Testing of Twisted-Tube Exchangers in Transition Flow Regime

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Abstract

In most cases the main barrier to a heat flow in shell-and-tube heat exchangers is the convective resistance on the inner and outer tube surface. To increase thermal effectiveness, a number of augmentative techniques are available: twisted tapes, wire coil inserts, internal or external fins are frequently used. Following those efforts, it is proposed to use twisted tubes, instead of cylindrical tubes.

This paper describes testing of an advanced shell-and-tube design with twisted tubes. Its performance has been tested under single-phase, fully developed, transition flow conditions. Subject to design constraints and based on specific optimization requirements, several configurations were tested, each test examining heat transfer and pressure drop for the particular heat exchanger. It was shown that overall heat transfer and pressure drop increase with a smaller tube twist pitch to diameter ratio, and that these exchangers have specific advantages and characteristics, previously open only to plate exchanger users. Twisted tubes increase the level of mixing and promote turbulence in the low Reynolds number range, on both, tube-side and shell-side of the tubes.

Testing over a wide range of geometric parameters and process conditions confirmed that heat transfer results for twisted-tube exchangers could be satisfactorily correlated in the Nusselt, Reynolds, Prandtl equation form. Total friction factor through a twisted-tube bundle was found to be the sum of axial friction loss component and a drag contribution from the swirl flow.

Introduction

Studies of the heat transfer and pressure drop of twisted-tube type exchangers were conducted, and the effects of changing various construction parameters were investigated. Experimental heat transfer and pressure loss data were obtained using the test rig with three different, commercial-size test exchangers. Seven different bundle configurations were tested, each design examining specific geometric parameters of the twisted-tube exchangers.

These studies overwhelmingly confirmed the merits of twisted-tube exchangers and it is now possible to provide accurate guidance on performance rating and sizing of such units. Also, it was demonstrated that these exchangers could be tuned to a particular application by optimizing design of the tube profile.

Background

Economics of the chemical and process industry is strongly related to the cost and size of operated heat transfer equipment. Consequently, considerable effort has been expended over the years to develop tubes of enhanced heat transfer characteristics. One of the promising, emerging techniques uses twisted tubes, instead of cylindrical tubes. From a thermal point of view these tubes not only promote turbulence in the low Reynolds number range, but also have larger heat transfer area per unit volume of the heat exchanger than the cylindrical bundles.

Limited studies available in open literature frequently suffered from limitations that became obvious by the results of this test program. For example, a simplified approach assumed that twisted-tube profiles do not affect the shell-side heat transfer coefficient, and as a result all benefits were attributed only to the tube-side heat transfer augmentation. This is a gross oversimplification of the real situation because the optimal tube design results in a substantial heat transfer increase over the plane tube value. However, if not optimized, a badly chosen combination can have an adverse effect. In addition, if considered independently for shell-side and tube-side flow, the ideal combination within tube design variables can in some cases be contradictory.

A schematic drawing of a typical tube of the type tested is shown on Fig. 1. Geometry of the device is simple: tube twisted into a constant pitch helix.

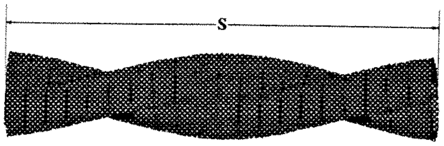

Fig. 1 Twisted Tube Geometry

Experimental Testing

The experimental testing of the prototype exchangers was performed at the Heat Transfer Research, Inc. (HTRI) Research Facility in an environment similar to the commercial service. Obtained data provided the basis for this analysis. Objectives of testing thermal-hydraulic performance, mechanical reliability, and manufacturing complexity were to,

- Maximize heat transfer rate
- Minimize heat exchanger volume
- Minimize pumping requirements

The two-year program was performed with the intent to examine one or a combination of the following tasks:

- Obtain heat transfer and pressure drop data for fluid flow inside and outside twisted-tubes
- Evaluate the magnitude of the bypassing stream in the conventional heat exchanger design
- Evaluate the performance improvement of the sealed heat exchanger design
- Evaluate the variation in the performance of the heat exchanger with different twist pitch to diameter ratios
- Evaluate the performance of the nested, non-shrouded heat exchanger design
- Evaluate the performance of the low-finned twisted-tubes, shrouded heat exchanger design
- Develop empirical equations and define the criteria which will facilitate the selection of configuration to suit the desired performance
- Develop generalized correlations which would indicate the probable performance of other scaled variations of the tube and exchanger geometry
- Define criteria for an economic evaluation of the exchanger performance in terms of increased heat transfer versus pressure drop

Experimental Setup

The experimental facility, a closed loop system, included three twisted-tube exchangers in series. One exchanger operated as a heater, the second as a cooler while the third exchanger operated as an inter-changer. The test fluid circulated on the shell-side in all exchangers, and the service fluids, condensing steam and cooling water, circulated through the tubes.

Instrumentation was provided for measurement of all pertinent data on test fluid, steam, condensate, and cooling water. Flow rates, temperatures, pressures and differential pressures were measured. All the instruments were calibrated prior to operation, and all experimental runs were systematically checked for the heat balance. Generally, the heat balances were in the range of $< \pm 10\%$ accuracy.

Swirl Flow Parameter

The presence of centrifugal forces in the swirl flow may influence the flow field similar to the way temperature stratification occurs in the gravity field. Depending on the velocity distribution in the radial direction, fluid particle's kinetic energy can be in some cases transformed to potential energy leading to dynamic stratification. The dynamic stratification could be described as formation of layers in the field of centrifugal forces. These layers in a swirl flow can be classified in two major categories: one, where centrifugal forces have conservative character, and in the other, where they have active character.

The action of active centrifugal forces promotes formation of random pulsation and increases the average flow turbulence, while in the zone of conservative forces the fluid viscosity damps most embryos of turbulence. As a consequence, if properly designed, a swirl flow device will have a geometry that produces active centrifugal force flow field. This will in turn promote the radial velocity change in a way that has positive effects on the heat and mass transfer processes. To allow optimization, a unique, twisted-tube swirl flow parameter, Φ, was defined.

Heat Transfer Results

The test objectives were to identify the phenomenological behavior, understand the involved mechanisms, and develop design correlations that have general capabilities. General correlations for the heat transfer coefficient in swirl flow used in the analysis of twisted-tube heat exchangers have the following form,

$$N_u = f_1\left(R_e, P_r, \Phi\right) \quad \text{.................................} \quad (1)$$

The correlation model which best represented twisted tube heat transfer data for different construction configurations was developed using the well-known Wilson plot technique. Predictive equations were developed that have very good general applicability and cut through a wide spectrum of operating conditions, including different fluids and heating and cooling duties. The beginning of transition flow occurred at the Reynolds number of about Re = 2,000. Using the correlation form,

$$N_u = C_h \cdot R_e^{m_1} \cdot Pr^{m_2} \cdot \Phi^{m_3} \cdot \left(\frac{\mu_b}{\mu_w}\right)^{m_4} \quad \ldots\ldots\ldots\ldots\ldots (2)$$

the constant C_h (C_c), Reynolds number exponent, m_1, and swirl parameter exponent, m_3, were determined. The correlation constants C_h, for heating the shell-side fluid and C_c, for cooling the shell-side fluid, were found to be different. The Prandtl number exponent, m_2, and viscosity correction exponent, m_4, were accepted as 0.4 and 0.14 respectively.

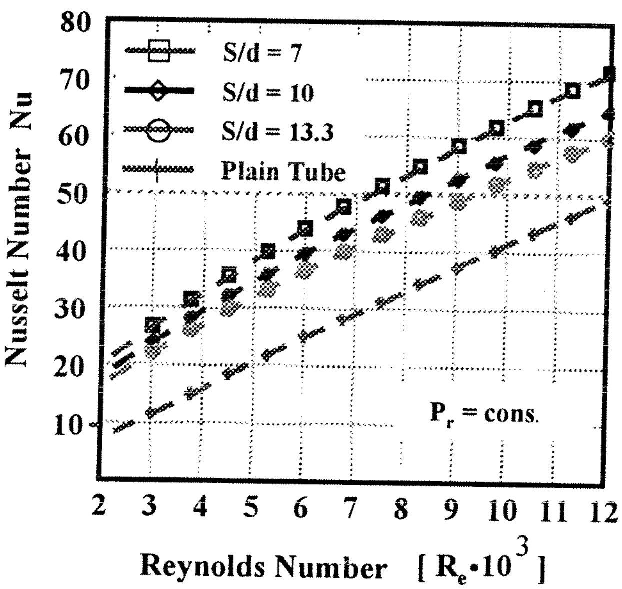

Fig. 2 Nusselt-Reynolds Number Dependence

Because of the predominantly longitudinal flow in the shell-side flow field, leakage between the shroud-to-shell and bundle-to-shroud were primary geometric parameters affecting constants C_h (C_c). Examples of this analysis for cooling a chosen shell-side fluid and for different twist pitch to hydraulic diameter ratios are given in Fig.2. The graph depicts Nusselt number as a function of Reynolds number for three different twist-pitch to diameter ratios. Shown on the same graph are Nusselt number values for a plain tube, based on the proration of the laminar and turbulent Nusselt numbers evaluated at Re=2,000 and Re=10,000. In regard to the relative magnitude of cross-flow entrance and exit regions, which were evident in shorter bundles, it was assumed that bundle length-to-diameter ratio L/D did not affect C_h (C_c) values. Because the build-up to a fully developed profile is continuously disturbed by the pattern of twisted tubes, no influence from the entrance was recognized.

To suggest how the shell-side high heat transfer coefficients in twisted-tube exchangers are achieved, the helical channels can be looked upon as series of consecutive short sections between which the build-up of a shell-side fluid steady velocity profile is interrupted by the constant direction change of the flow. Good transverse mixing is achieved by these interruptions, and the numerous disturbances keep the flow turbulent even at relatively low Reynolds numbers.

If the cold fluid is on the shell side, as a result of centrifugal force, the radial pressure gradient will force heavier, colder fluid outwards, with a consequent rise of hot fluid toward the center of the pipe. This has a negative effect on the local heat transfer coefficient. The opposite will happen for hot fluid on the shell side. However, the flow mal-distribution through the bundle is another factor influencing overall heat transfer, and data show that cooling shell-side fluid (with favorable density gradient) results in a less favorable flow distribution, notably lowering overall heat transfer coefficient.

For the tube-side flow, there are several equally important mechanisms influencing heat transfer in an individual twisted tube. The induced swirl in each tube will enhance turbulence mixing process, and enhance or suppress other mechanisms. The swirling flow inside the twisted tube in general is characterized by the increase in the average flow velocity, increase in velocity gradient at the tube wall, additional flow turbulence due to centrifugal force, and the formation of a local secondary flow.

Analytical correlations for heat transfer coefficients were developed, and will be used for new equipment design. Maximum variation between measured and calculated heat transfer coefficients was < ±10%.

Pressure Drop Results

General correlations for the friction factor in swirl flow used in the analysis of twisted-tube heat exchangers have the following form,

$$f = f_2\left(R_e, \Phi\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

The conducted tests confirmed that the friction factor is nearly independent of the heat flux and its direction. The most important parameter influencing the pressure drop in an individual twisted tube and in a bundle is the twist pitch to hydraulic diameter ratio. A comparison of friction factors in a twisted tube and plain tube bundles for the longitudinal shell-side flow is shown on Fig. 3.

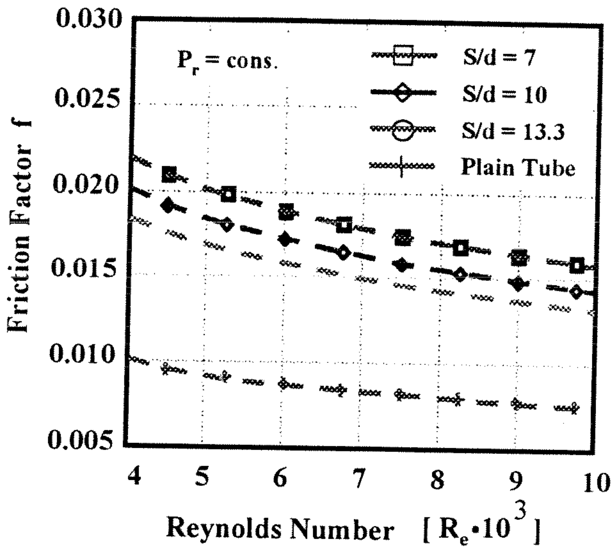

Fig. 3 Friction Factor-Reynolds Number Dependence

The plain tube values were calculated using the Round's [2] correlation, Eq. (4), valid for the transition regime and Reynolds numbers Re = 4,000 –10,000.

$$f = \left[\frac{1}{1.563 \cdot \ln \left[\dfrac{R_e}{0.135 \cdot R_e \cdot \dfrac{\varepsilon}{d_{eq}} + 6.5} \right]} \right]^2 \qquad \ldots\ldots\ldots\ldots(4)$$

It seems that pressure drop is not affected by the density gradient imposed by the heat flux through the wall. This implies that, while momentum exchange may increase with the change of the density gradient, the overall momentum loss in axial direction does not increase proportionally.

An explanation for this may be argued by examining reported experiments on axially rotating tubes. By flow visualization, White [4] showed that, turbulence normally present over the entire cross section is suppressed by rotation over the part of the tube. His measurements of the pressure drop showed a reduction of pressure drop with the flow rotation. At the highest rotational speeds, this effect amounted to a reduction of the frictional coefficient of 60% of its value for the same non-rotating tube.

White's model could be used to qualitatively describe the flow inside the twisted tube. The axial pressure gradient along the tube forces the flow through the tube. As a result of the curvature of spiral channels, the twisted tube generates rotation of the flow within the channels and of the core flow. The core flow is basically in a stable, solid body rotation. In the region between the core flow and the spiral channel flow, interchange of angular momentum occurs, resulting in the decrease of the angular momentum in the channels. This is the primary cause of instability that increases radially with inward heat flux and decreases when the direction of the heat flux is outward. Since most of the resistance to heat flow is in the laminar sub-layer, this instability enhances the turbulent exchange near the wall, resulting in improved heat transfer. The imposed core flow rotation has a positive effect on the axial momentum loss, as White's experiment suggests. Thus, the friction factor does not increase proportionally to the heat transfer coefficient increase.

Discussion

Performed tests indicate that the mechanisms of heat transfer enhancement and turbulence promotion in twisted-tube exchangers appears most effective in the transition regime. Therefore, it is reasonable to suggest that the distortion of the temperature and viscosity profile across the tube and the forward moving of the tube oval profile account for the observed effect. The decrease in tube cross section area contributes also to a fraction of the observed increase in heat transfer coefficient. The extent, to which in non-isothermal laminar and transition flow of highly viscous fluids, the twisted tube spiraling channels impose a different velocity and temperature gradients near the wall, will have to be resolved by a more detailed investigation. Additional experiments in different twist pitch configuration, based on flow visualization and numerical modeling, in laminar and transition flow regime are planned. Coupled perhaps with velocity and temperature profiles across the tube, these experiments will allow further twisted tube and bundle optimization.

Increased turbulence in twisted-tube bundles improves the heat transfer through the reduction of the boundary layer thickness and through the better mixing in the bulk. For laminar and transition flow, the dominant thermal resistance is not limited to a thin boundary layer adjacent to the wall. Therefore, twisted tubes are more efficient in low Reynolds number flow regimes. For turbulent flow, it is more effective to mix the flow in the viscous boundary layer at the wall than to mix the bulk flow. This is because the dominant thermal resistance is very close to the wall. Integral roughness in combination with swirl flow can generally provide a higher efficiency than is provided by the twisted tube alone.

In addition, pure counter-current flow in twisted-tube bundles allows better use of the available temperature difference, and since local velocity field and flow distribution are good, the heat transfer is more uniform, and the shell-side heat transfer coefficient is generally the highest possible in tubular heat exchangers.

Conclusion

Heat transfer and pressure drop data have been obtained on various configurations. It was shown that, both, overall heat transfer and pressure drop, increase with a smaller tube twist pitch to diameter ratio. Extensive testing over a wide range of geometric parameters and process conditions confirmed that, when coupled with a proper swirl flow parameter, heat transfer results for twisted-tube exchangers could be satisfactorily correlated in the Nusselt, Reynolds, Prandtl equation form. The variable correlation constants C_h and C_c appearing in the heat transfer model were found to be a function of leakage bundle-to-shroud area, and twisted tube pitch to diameter ratio. Total friction factor through a twisted-tube bundle was found to be the sum of axial friction loss component and a drag contribution from the swirl flow.

Nomenclature

C_h	Correlation constant for heating, dimensionless
C_c	Correlation constant for cooling, dimensionless
d	Nominal tube diameter, m
d_{eq}	Equivalent tube diameter, m
f	Friction factor, dimensionless
m_1-m_4	Exponents, dimensionless
N_u	Nusselt number, $h \cdot d_{eq}/k$, dimensionless
P_r	Prandtl number, , $\mu \cdot C_p/k$, dimensionless
R_e	Reynolds number, $v \cdot d_{eq}/\nu$, dimensionless
S	Twist pitch, m

Greek symbol

ε	Pipe roughness, dimensionless
Φ	Swirl flow parameter, dimensionless
μ_b	Fluid viscosity in the bulk, $Pa \cdot s$
μ_w	Fluid viscosity at the wall, $Pa \cdot s$

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