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Experimental and numerical study of fluid flow and heat transfer in an annulus of inner twisted square duct and outer circular pipe

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A B S T R A C T

Heat transfer and friction factor characteristics of air flow in an annulus formed by an inner twisted square duct and an outer circular pipe is studied experimentally for Reynolds number range of 400–60,000. Experiments were conducted with air as working fluid. A uniform wall temperature at the inner wall of annulus was maintained while the outer pipe was kept insulated. Twist ratios (ratio of the pitch to outer width of twisted square duct) of 10.6 and 15 were used in the experiments. The transitional Reynolds number for laminar flow to turbulent flow was identified as 3000. Results were compared with the flow inside the annulus of an equivalent straight square duct in a circular pipe for identical pumping power. Results showed considerable enhancement in the heat transfer and pressure drop in both the laminar and turbulent flow regimes. The influence of the annulus parameter on the friction factor and heat transfer was also studied by varying the outer pipe diameter. At a given twist ratio, higher values of the friction factor and the Nusselt number were observed for smaller annulus parameters (ratio of cross section area of square twisted duct to cross section area of circular pipe). In order to extend the experimental results to a larger range of fluids, a numerical analysis was undertaken for fully developed flow inside the annulus for the experimental twist ratios. The parameters range covered in the simulations were Reynolds numbers of 100–100,000 and Prandtl numbers of 0.7–20. The local velocity distribution across the annulus cross-section were analysed. Empirical correlations for friction factor and Nusselt number are provided separately for laminar and turbulent regimes. Guidelines for selection of twisted square duct are provided in terms of Reynolds number and enhancement factor. The results are significant because it will contribute to the development of compact double pipe heat exchangers.

1. Introduction

Heat transfer enhancement techniques can be classified as passive, which require no direct application of external power, or as active, which require external power. The effectiveness of both types of techniques is strongly dependent on the mode of heat transfer. The passive techniques use surface or geometrical modification to the flow channel by incorporating inserts or other devices. They increase heat transfer coefficients by disturbing existing flow behaviour (except for extended surfaces) which also leads to increases in pressure drop. Some of common methods for heat transfer enhancement by passive techniques are achieved by using [6];

(i) Treated surfaces: In these methods fine scale alteration is made to surface finish or coating which may be continuous or discontinuous. These methods are used in boiling and condensation applications.
(ii) Rough surfaces: These are the surface modifications done to promote turbulence in the flow field near the wall region. These methods are used for single phase fluids, mostly in turbulent flow regime. Heat transfer area remains unaffected even after surface modification.
(iii) Extended surfaces: Extended surface include small surfaces of various shapes, or perforations of on primary heat transfer surface. They are used in heat transfer enhancement of single phase flow, condensation and boiling.
(iv) Coiled tubes: It produces secondary flows and vortices which promote higher heat transfer coefficients in single phase flows as well as in most regions of boiling.
(v) Swirl flow devices: They generate and superimpose swirl flow (secondary flow) on axial flow in a channel. These

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devices include twisted tape, twisted duct and screw inserts. They find application for single phase and two phase fluid.

Twisted ducts work on the principle of passive heat transfer enhancement technique. Twisted ducts are used as a replacement for straight circular tube in shell and tube heat exchangers. Twisted ducts [1] not only enhance heat transfer for the fluid inside the ducts [2] but also for the fluid on the shell side. It results in increase in the heat transfer coefficients for both shell side and tube side fluids. Hence size of heat exchangers get reduced resulting in compact heat exchangers (surface area >700 m^2 per unit volume of heat exchanger). The manufacturing cost of industrial [10] shell and tube heat exchanger is only justified if the area of heat exchangers is above 15–20 m^2. Double pipe heat exchanger is a preferred choice for heat exchangers with area less than 15 m^2. The significant reduction in the area of double pipe heat exchanger can be made if it is constructed with twisted duct instead of normal straight duct [1]. In the present study, double pipe heat exchangers were constructed with a square twisted duct inserted in a circular pipe. Heat transfer and fluid flow characteristics were studied for flow inside the annulus formed by an inner twisted square tube and an outer circular pipe. It was expected that the twisted surface of the duct will improve mixing in the fluid in the annulus and enhance the heat transfer.

In the past, researchers numerically and experimentally investigated heat transfer and fluid flow characteristics of flow through a concentric annulus. Few have also analysed flow past twisted tube bundle in shell and tube heat exchangers. No information on the double pipe heat exchangers with a twisted square duct is available in the literature. Dzyubenko [3] developed a numerical model for longitudinal flow over elliptically twisted tube bundles. In his study, a bundle of twisted tubes were replaced by a porous mass with a diameter equal to the tube diameter. The transport equations were solved for developed flow to calculate the friction factor values. The validation of data was carried out through experimental results with air as working fluid.

Ievlev et al. [8] presented the results of an experimental study on heat transfer and hydraulic resistance over the bundles of elliptically twisted tubes in longitudinal and cross flow using air as the working fluid. The twist ratios used were in the range of 6.5–35. The Reynolds number range covered was 3000–10,000. The study showed that twisted tubes permit an appreciable increase in the heat transfer and a substantial reduction of the heat exchanger dimensions for the range of parameters studied. Dzyubenko and Dreister [4] experimentally analysed the results of heat transfer and friction factor in bundles of twisted tubes and rods with spiral wire wrap spacers. Recommendations were given for calculating the heat transfer coefficient in heat exchanger using twisted tubes. A correlation was provided for calculating Nusselt number and friction factor for flow over a bundle of 19 tubes. In their study, the twist ratio range considered was 4.2–12.4 and the Reynolds number range was 3000–10,000. Gnielinski [5] analysed various correlations available in the literature for turbulent flow in concentric annuli. The effect of the diameter ratio (ratio of inner tube diameter to outer tube diameter) on the velocity profile in an annulus was explained. By evaluating a large number of experimental data from the literature, correlations for the friction factor and heat transfer were presented. Van Zyl et al. [12] experimentally analysed the flow and heat transfer characteristics of the flow of water through an annulus of concentric tubes. The effect of the diameter ratio was studied in the turbulent regime (*Re* range of 10,000–45,000). They concluded that, for the same Reynolds number, the friction factor and heat transfer increase with the increased with the increase in diameter ratio.

The literature review suggests that heat transfer and flow resistance of fluid flow over twisted tube bundles of elliptical cross section have been studied. In particular, flows over elliptical tubes were dealt with for a *Re* range of 3000–10,000 with air and water as working fluids and for twist ratios in the range of 4–35. However, other cross sections have received much less attention specifically regarding experimental or numerical data on double pipe heat exchangers with square twisted duct.

In the present study, an effort is made to experimentally and numerically investigate the pressure drop and heat transfer in an annulus consisting of an inner twisted square duct and an outer circular pipe. The following are the objectives of the present study:

- To measure the pressure drop and heat transfer coefficient in an annulus for a wide range of *Re* and twist ratios. The parameter ranges covered here were twist ratios of 10.6 and 15 for a Reynolds number range of 400–60,000 with air as the working medium.
- To examine the influence of the annulus gap on the friction factor and heat transfer. Measurements were done for both the twist ratios of 10.6 and 15.
- To extend the results for larger range of Reynolds number and Prandtl number using a numerical approach. The Reynolds number range for numerical study was 100–1,00,000 with a Prandtl number range of 0.7–20.

The heat transfer enhancement mechanism is explained by examining the local velocity distribution in the flow cross section. A general correlation to calculate the average friction factor and average Nusselt number for the entire range of parameters studied is provided. This serves as a guideline for the selection of square twisted ducts for double pipe heat exchangers.

2. Terminologies employed

Common terminologies associated with twisted ducts (highlighted in Fig. 1) are.

- Pitch (*S*): The distance between two consecutive points along the length of twisted square tube, where the orientation of tube cross section exactly coincide with each other. The cross section rotates 360° along one pitch distance.
- Twist Ratio (*M*): The geometrical parameter used to describe twisted ducts. It is defined as the ratio of the pitch (*S*) to the outer characteristic length of twisted square duct (*Do*) of the duct as given by Eq. (1)

$$M = \frac{S}{Do} \tag{1}$$

- Annulus Parameter (*AP*): Defined as ratio of the cross section area of the square twisted duct to the cross section area of the shell and is given by Eq. (2).

$$AP = \frac{4\,Do^2}{\pi\,di^2} \tag{2}$$