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Beneficial design of unbaffled shell-and-tube heat exchangers for attachment of longitudinal fins with trapezoidal profile

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ABSTRACT

A parametric variation followed with Kern's method of design of extended surface heat exchanger has been made for an unbaffled shell-and-tube heat exchanger problem. For this analysis, the rectangular and trapezoidal fin shapes longitudinally attached to the fin tubes are taken. In comparison with the attachment of trapezoidal fins, it is found that the heat transfer rate was lesser than the rectangular cross section by keeping a constant outer diameter of the shell along with all other constraints of a heat exchanger design, namely, number of passes, tube outer diameter, tube pitch layout, etc. But when the total volume of the fin over a tube was kept constraint, using trapezoidal fins the heat transfer rate is found to be increased and consequently the pressure drop decreases much more than in the case of fins with rectangular cross section. This optimization has shown beneficial results in all the cases of different constraints of heat exchanger design analysis.

1. Introduction

The shell and tube heat exchanger is suited for high pressure applications. This type of heat exchangers consists of a shell with a bundle of tubes inside it. One fluid runs through the tubes and another fluid flows over the tubes to transfer heat between the two fluids. The set of tubes is called a tube bundle, and may be composed of several types of tubes: plain, longitudinally finned, etc. From these categories, tubes with longitudinal fins have an ability to transfer more heat. Shell-and-tube heat exchangers are used in various applications for enhancing rate of heat transfer between two fluids.

The enhancement of heat transfer is critically important in industrial applications such as process cooling, refrigeration, chemical processing, air separation, etc. Fins or extended surfaces play an important role to augment the rate of heat transfer. In situations of combined conduction-convection effects, where the objective is to enhance the rate of heat transfer between a solid and an adjoining fluid, fins are commonly employed. The only way to increase the heat transfer rate in a heat exchanger with a given constant Log Mean Temperature Difference (LMTD) is made by increasing the surface area. Surface area can be increased by a number of ways, using finned surfaces being one of the oldest methods. In applications consisting of fluids (liquids, gases or halogen compounds), it can be mentioned that the heat transfer coefficient on the liquid side is much greater than that of the gas side. Fins are then used on the gas side so that heat transfer rate may be brought to same value on both sides of the boundary separating the two fluids and thus fins bring about equality in resistance to heat transfer. Broadly fins can be classified as those with constant cross section and another being those with varying cross section. It is well understood that as conductive heat transfer rate decreases along the length of the fin, taper fin is the better option for transferring heat effectively.

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Nomenclature

a_s	area of cross section in the shell side
a_t	area of cross section in the tube side
A_0	area defined in Eq. (14)
A_i	area expressed in Eq. (15)
c_s	specific heat capacity of fluid in the shell side
c_t	specific heat capacity of fluid in the tube side
d	thickness of rectangular fin
d_e	equivalent diameter for heat transfer calculations
D	inner diameter of tubes
D_1	outer diameter of tubes
D_2	inner diameter of shell
D_b	tube bundle diameter
De_s	equivalent diameter in the shell side
De_t	equivalent diameter in the tube side
f_s	friction factor in the shell side fluid flow
f_t	friction factor in the tube side fluid flow
h_f	heat transfer coefficient over the fin surface
h_{f_i}	heat transfer coefficient outside surface and fins with respect to the inner surface of tubes
h_i	heat transfer coefficient of inside tube surface
H_f	height of individual fin
G_s	mass velocity of fluid in the shell side
G_t	mass velocity of fluid in the tube side
$I_n(Z)$	modified bessel function of first kind order n and argument z
k_f	thermal conductivity of fin material
K	constant, see Table 1
$K_n(Z)$	modified bessel function of second kind order n and argument z
K_s	thermal conductivity of fluid in the shell side
K_t	thermal conductivity of fluid in the tube side
L	length of a tube
m	defined in Eq. (20)
M	constant used in Eq. (1)
n	number of tube side passes
N_f	number of fins per tube
N_T	total number of tubes
P	perimeter, see Eq. (16)
P_t	tube pitch
P_w	wetted perimeter
Pr_s	prandtl number based on shell side fluid
Pr_t	prandtl number based on tube side fluid
Q	heat transfer rate per unit lmtd
Re_s	reynolds number in shell side fluid flow
Re_t	reynolds number in tube side fluid flow
s_s	specific gravity in shell side fluid
s_t	specific gravity in tube side fluid
U	overall heat transfer coefficient
V	fin volume
w	tube side fluid mass flow rate
W	shell side fluid mass flow rate
y_b	semi-base thickness of trapezoidal fin
y_t	semi-tip thickness of trapezoidal fin
ΔP_s	pressure drop of fluid in the shell side
ΔP_t	pressure drop of fluid in the tube side

Greek letters

η_f	fin efficiency
μ_s	viscosity of shell side fluid
μ_t	viscosity of tube side fluid

The description of shell-and-tube heat exchanger with its tube either finned or bare can be found elaborately in several text books [1–6]. The design of heat exchangers is a fairly complex thing to accomplish mainly owing to the fact that there are many qualitative decisions to be taken along with the quantitative aspects. The process and problem specification is one of the major steps in heat exchanger design. Information is needed on size, weight and other constraints like mass flow rates, inlet temperatures and pressures on both streams, maximum allowed pressure drops on both fluid sides, fluctuations in inlet temperature and pressure and also the environment parameters. Thermal and hydraulic design procedures, surface basic characteristics, surface geometrical properties and thermo physical properties are also taken into consideration. Mechanical design aspects and manufacturing considerations are also of importance here. Therefore, the common attempt is to create optimum design for maximum heat transfer rate with minimum space occupancy with given constraints.

During the heat transfer process in the shell and tube heat exchanger, a lot of constraints come into play in order to achieve maximum feasible rate of heat transfer for a given size of heat exchanger. It has been seen that the increase in fin height in a longitudinal fin, heat transfer area increases but at the same time, the driving force for the motion of the fluid increases. Both these two phenomena act simultaneously and counter to each other which may be a desirable condition. Hence there is an optimum dimension of the fin for a particular arrangement inside a heat exchanger with a number of variable constraints which gives best overall performance of the heat exchanger [4]. In addition, it has already been mentioned that taper fins are better with respect to heat transfer rate per unit fin volume. Due to variable cross section, the flow passage area for fluid flow increases and therefore pressure drop may be decreased in the shell side with adopting taper fins.

In the present study, design performance of longitudinal fins inside a shell and tube heat exchanger has been analyzed using Kern's method which may give easy and reasonably accurate measurement of heat transfer rate and pressure drop. It has been seen that with various constraints such as number of passes, tube outer diameter and tube pitch layout remaining constant, increase in fin height causes heat transfer rate of longitudinal fins with rectangular and trapezoidal profiles to decrease after attaining a maximum value. Upon further considerations with the fin shape, it can be mentioned that heat transfer rate per unit volume is the maximum if the profile of fins were changed to a parabolic one. Unfortunately parabolic