



Pressure loss and heat transfer for single phase flow in tubes containing twisted tape inserts

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PRESSURE LOSS AND HEAT TRANSFER FOR SINGLE PHASE FLOW IN TUBES CONTAINING TWISTED TAPE INSERTS

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PRESSURE LOSS AND HEAT TRANSFER FOR SINGLE PHASE FLOW IN TUBES CONTAINING TWISTED TAPE INSERTS

1. NOTATION AND UNITS

		Units	
		SI	British
A, B, C	constants in fluid viscosity – temperature relationship (Equations (8.2) and (8.3))	–	–
A_c	axial flow cross-sectional area = $\pi D^2/4 - \delta D$	m ²	ft ²
A_o	empty* tube flow cross-section area = $\pi D^2/4$	m ²	ft ²
c_p	specific heat capacity	J/kg K	Btu/lb °F
D	tube inside diameter	m	ft
D_{ext}	tube outside diameter	m	ft
D_h	tube hydraulic diameter	m	ft
f	Fanning friction factor = $\Delta p(D/L)\rho g_c/2\dot{m}^2 = \Delta p(D/L)g_c/2\rho v_o^2$	–	–
g	acceleration due to gravity	m/s ² (9.81)	ft/h ² (4.17×10 ⁸)
g_c	constant of proportionality in Newton's second law of motion	– (1.0)	lb ft/lbf h ² (4.17×10 ⁸)
Gr	Grashof number = $g\rho^2 D^3 \beta \Delta T_w / \eta^2$	–	–
Gz	Graetz number = $\dot{M} c_p / (\lambda L)$	–	–
H	180° twist pitch	m	ft
L	axial length	m	ft
m	exponent of viscosity ratio correction factor for liquid flows	–	–
$\dot{m}$	mass flux	kg/s m ²	lb/h ft ²
$\dot{M}$	mass flow rate	kg/s	lb/h
n	exponent of temperature ratio correction factor for gas flows	–	–
Nu	Nusselt number = aD/λ	–	–
Δp	pressure drop	N/m ² (Pa)	lbf/ft ²
Pr	Prandtl number = $\eta c_n / \lambda$	–	–

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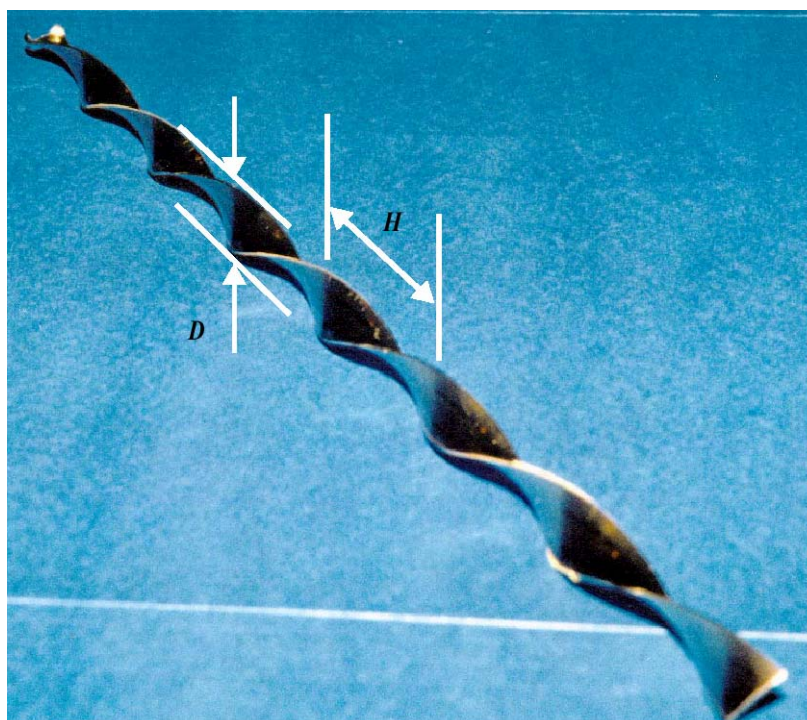
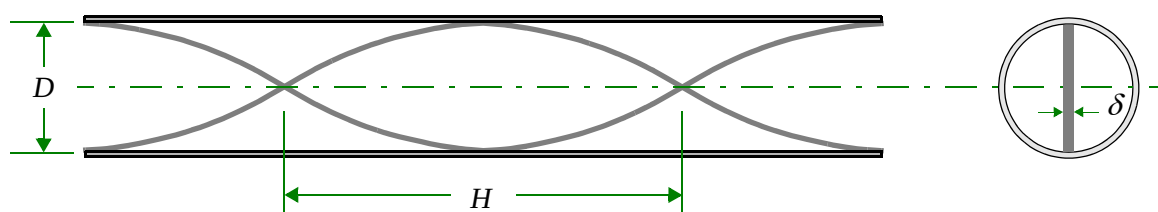
r_f	fouling resistance	$\text{m}^2\text{K/W}$	$\text{ft}^2\text{h }^\circ\text{F/Btu}$
Ra	Rayleigh number = $Gr \ Pr$	—	—
Re_o	empty tube Reynolds number, <i>i.e.</i> in the absence of a tape	—	—
Re_{ax}	Reynolds number based on axial velocity = $\rho v_a D / \eta$	—	—
Re_{Sw}	Reynolds number based on swirl velocity = $\rho v_s D / \eta$	—	—
Sw	swirl parameter = $Re_{Sw} / \sqrt{y}$	—	—
T	absolute temperature	K	$^\circ\text{R}$
ΔT_w	wall-to-bulk fluid temperature difference = $T_w - T_b$	K or $^\circ\text{C}$	$^\circ\text{F}$
v_a	mean axial velocity = $\dot{M} / (\rho A_c)$	m/s	ft/h
v_s	swirl velocity at tube wall = $v_a [1 + (\pi/2y)^2]^{1/2}$	m/s	ft/h
U'	combined heat transfer coefficient, referred to inside diameter of tube, for service fluid, tube wall and any fouling	$\text{W/m}^2\text{K}$	$\text{Btu/ft}^2\text{h}^\circ\text{F}$
y	twist ratio = H/D	—	—
α	heat transfer coefficient	$\text{W/m}^2\text{K}$	$\text{Btu/ft}^2\text{h}^\circ\text{F}$
β	coefficient of volumetric expansion	$1/\text{K}$	$1/^\circ\text{F}$
δ	thickness of twisted tape	m	ft
η	dynamic viscosity of fluid	kg/m s	lb/ft h
λ	thermal conductivity of fluid	W/m K	$\text{Btu/ft h}^\circ\text{F}$
λ_{wall}	thermal conductivity of tube wall	W/m K	$\text{Btu/ft h}^\circ\text{F}$
ρ	fluid density	kg/m^3	lb/ft^3
ϕ	correction factor for variable fluid properties	—	—

Additional subscripts

b	bulk fluid value
ext	conditions at tube outside surface
h	based on hydraulic diameter, D_h
$high$	higher of fluid inlet or outlet temperatures
in	inlet
lam	laminar flow
low	lower of fluid inlet or outlet temperatures

<i>mean</i>	mean value
<i>o</i>	applies to tube with no insert
<i>out</i>	outlet
<i>s</i>	service fluid
<i>Sw</i>	swirl flow condition
<i>trans</i>	transition flow
<i>turb</i>	turbulent flow
<i>w</i>	conditions at tube inside surface

* The term 'empty' in this Data Item refers to the tube under consideration with no insert present.



Sketch 1.1 Twisted tape geometry

2. INTRODUCTION

A variety of methods are available to enhance forced convective heat transfer of fluids flowing inside tubes. One such method is the use of tube inserts such as twisted tapes. Compared with a plain tube of the same internal diameter, tubes with these devices give a greater heat transfer coefficient for a similar pressure drop¹⁸. Tube inserts are inexpensive to manufacture and may be readily introduced into existing tubular heat exchangers. Consequently they are suitable for use in retrofit situations where tubeside performance needs to be upgraded. Additionally, easy removal can facilitate tubeside cleaning. In new equipment use of inserts should result in a more compact heat exchanger, in a suitable application.

Twisted tape inserts are particularly useful when the tubeside fluid is in laminar flow. Here the enhancement for a prescribed allowable pressure drop can be considerable. They have also been shown to be effective in turbulent flow¹⁸.

Applications of twisted tape inserts include

- (i) heating and cooling of viscous liquids where the flow is laminar and the tubeside provides the controlling resistance to heat transfer,
- (ii) the flue tubes of fire-tube boilers,
- (iii) high temperature waste heat recuperators,
- (iv) air-to-water heat exchangers in air-conditioning systems.

Twisted tape inserts may also be used with boiling flows to ensure liquid contacts the tube wall around its entire circumference, thereby delaying the onset of dryout. This two-phase flow application is, however, outside the scope of this Data Item.

A computer program, ENTRANS, is provided in ESDU 97007²⁰ for the analysis of the benefits of several types of tube insert in shell-and-tube exchangers. These insert types are twisted tapes, wire matrix, helically-coiled, static mixers and core tubes.

3. PURPOSE AND SCOPE

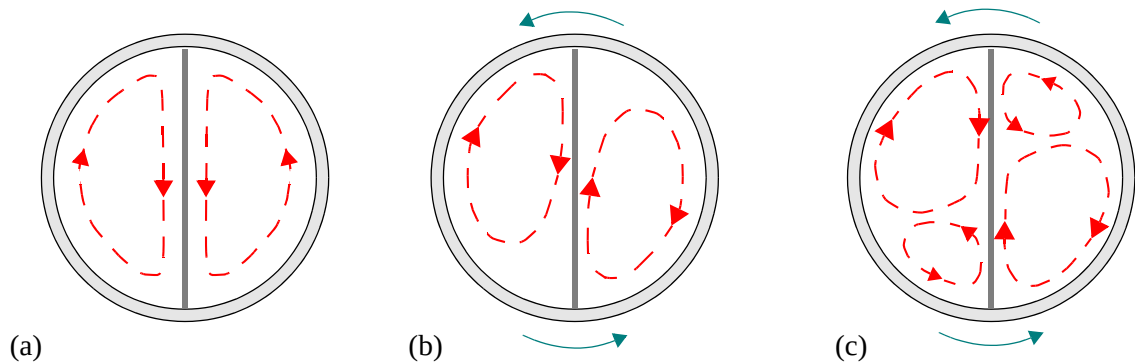
This Data Item is concerned with the flow of a single-phase fluid in a horizontal tube of circular cross-section containing a plain twisted tape insert. Correlations are presented for predicting the pressure drop and mean heat transfer coefficient of a gas or Newtonian liquid flowing through a horizontal tube of specified length. The correlations embrace laminar, transition and turbulent flows.

4. GENERAL CONSIDERATIONS RELEVANT TO CONVECTIVE HEAT TRANSFER IN TUBES WITH TWISTED TAPE INSERTS

4.1 Causes of Heat Transfer Enhancement

- (i) Increased flow velocity results from the small reduction in net free cross-sectional area.
- (ii) The flow cross-sectional area is that of a segment tending to a semi-circle, resulting in a reduced hydraulic diameter.

- (iii) Swirl flow mixing arises from secondary flows generated by the tape twist.
- (iv) Increased flow length, because the helical flow path follows the twist of the tape.
- (v) The tape has a fin effect which is significant only if the tape is metallic, of high thermal conductivity and bonded (brazed, for example) to the tube.



Sketch 4.1 Secondary flow patterns due to free convection and swirl flow (after Manglik and Bergles¹⁶).

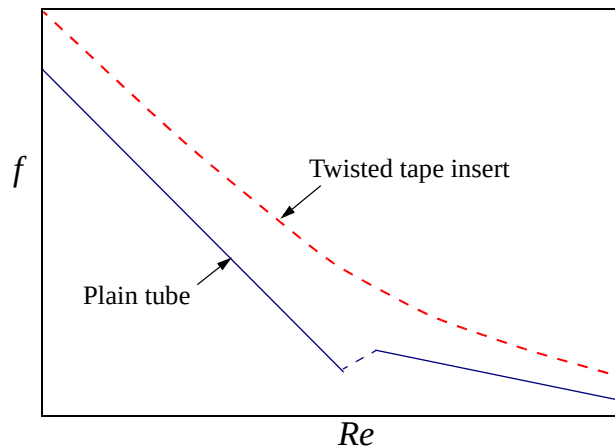
- (a) Buoyancy driven free convection.
- (b) Tape twist at low Reynolds number or moderate twist ratio.
- (c) Tape twist at high Reynolds number or small twist ratio.

External arrows show tape twist in direction of flow.

4.2 Flow Regions

Manglik and Bergles^{16,17} identified four flow regions, as follows.

- (1) At very low Reynolds numbers, less than about 300, enhanced heat transfer results from flow blockage and the longer flow path, together with the thermally-developing boundary layer and free convection superimposed secondary flow (Sketch 4.1a). Swirl effects are minimal, or even absent.
- (2) With increasing Reynolds numbers swirl flow is superimposed on the axial flow whilst any free convection effects are suppressed (Sketches 4.1b and 4.1c).
- (3) Swirl-turbulent flow transition. This region is characterised by the onset of turbulent fluctuations opposed by the centrifugal forces. The net effect is that the discontinuity in the friction factor – Reynolds number curve, characteristic of plain tubes, no longer occurs (see Sketch 4.2). The complex nature of this transition is such that its onset cannot be described solely by a Reynolds number.
- (4) Fully developed turbulent swirl flow corresponding to Reynolds numbers greater than 10 000. Enhanced heat transfer is now the result of flow blockage, the longer flow path and swirl-induced vortex mixing.



Sketch 4.2 Friction factor versus Reynolds number for plain tubes and tubes with twisted tape inserts (logarithmic scales).

4.3 Thermal Boundary Conditions

In laminar flow the thermal boundary condition affects the heat transfer process. There are two limiting boundary conditions: **uniform heat flux (UHF)** and **uniform wall temperature (UWT)**.

- The **UHF** condition is where the heat flux is constant along the heated length of the tube and uniform around the tube circumference. It is typically applicable to counter-current flow heat exchangers where the heat capacity rates ($\dot{M}c_p$) of the fluids inside and outside the tubes are close and the heat transfer coefficients are sensibly constant along the heated tube length.
- The **UWT** condition is where the tube wall temperature is constant along the heated length of the tube. It is closely approached when the outside tube surface is heated by an isothermally condensing fluid, or cooled by an isothermally boiling fluid. It is also approached in a number of other situations, such as co-current flow heat exchangers where the outside and inside heat transfer coefficients and heat capacity rates are approximately equal, or in heat exchangers where the heat transfer coefficient and heat capacity rate for the outside are much larger than the corresponding values for the flow inside the tubes.

The heat transfer coefficients for these two cases are not the same. For tubes with no inserts (plain tubes) the heat transfer coefficient with fully developed laminar flow and UHF is some 17% greater than the UWT case under similar hydrodynamic conditions¹⁵. Likewise, with twisted tape inserts the UHF coefficient may be expected to be higher than the UWT coefficient, although not necessarily by the same amount.

There are situations where the actual thermal boundary conditions cannot be approximated by either the UHF and UWT conditions. Such a situation occurs where the tubeside presents the controlling resistance to heat transfer but the temperature change of the outside fluid is large, the heat capacity rates of the two streams not necessarily being of the same order. In cases such as these it is reasonable, in the absence of sufficient data, to assume UWT conditions since predictions on this basis are the more conservative.

Thermal boundary conditions are not generally significant with turbulent flow inside tubes.

Reference 19 should be consulted for a fuller treatment of thermal boundary conditions.

4.4 Entry Flow

A thermal boundary layer will develop along a heated or cooled tube until it fills the whole cross-section in a similar way that the hydrodynamic boundary layer develops. The axial distances required to attain the fully-developed boundary layers are known respectively as the thermal and hydrodynamic entry lengths. The local heat transfer coefficient decreases from a maximum value at the point of commencement of heating to a constant value at the point where the temperature profile is fully developed.

Heat transfer in the thermal entry length is a function of the Graetz number, Gz , where

$$Gz = \frac{\dot{M} c_p}{\lambda L} = \frac{\pi}{4} Re_o Pr \frac{D}{L}. \quad (4.1)$$

For a Prandtl number of unit the hydrodynamic and thermal entry lengths will be the same. With increasing Prandtl number the thermal entry length becomes progressively greater than the hydrodynamic length to the extent that for viscous liquids (high Prandtl number fluids) the hydrodynamic entry effect can normally be ignored without introducing significant error into any heat transfer calculation.

In most industrial heat exchanger applications, turbulent flow entry effects are small ($L/D \gg 40$) and may be neglected without introducing significant error¹⁴.

References 14 and 15 should be consulted for a fuller treatment of tube entry effects.

5. DATA FOR PRESSURE LOSS AND HEAT TRANSFER IN TUBES CONTAINING TWISTED TAPES

Comprehensive thermal-hydraulic design calculations have been presented by Manglik and Bergles^{16,17}. Data for both gases and liquids under heating and cooling conditions and for a wide range of tape geometries were used to establish their correlations. Previously published correlations in the laminar flow region were based mainly on limited water and ethylene glycol data. Manglik and Bergles^{16,17} greatly extended the range of the available data for the inclusion of polybutene 20. Details of their database are given in Tables 1 and 2. The range of data for laminar flow heat transfer is presented in Table 3.

In view of the comprehensive nature of the data, derivation of further calculation methods based almost entirely on those data was not considered appropriate. Consequently the Manglik and Bergles^{16,17} expressions for heat transfer and pressure drop are the ones presented here.

The expressions for heat transfer are for mean values of the Nusselt number over an axial distance, L , from the point of commencement of heating or cooling and under uniform wall temperature (UWT) conditions*. Expressions for the corresponding pressure drop are also presented.

* Much earlier work was based on uniform heat flux (UHF) conditions, the test sections being electrically heated. The limited range of fluid properties and sometimes otherwise limited data make wide application of the resultant correlations, outside the range of the data bases, unwise.

6. CALCULATION OF HEAT TRANSFER AND PRESSURE DROP

The basis of the following expressions for Nusselt number and Fanning friction factor is outlined in Appendix A. References 16 and 17 should be consulted for further details.

6.1 Laminar to Turbulent Flow Transition

In the absence of flow visualisation data, Manglik and Bergles^{16,17} concluded from their data plots that laminar flow occurred when $Sw \leq 1400$ and fully developed turbulent flow when $Re_o > 10\,000$.

6.2 Mean Nusselt Number in Laminar Flow ($Sw \leq 1400$)

$$Nu_{lam} = 4.612 \left[\left\{ \underbrace{(1 + 0.0951 Gz^{0.894})^{2.5}}_{\text{thermal entry effect}} + \underbrace{0.6413 \times 10^{-9} (Sw Pr^{0.391})^{3.835}}_{\text{swirl flow effect}} \right\}^2 + \underbrace{2.132 \times 10^{-14} (Re_{ax} Ra)^{2.23}}_{\text{combined forced and free convection effect}} \right]^{0.1} \left(\frac{\eta_b}{\eta_w} \right)^{0.14} \quad (6.1)$$

where $Gz = \dot{M}_c / (\lambda L)$, the Graetz number, (6.2)

$Sw = Re_{Sw} / \sqrt{y}$, the swirl parameter, (6.3)

$Re_{Sw} = \rho v_s D / \eta$, (6.4)

$v_s = v_a \left[1 + \left(\frac{\pi}{2y} \right)^2 \right]^{1/2}$, the swirl velocity at the tube wall, (6.5)

$v_a = \dot{M} / (\rho A_c)$, the mean axial velocity, (6.6)

$y = H/D$, the twist ratio, (6.7)

$Re_{ax} = \rho v_a D / \eta$, (6.8)

$Gr = g \rho^2 D^3 \beta \Delta T_w / \eta^2$ (6.9)

and $Ra = Gr Pr$, the Rayleigh number. (6.10)

Stated accuracy: data mostly within $\pm 15\%$. See error plot, Figure 1.