔP_{TT}), pressure drop in the tube entrances, exits and reversals (ΔP_{TE}), and pressure drop in nozzles (ΔP_{TN}) [3].

$$\Delta P_T = \Delta P_{TT} + \Delta P_{TE} + \Delta P_{TN} = K_{PT1} N_P L v_T^{2+m_f} + K_{PT2} v_T^2 + K_{PT3} \tag{8}$$

where

$$K_{PT1} = \frac{2 F_C \left(\frac{\rho d_I}{\mu} \right)^{m_f} \rho}{d_I} \tag{9}$$

$$K_{PT2} = 0.5 \alpha_R \rho \tag{10}$$

$$K_{PT3} = \rho \left(C_{TN,inlet} v_{TN,inlet}^2 + C_{TN,outlet} v_{TN,outlet}^2 \right) \tag{11}$$

where

$F_C = 16, \quad m_f = -1 \quad$ for $Re \le 2100$

$F_C = 5.36 \times 10^{-6}, \quad m_f = 0.949 \quad$ for $2100 < Re < 3000$

$F_C = 0.0791, \quad m_f = -0.25 \quad$ for $Re \ge 3000$

$\alpha_R = 3.25 N_P - 1.5 \quad$ for $500 \le Re \le 2100$

$\alpha_R = 2 N_P - 1.5 \quad$ for $Re > 2100$

$C_{TN,inlet} = 0.75 \quad$ for $100 \le Re_{TN,inlet} \le 2100$