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Experimental investigation of heat transfer enhancement and fluid flow characteristics in a protruded surface heat exchanger tube

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ABSTRACT

This study emphasize on experimental work to enhance the heat transfer by some modification on the surface of the heat exchanger tube. The modification has been made by inserting some protruded shape sheet metal surface within the tube and the surface of sheet metal attached to the surface of the tube. The experiment has been done by using the parameter viz. stream wise spacing $(x/d) = 10, 20, 30, 40$, span wise spacing $(y/d) = 10, 20, 30, 40$ and Reynolds number in the range of 6000–35,000. The experimental results shows a significant enhancement in heat transfer rate and friction factor due to the presence of the protrusion on the heat exchanger tube over the conventional smooth heat exchanger tube. On the basis of experimental results and observations, it was found the enhancement of 3.43 times in heat transfer and 2.31 times enhancement in thermo hydraulic performance factor in the case of stream wise spacing of $(x/d) = 10$ and span wise spacing $(y/d) = 10$ respectively. The statistical correlations for Nusselt number, friction factor and thermo hydraulic performance factor as a function of flow and roughness parameters have been developed with the aid of experimental data collected for different set of roughness parameters. The deviation between predicted and experimental values for Nusselt number, friction factor and thermo hydraulic performance factor are of the order of ±10%, ±15%, and ±12% respectively.

1. Introduction

In modern applications viz. air conditioning, refrigeration system, radiators, and chemical process, the heat exchanger is used to enhance the heat transfer rate from one medium to other. In general, there are two methods of heat transfer enhancement in heat exchanger. First is the active method that requires the extra external power sources such as fluid vibration, jet impingement and injection. The other one is the passive method that requires no other power sources it means that there is no need of any kinds of external forces.

Several investigations have been carried out to study the effect of turbulators for example dimpled or grooved tube, twisted tapes and protruded tube. Afanasyev et al. [1] experimentally studied the heat transfer enhancement mechanism for flows in a dimpled channel with various different shapes. Enhancements in heat transfer was found to be about of 30–40%, with pressure losses that are not increased appreciably relative to a smooth surface are presented. Mahmood et al. [2] indicates that important Nusselt number variations are observed as the array of protrusions is changed with respect to the locations of the dimples. With protrusions, form drag and channel friction are increased. As a result, thermal performance parameters are then generally slightly lower when protrusions and dimples are employed, compared to a channel with a smooth dimple arrangement. Xiao et al. [3] reported that the thermal performance parameters were higher in a heat exchanger with a dimpled bottom and smooth top than in a heat exchanger with a dimpled bottom and protrusions on top in laminar region, and proposed friction factor ratio and Nusselt number ratio correlations for a heat exchanger with dimpled bottom and smooth top. Ligrani et al. [4] discusses flow structure and local Nusselt number variations in a channel with dimples and protrusions on opposite channel walls. Instantaneous flow visualization images and surveys of time-averaged flow structure show that the protrusions result in added vortical, secondary flow structures and flow mixing. As a result, local friction factors and Nusselt numbers are augmented compared to a channel with no protrusions on the top wall. Burgess and Ligrani [5] proposed a Nusselt number ratio correlation as a function of the dimple print diameter and dimple depth in a heat exchanger with dimple on the bottom and a smooth top. Bhushan and Singh [6] studied the influence of Reynolds number on heat transfer coefficient distribution on the surface having staggered array of the protrusion geometry.

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Nomenclature

Re	Reynolds number	Q_u	useful heat transfer (W)
Pr	Prandtl number	h	heat transfer coefficient (W/m^2 K)
$\dot{m}$	mass flow rate (kg/s)	k	thermal conductivity of air (W/m K)
V	air flow velocity (m/s)	e	height of protrusion (mm)
C_p	specific heat (J/kg K)	D	hydraulic diameter of pipe (mm)
d	diameter of protrusion (mm)	T_o	outlet temperature (K)
T_i	inlet temperature (K)	T_{pm}	mean pipe temperature (K)
T_{fm}	mean fluid temperature (K)	A_s	area of pipe (m^2)
A_o	area of orifice plate (m^2)	β	open area ratio
C_d	coefficient of discharge	ρ	density of air (kg/m^3)
μ	dynamic viscosity (N s/m^2)	ΔP_o	pressure drop across orifice plate
ΔP	pressure drop (Pa)	η	thermo hydraulic performance factor
H	head difference (mm)	y/d	span wise spacing
x/d	stream wise spacing		
Nu	Nusselt number		

The enhancement in heat transfer rate was about 2.5 times than smooth surface value over a range of Reynolds number. Terekhov et al. [7] stated that the heat transfer enhancement of the dimple is mainly due to auto oscillations generated by the dimple under turbulent flow regime, which depends on the depth and radius of the dimple. Promvonge [8] studied the thermal performance of a tube with square cross section coiled wire and compared the experimental results with the results obtained from circular cross section wire. The results showed that the square coiled wire insert provides better overall enhancement than the circular one under the same conditions. Elyyan and Tafti [9] numerically investigated the flow characteristics and Nusselt number distribution in a heat exchanger with a dimple bottom and protrusion on top using a large-eddy simulation, and reported that heat transfer augmentation was higher in the turbulent region due to oscillatory flow. Chen et al. [10] numerically studied the flow and heat transfer features in fully developed and found that the heat transfer rate distributes asymmetrically on protrusions when the protrusion height ratio (h/D) is large, the associated enhanced heat transfer is attributed to the asymmetric flow structure and vortex inside the wake behind the protrusion. Mashub and Rahman [11] found that the grooved radiating fin losses approximately 1.23 times greater heat per unit area compared to the threaded fin. As pressure decreases heat loss reduces and contribution of radiation heat transfer on total heat loss increases. Kore and Sane [12] concluded that the dimple surface with uniform heat flux have relatively low heat transfer coefficient on the leading edge of the dimple and high on the trailing edge and the flat area immediately downstream of the dimple. Varun et al. [13] studied the experimental investigation on combination geometries viz. transverse and inclined discrete ribs, which gives better heat transfer enhancement as compared to single roughened geometry. Sahu and Bhagoria [14] experimentally investigated the effect of 90̊ broken ribs as roughness elements and found that thermal efficiency lies in between 51% and 83.5%. Karwa et al. [15] investigated the effect of rib chamfered angle, duct aspect ratio on heat transfer and friction factor using integral chamfered ribs. The experimental data show that the chamfered angle of 15° gives highest Nusselt number as well as friction factor. Chyu et al. [16] studied the pin fin array with the stream wise and transverse spacing of 2.5 and achieved a heat transfer enhancement of 2–4 times and an increase of 20–30 times of friction factor over the range of Reynolds number 10,000–100,000. Similarly some more work which have been carried out in this field includes, Barik et al. [17], Rao et al. [18], Mahmood and Ligrani [19], Liu et al. [20], Chang et al. [21,22], Turnow et al. [23], Ligrani et al. [24], Kumar [25] and Singh et al. [26] who worked on different surface protrusions, spherical and Tear drop dimple, Dimple channel with different aspect ratios, hemispherical Protrusion and Dimple, Concave and Convex dimple, Surface dimple and staggered arrangement and dimple–protrusion on opposite wall as turbulence promoters in their study.

This paper presents an experimental study of fluid flow and heat transfer enhancement in protruded surface heat exchanger tube. The main aim of the present study is to enhance the heat transfer rate and thermo hydraulic performance factor by using the protruded surface heat exchanger tube for Reynolds number range of 6000–35,000. Using the experimental results, correlations for heat transfer, friction factor and thermo hydraulic performance factor are also developed. These correlations has been developed with the help of previous work [25–28] such as, Hans et al. [27], Sethi et al. [28], Singh et al. [29], and Yadav et al. [30], who developed statistical correlations for heat transfer and friction factor as a function of geometrical and flow parameters. On the basis of literature and previous studies protruded surface with different spacing in flow and alternate direction is taken as insert geometries.

2. Experimental setup

The experimental setup is associated with an open loop flow system that has been fabricated and designed to conduct an experiment to obtain the heat transfer and fluid flow characteristics in a protruded surface heat exchanger tube.

The schematic diagram of the experimental setup is shown in Fig. 1. It consists of test tube together with exit and entrance sections, a control valve, a calibrated orifice plate, a suction blower and various instruments has been used for the measurement of temperature and pressure drop. The blower sucks the atmospheric air through the heat exchanger tube. The flow through the tube was managed by means of control valve provided near the blower. The mass flow rate of air was measured with the help of an orifice plate on the suction side and is connected to an inclined manometer.

2.1. System descriptions

The values of flow and roughness parameters for this study are listed in Table 1. The schematic of the roughness geometry used in the present investigation is shown in Fig. 2 and the pictorial view of roughness geometry is depicted in Fig. 3. Fig. 4 shows the