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A simple method to calculate shell side fluid pressure drop in a shell and tube heat exchanger

B. Parikshit *, K.R. Spandana, V. Krishna, T.R. Seetharam, K.N. Seetharamu

Department of Mechanical Engineering, PES Institute of Technology, 100 ft. Ring road, Banashankari III stage, Bangalore 560 085, India

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ABSTRACT

Pressure drop predictions on the shell side of a shell and tube heat exchanger (STHX) are investigated using the concept of Finite Element Method (FEM). In this model the shell side region is discretised into a number of elements and by taking into account the effect of flow pattern, the pressure drop on the shell side of a STHX is determined. The present method is simple to apply and the predictions agree reasonably well with a large number of experimental data available in the literature. The range of applicability of the present method extends beyond that used by others in the literature. The earlier predictions were restricted to tubes in the window region, however, the predictions of the present method are extended to the cases of no tubes in the window (NTIW) region also.

1. Introduction

Shell and tube heat exchangers are very widely used in a number of industries and its applications include transformer oil cooling, exhaust gas heat recovery, solvent distillate process, ethanol mash-stillage, power plants, air-conditioning units, etc. This heat exchanger (HX) comprises of one fluid flowing through the tubes and the other fluid flowing in the shell across the tube bundle. The flow in the shell side of a shell and tube heat exchanger (STHX) with segmented baffles is quite complex. The flow in baffle region is illustrated in Fig. 1, in terms of main stream S_H, leakage stream between tubes and baffle S_L and bypass stream between tube bundle and shell S_B. The gaps between a baffle and the tube cause leakage stream S_L, which may modify the main stream S_H significantly. As the tubes cannot be placed very close to the shell, bypass streams S_B may be formed, which also influences the main stream. The flow direction of the main stream relative to the tubes is different in the window sections created by the baffle cut from that in the cross flow sections existing between the segmental baffles. This necessitates the use of different equations to calculate the pressure drop in the window sections to those used in the cross flow sections. The spacing between the tube plates and the first and the last baffle differs in many cases from the spacing between two adjacent baffles. Some of the afore mentioned streams are not present in the first and the last section of the HX. A large number of investigations which describe methods to calculate the shell side pressure drop in a STHX have been published [1–5]. A critical review of these methods is given by Palen and Taborek [6]. They have compared the different calculation procedures against a large number of experimental measurements on small units and on industrial HXs. According to them, the methods of Tinker [3,4] and of Delaware [5] gave the best result as compared to the other methods. The method of Tinker [3,4] has been criticized as it is relatively complicated.

Gaddis and Gnielinski [7] have followed the Delaware method [5] to calculate the shell side pressure drop, except that, instead of using diagrams – as in the Delaware method – to calculate the pressure drop in the ideal tube bank, they have used equations previously presented by them in [8,9]. Correction factors are then used, as in the Delaware method, to take into account the deviation of the flow inside the shell from that in the ideal case of a tube bank. They have compared pressure drops predicted by their model with those obtained experimentally by different investigators. The comparison is represented by the ratio, $\Delta p_m/\Delta p_c$ for all available experimental points, as a function of Reynolds number (Re). The comparison shows that for large number of experimental points the deviations between measurements and theoretical predictions are as high as ±35% for Reynolds number range from 10 to 10^5. Further, about one third of the experimental points have deviations more than ±35%. They also found that in extreme cases, the measured pressure drop is as low as one fifth or as high as four

* Corresponding author.
E-mail addresses: parikshit15b@gmail.com (B. Parikshit), spandi93@gmail.com (K.R. Spandana), vkrishna@pes.edu (V. Krishna), tr.seetharam@pes.edu (T.R. Seetharam), knseetharamu@yahoo.com (K.N. Seetharamu).

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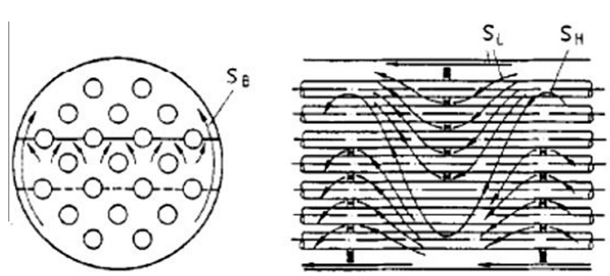
Nomenclature

a	relative transverse pitch of tube bundle, $a = x_t/d_o$
A_{bmin}	minimum cross- sectional area at the baffle tip (m^2)
A_{min}	minimum area of flow of fluid in a cross section (m^2)
A_{nozzle}	nozzle cross-section area (m^2)
b	relative longitudinal pitch of tube bundle, $b = x_l/d_o$
c	relative diagonal pitch of tube bundle, $c = x_d/d_o$
d_0	outer tube diameter (m)
D_{otl}	diameter of the circle encompassing the end tubes (m)
D_s	inner diameter of shell (m)
Eu	Euler number (friction factor), $Eu = \frac{2\Delta P A_{min}^2}{n\rho Q^2}$
Eu_c	corrected friction factor after using angular correction factor
K_ψ	correction for angular flow
k_n	nozzle pressure drop coefficient for each nozzle
l_c	baffle cut in (m)
N	number of rows in a particular element, if it is inter-baffle element it is n_{ecf} and if it is window element it is n_{ew}
N_c	number of elements in inter-baffle region
n_{ecf}	number of rows in inter-baffle element = $\frac{D_s - 2l_c}{x_l N_c}$
n_{ew}	number of rows in window section = $\frac{0.8 l_c - \frac{D_s - D_{otl}}{2}}{\frac{N_w}{2} x_l}$
N_{tw}	number of tubes in window section
N_w	total number of elements in upper and lower window section
P_i	pressure at ith node
p	tube pitch, $p = \frac{p_t}{d_o}$
p_t	tube pitch (m)
Q	volume flow rate in (m^3/s)
Re	Reynolds number, $Re = \frac{\rho Q d_0}{A_{min}\mu}$
S	baffle spacing (m)
x_d	diagonal pitch of tube bundle (m)
x_i	distance between outer most tubes at cross section (Fig. 2(b)) at the end of ith element (m)
x_e	space between the outer most tube in a shell and shell outer diameter (m): $D_s - D_{otl}$
x_l	longitudinal pitch of tube bundle (m)
x_t	transverse pitch of tube bundle (m)
$x(i)$	net distance at the exit of ith element within the tube outer limit (m), $x(i) = \frac{x_i + x_{i-1}}{2}$
μ	viscosity of the fluid (in Pa s)
ρ	density of shell side fluid (kg/m^3)
ψ_m	acute angle the fluid makes with the tube in the mid-section in radians
ψ_w	acute angle the fluid makes with the tube in the window section in radians
ΔP	pressure drop (Pa)
ΔP_c	calculated shell side pressure drop
ΔP_{exp}	experimental pressure drop
ΔP_{fem}	pressure drop predicted by Finite element model
ΔP_m	measured shell side pressure drop
θ_b	angle subtended by the baffle cut

times the calculated values. This clearly indicates that the method of Gaddis and Gnielinski [7] cannot be applied safely in the form suggested by them. Kapale et al. [10] have proposed a theoretical model to calculate the shell side pressure drop. Their model incorporates the effect of pressure drop in inlet and out let nozzles along with the losses in the segments created by baffles. For the range of Reynolds number between 10^3 and 10^5, they found that their results match more closely (deviation between +2.4% and −4%) with the available experimental results. But they have not shown the validity of their model to predict pressure drop in HXs with NTIW. The calculation adopted by Kapale et al. [10] is complex. They have not predicted pressure drop for all the cases for which experimental data is available. Thus, there is a need to develop a simple model to calculate pressure drop on the shell side of STHXs. All the theoretical models reported in literature to calculate the shell-side pressure drop in a STHX require a lot of calculations with a number of variables involved in the calculations. Further these models use different correlations for window section and cross flow section. In all the above mentioned references, the methodology to find pressure drop coefficient involves tedious calculations which include various geometrical parameters and is time consuming. These pressure drop coefficients have been changed time and again, yet no coefficient has been found which works satisfactorily for all cases.



SH Main stream
SL Leakage stream
SB Bypass stream

Fig. 1. Flow through shell of shell and tube heat exchanger with segmental baffle with leakage streams. [7].

Friction factors for flow over rectangular tube banks have been given by Zukauskas [11] and Gunter and Haw [12]. A Finite element model of STHX for determining amount of heat transfer has been developed by Ravikumaur et al. [13] in 1988 but application of such a model to determine pressure drop in STHX has not been carried out so far.

Yonghua et al. [14], experimentally investigated the shell-side thermo-hydraulic performance of a shell and tube HX with trefoil hole baffles under turbulent flow regime. Based on the experimental results, empirical correlations of the Nusselt number and pressure loss as a function of the Reynolds number are obtained. To analyze the mechanisms of these thermo-hydraulic characteristics, numerical computation is carried out. Ender and Ilker [15], investigated the baffle spacing, baffle cut and shell diameter dependencies of the heat transfer coefficient and the pressure drop by numerically modeling a small HX. The authors refer to the Bell-Delaware [5] method as a very detailed and an accurate method to estimate the outlet parameters and have compared their results to that method, but Bell-Delaware method itself does not predict pressure drop values close to the experimental values. The authors have also compared the pressure drop results to Kapale's [10] model and have found a deviation of up to 34%. Results obtained from the CFD simulations show that the existing analytical methods under predict the pressure drop in many cases. Vera et al. [16], present a model to determine the outlet conditions of a shell and tube HX working in a refrigeration cycle either as a condenser or evaporator only. The model does not take the internal geometrical information into