



Heat transfer enhancement in heat exchanger design and utilisation

Part 1. Tube inserts in single-phase flow

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(Continued on inside back cover)

HEAT TRANSFER ENHANCEMENT IN HEAT EXCHANGER DESIGN AND UTILISATION

PART 1. TUBE INSERTS IN SINGLE-PHASE FLOW

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HEAT TRANSFER ENHANCEMENT IN HEAT EXCHANGER DESIGN AND UTILISATION

PART 1. TUBE INSERTS IN SINGLE-PHASE FLOW

1. NOTATION AND UNITS

		Units	
		SI	British
A	heat transfer area of tube internal surfaces, $N_t \pi D L$	m ²	ft ²
A_c	axial flow cross-sectional area associated with total mass flow	m ²	ft ²
A_{ext}	heat transfer area of tube external surfaces, $N_t \pi D_{ext} L$ (see Appendix B3)	m ²	ft ²
a	coefficient in j -factor equation (see Section 8)	—	—
b	exponent in j -factor equation (see Section 8)	—	—
c	heat exchanger cost, in UK Sterling (see Equation (9.5))	£	£
c_a	heat exchanger area cost factor (see Equation (9.5))	£/m ²	£/ft ²
c_b	heat exchanger base cost, in UK Sterling (see Equation (9.1))	£	£
c_l	heat exchanger length cost factor (see Equation (9.3))	£/m	£/ft
c_m	heat exchanger material cost factor (see Equation (9.3))	£	£t
c_p	specific heat capacity at constant pressure	J/kg K	Btu/lb °F
c_y	heat exchanger year cost factor (see Equation (9.5))	£	£t
D	tube inside diameter	m	ft
D_{ext}	tube outside diameter	m	ft
D_s	shell diameter	m	ft
f	Fanning friction factor, $D \Delta p g_c / (2 \rho N_p L v_a^2)$	—	—
g_c	constant of proportionality in Newton's second law of motion	— (1)	lb ft/lbf h ² (4.17 × 10 ⁸)
H	180° twist pitch of twisted tape insert (see Section 7.1)	m	ft

j	Colburn j -factor (dimensionless heat transfer coefficient, see Section 3.2)	–	–
k_1, k_2, k_3	parameters defined in Equations (8.4), (8.7) and (8.10) respectively	–	–
L	axial tube/shell length	m	ft
$\dot{M}$	total mass flow rate	kg/s	lb/h
m	exponent defined in Equation (8.13)	–	–
N_p	number of tubeside passes	–	–
N_{pp}	number of pass partitions	–	–
N_t	number of tubes in tube bundle	–	–
Nu	Nusselt number, $\alpha_t D / \lambda$	–	–
Pr	Prandtl number, $c_p \eta / \lambda$	–	–
Δp	tubeside pressure drop	N/m ² (Pa)	lbf/ft ²
PI	performance index for heat exchanger, $\sqrt{St^3/f}$	–	–
P_t	tube pitch	m	ft
$\dot{Q}$	rate of heat transfer between streams (heat load)	W	Btu/h
Re	Reynolds number of tubeside flow, $\rho v_a D / \eta$ or $\frac{\dot{M}}{A_c} \cdot \frac{D}{\eta}$	–	–
r_f	fouling resistance	–	–
$r_{f,ext}$	total fouling resistance at external surface of tube wall	–	–
St	effective mean Stanton number at tube walls (see Equation (3.7))	–	–
T	temperature	K	°R ($\approx$ °F + 460)
ΔT_m	effective mean temperature difference between streams (see Section 3.2)	K or °C	°R or °F
U_i	overall heat transfer coefficient, referred to bare internal surface area of single tube	–	–
v_a	superficial axial fluid velocity in tubes,	m/s	ft/h
x_1	coefficient in friction factor equation (see Section 8)	–	–
x_2	exponent in friction factor equation (see Section 8)	–	–

y	twist ratio of twisted tape insert = H/D (see Section 7.1)	–	–
α_t	film heat transfer coefficient on tubeside	$\text{W/m}^2 \text{ K}$	$\text{Btu/ft}^2 \text{ h}^\circ\text{F}$
α_s	film heat transfer coefficient on shellside	$\text{W/m}^2 \text{ K}$	$\text{Btu/ft}^2 \text{ h}^\circ\text{F}$
β_1	tube pitch factor in Equations (8.15) and (8.16)	–	–
β_2	shell diameter factor in Equations (8.15) and (8.16)	m	ft
η	dynamic viscosity	kg/m s	lb/ft h
λ	thermal conductivity	W/m K	$\text{Btu/h ft }^\circ\text{F}$
ρ	fluid density	kg/m^3	lb/ft^3

Subscripts

in	inlet conditions
max	maximum value
s	shellside
t	tubeside
out	outlet conditions
1,2	streams 1 and 2 respectively

2. INTRODUCTION

2.1 Purpose and Scope of This Item

This Item is one of a group concerned with the estimation of heat transfer and pressure loss in tubular heat exchangers. It provides the engineer with a practical guide to the benefits of in-tube heat transfer enhancement devices (tube inserts) and illustrates how these benefits can be rapidly evaluated. It is supplemented by a computer program, referred to as 'ENTRANS', which generates approximate designs for shell-and-tube exchangers with plain tubes and a range of tube inserts. The methods of this Item and the computer program are applicable to the flow of single-phase Newtonian fluids. The correlations for heat transfer and pressure drop were derived from published data for the various tube insert types.

Applications for tube inserts fall into two separate categories:

- their use in new exchangers, and
- their use in increasing the performance of existing units (that is, revamping).

This Item concentrates mainly on the use of tube inserts in new equipment. However, guidance is given in Section 6 on the benefits of their use to improve the performance of existing exchangers.

The following types of tube insert are considered in this Item:

1. twisted tapes (high and low twist ratios), Section 7.1,
2. helically-coiled wires, Section 7.2,
3. wire matrix inserts (high and low matrix density), Section 7.3,
4. static mixers, Section 7.4,
5. core tubes, Section 7.5.

There are some misconceptions surrounding the use of heat transfer enhancement devices. For example, it is commonly but incorrectly stated that the use of tube inserts invariably leads to increased tubeside pressure drop. However, it can be shown that, in comparison with the plain tube case, their use often provides improved heat transfer at the same, or even lower, pressure drop. It is also often assumed that the use of tube inserts is only worthwhile when applied to a stream having a controlling heat transfer coefficient. Again this is not necessarily true, and the point is demonstrated in the worked examples of Section 10.

Some tube inserts (for example, wire matrix inserts and static mixers) are specifically designed to provide good heat transfer at low superficial Reynolds numbers. Since low Reynolds number flows are often associated with high viscosity liquids, the significant benefits that can frequently be obtained by using inserts with low viscosity fluids are often overlooked.

2.2 Scope of the Program ENTRANS

The computer program, ENTRANS, allows the engineer to assess rapidly the benefits of using tube inserts in tubular heat exchangers. This assessment can be made by inspection of approximate designs for two-stream shell-and-tube exchangers with and without inserts on the basis of the design parameters or least capital cost.

The program produces a series of designs over the specified range of tubeside pressure drop for both plain and enhanced tubular exchangers, using the methods described in Section 8. The key capital cost benefits that can result from the application of tube inserts in heat transfer enhancement can also be assessed.

The program can be used to investigate the effects of tube inserts on the following factors:

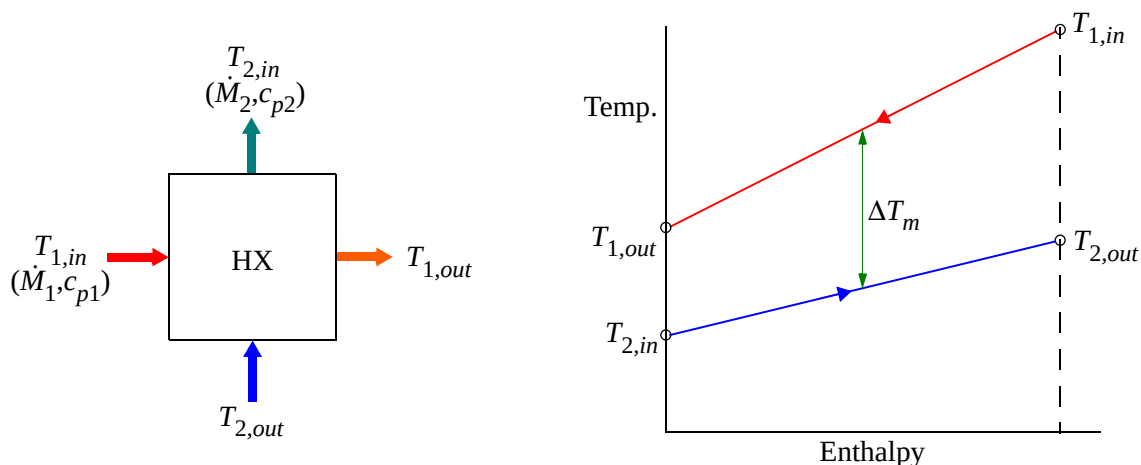
- area reduction (see Section 4.1),
- tube pass reduction (see Section 4.2)
- reduction in the number of shells (see Section 4.3)
- effective use of pressure drop (see Section 4.4).

Since the program provides predictions of the film temperature at the tube wall, it can also be used to assess fouling mitigation in thermally-controlled fouling situations (see Section 4.6). Other areas of potential benefit outside the direct scope of the program are operability improvements (see Section 4.5), increased heat recovery (see Section 4.7), and improved flow distribution (see Section 4.8).

The scope of the program and installation notes are summarised in Appendix B. A description of the required input data is given in Section B2 and the output from the program is summarised in Section B3.

3. DEFINITIONS

3.1 Two Stream Tubular Heat Exchanger Model



Sketch 3.1 Simple two stream tubular exchanger model

A simple model of a two stream tubular heat exchanger is shown in Sketch 3.1, in which heat is transferred from a hot stream (stream 1) to a cold stream (stream 2).

In passing through the heat exchanger, the hot stream cools from $T_{1,in}$ to $T_{1,out}$ while the cold stream is heated from $T_{2,in}$ to $T_{2,out}$. By a simple heat balance the rate of heat transfer between the streams (which is commonly referred to as the “thermal duty” or “heat load”) is

$$\dot{Q} = (\dot{M}c_p)_1(T_{1,in} - T_{1,out}) = (\dot{M}c_p)_2(T_{2,out} - T_{2,in}), \quad (3.1)$$

where $(\dot{M}c_p)_1$ and $(\dot{M}c_p)_2$ are the average thermal capacity rates in each stream. In addition, the rate of heat transfer is

$$\dot{Q} = U_i A \Delta T_m, \quad (3.2)$$

where U_i is the overall heat transfer coefficient between the streams at the operating conditions, A is the effective heat transfer surface area and ΔT_m is the effective mean temperature difference. Further guidance on heat exchanger models is given in ESDU 98003.

In the analysis of heat transfer and pressure drop of this Item, U_i is calculated on the basis of the internal surface area of a single tube and is assumed to be constant along the length of the exchanger. The effective heat transfer area is

$$A = N_t \pi D L \quad (3.3)$$

where D and L are average values of the tube inside diameter and tube length respectively, and N_t is the number of tubes in the bundle.

In this Item it is assumed that there are the same number of tubes in each pass, and the superficial average flow velocity in the tubes is given by

$$v_a = \frac{\dot{M}}{\rho A_c}, \quad (3.4)$$

in which A_c is the axial flow cross-sectional area associated with the total mass flow rate, $\dot{M}$, and is defined as

$$A_c = \frac{N_t \pi D^2}{4 N_p}. \quad (3.5)$$

In this definition, N_t and N_p are, respectively, the number of tubes in the bundle and the number of tubeside passes in the exchanger.

The rate of heat transfer between the streams can also be expressed in terms of tube conditions as

$$\dot{Q} = \alpha_t A \Delta T_m = (\dot{M} c_p)_t |T_{t,in} - T_{t,out}|, \quad (3.6)$$

where α_t is the tubeside film heat transfer coefficient and is assumed constant along the length of the exchanger.

Under all flow conditions the effective mean Stanton number at the tube wall is evaluated from the expression

$$St = \frac{\alpha_t}{\rho v_a c_p} = \alpha_t \frac{A_c}{\dot{M} c_p}, \quad (3.7)$$

where, within the tubes, ρ and c_p are average values of the density and specific heat capacity of the fluid respectively.

3.2 Correlations for heat transfer and pressure drop

Reliable heat transfer and pressure drop data for heat exchangers can be obtained from experimental measurements or analytical correlations such as those provided by ESDU. The correlations presented in this Item use the following terms.

Heat transfer performance is modelled by the Colburn j -factor, which is a dimensionless heat transfer coefficient and is given by

$$j = \frac{Nu}{Re Pr^{1/3}} = St Pr^{2/3}, \quad (3.8)$$

where Reynolds number is based on superficial velocity and the inside tube diameter, and is given by

$$Re = \frac{\rho v_a D}{\eta}. \quad (3.9)$$

The Nusselt number is

$$Nu = \frac{\alpha_t D}{\lambda} . \quad (3.10)$$

Prandtl number is defined as

$$Pr = \frac{c_p \eta}{\lambda} . \quad (3.11)$$

The frictional pressure drop on the tubeside (excluding nozzle and turnaround losses) is

$$\Delta p = \frac{2fN_p L \rho v_a^2}{Dg_c} . \quad (3.12)$$

Expressions for j and f for the range of tube inserts considered in this Item, over the laminar, transitional and turbulent flow regimes as appropriate, are given in Appendix A.

4. APPLICATIONS OF HEAT TRANSFER ENHANCEMENT

The best known application of in-tube heat transfer enhancement is capital cost reduction through reduced heat transfer area as discussed in Section 4.1. However, potential benefits are far wider ranging and the use of tube inserts can, for a given duty, lead to a reduction in the number of tube passes or the number of exchanger shells required: these benefits are discussed in Section 4.2 and 4.3 respectively. Tube inserts can also allow fuller use to be made of available pressure drop (that is, utilising pressure drop that would otherwise have to be wastefully dissipated across valves): this benefit is discussed in Section 4.4. Other potential benefits include operability improvement (see Section 4.5), fouling mitigation (Section 4.6), increased heat recovery (Section 4.7) and improved flow distribution (Section 4.8).

4.1 Area Reduction

As noted in Section 2.1, a widespread misconception about the use of tube inserts is that heat transfer enhancement is achieved at the inevitable cost of added pressure drop. It is true that at a given Reynolds number both the heat transfer coefficient and the friction factor of a tube fitted with an insert are generally much higher than those found for the plain tube, and it is found that at a fixed velocity heat transfer enhancement leads to increased pressure drop. However, when designing a new exchanger, the engineer has the freedom to reduce the velocity at which tubes fitted with inserts operate. Starting with a plain tube design, the designer can increase tube count to reduce velocity and then install an insert to promote the heat transfer whilst restoring the pressure drop. The result is often an improvement of heat transfer at the same, or lower, pressure loss.

This result can be demonstrated analytically. By rearrangement of Equation (3.6), the heat transfer on the tubeside of an exchanger can be related to the tubeside heat balance by

$$A = \frac{(\dot{M}c_p)_t}{\alpha_t} \frac{|T_{t,in} - T_{t,out}|}{\Delta T_m}, \quad (4.1)$$

and by combining Equations (4.1) and (3.7), the required heat transfer surface area of the exchanger can be expressed as

$$A = \frac{A_c}{St} \frac{|T_{t,in} - T_{t,out}|}{\Delta T_m}. \quad (4.2)$$

By introducing Equation (3.3), (3.4) and (3.5) into Equation (3.12), the pressure drop associated with flow through the tube is given by

$$\Delta p = \frac{1}{2} \frac{f}{\rho g_c} \frac{A}{A_c} \left(\frac{\dot{M}}{A_c} \right)^2. \quad (4.3)$$

This equation can be re-written as

$$A_c = \left(\frac{1}{2} \frac{\dot{M}^2}{\rho g_c} \frac{A}{\Delta p} f \right)^{1/3}. \quad (4.4)$$

Substitution into Equation (4.2) then yields

$$A = \left(\frac{|T_{t,in} - T_{t,out}|}{\Delta T_m} \right)^{3/2} \left(\frac{\dot{M}^2}{2\rho g_c \Delta p} \right)^{1/2} \left(\frac{f}{St^3} \right)^{1/2}. \quad (4.5)$$

The significance of the three terms in this equation is as follows.

- The first term considers the effect of the tubeside film temperature difference. It therefore models the interaction between shellside and tubeside performance. **Assuming that the overall heat transfer is controlled by the tubeside thermal resistance** this term can be considered fixed once the rate of heat transfer between the streams (that is, the thermal duty) of the exchanger is fixed and is independent of tube type.
- The second term is associated with the hydraulic performance required from the exchanger. Again, being a duty requirement, it is independent of tube type.
- The third term deals with the thermo-hydraulic performance of the tubes. The lower the value of this term, the less the heat transfer surface area, A , required for the specified thermal duty, $\dot{Q}$, at the specified tubeside pressure drop, Δp . This is the only term that depends on tube configuration.

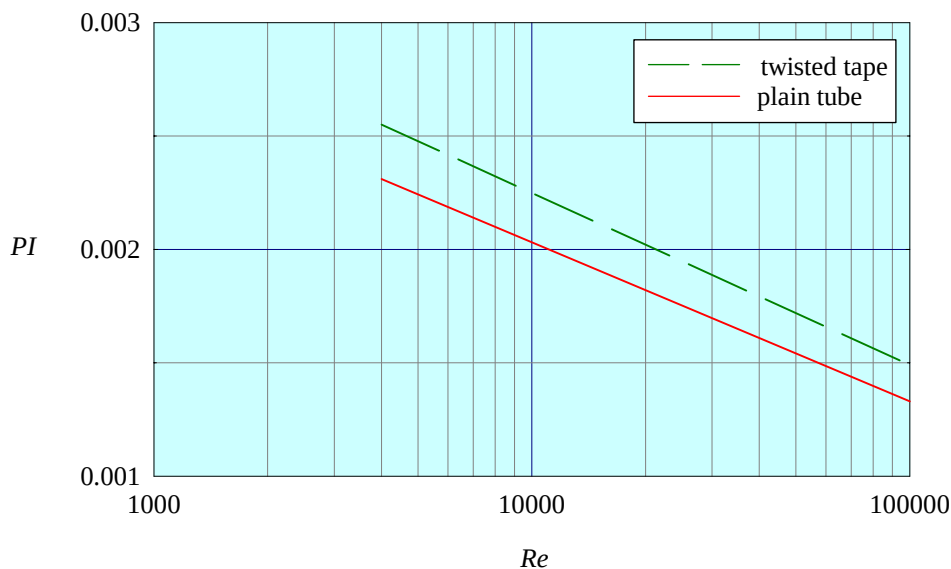
The third term can be inverted to yield a heat exchanger 'Performance Index',

$$PI = \sqrt[3]{\frac{St^3}{f}}. \quad (4.6)$$

The higher the value of this index the smaller is the heat transfer surface area required for a given duty.

Performance indices for a tube fitted with a twisted tape insert are compared with the plain tube case in Sketch 4.1 over a typical range of Reynolds number, Re . (Note: This sketch is simply representative. Different tape geometries will have different performance indices).

It is seen that, at any given Reynolds number, the tubes with the twisted tapes have higher Performance Indices than the plain tubes. (Similar improvements are found for other devices.)



Sketch 4.1 Typical performance indices

It is also seen that the Performance Index increases as the Reynolds number decreases. So, given that tubes with twisted tape inserts operate at a Reynolds number lower than that used for an equivalent plain tube, the enhanced tubes require less area at a given pressure drop.

The Performance Indices are valuable for illustration. However, they cannot be used for device selection, for without knowledge of operational Reynolds numbers, it is impossible to determine actual area benefits.

Operational Reynolds numbers depend on insert-type and are design case specific. Each insert type will have a different value for a given problem and these values will vary from case to case. An alternative approach is required for tube insert selection. Such an approach is described in Section 8.

4.2 Tube Pass Reduction

It is normal practice in exchanger design to restrict the length of the tubes due to mechanical constraints, layout constraints, cost or availability factors. Length constraints play a significant role in the design of tubular heat exchangers. For instance, exchangers requiring large surface areas are driven towards large tube counts. If such tube counts then lead to low Reynolds numbers the designer is tempted to increase the number of tubeside passes in order to maintain a reasonable tubeside heat transfer coefficient. The obvious alternative is the use of tube inserts to provide heat transfer enhancement.

Heat transfer enhancement is particularly effective in reducing the number of tube passes required in air cooled heat exchangers handling high and moderate viscosity fluids (such as lubricating oil). With plain tubed units, a large number of tube passes (typically 7 to 10) is required in order to obtain a reasonable tubeside heat transfer coefficient. Designs involving just one pass can be achieved if heat transfer enhancement is employed. Such reductions in tubepass arrangements lead to much simpler bundle designs and reduced capital cost. As the two different design cases presented in Table 4.1 show, significant size reductions are also possible.

TABLE 4.1 ILLUSTRATION OF AIR COOLED HEAT EXCHANGER DESIGNS

Case:	Lubricated Oil Cooler		Seal Oil Cooler	
	Plain Tubes	Wire matrix	Plain Tubes	Wire matrix
Area (m ²)	7072	1262	1010	163
Length (m)	9.75	7	9.75	7
Width (m)	1.7	1.4	0.24	0.18
Number of rows	14	6	14	6
Number of passes	10	1	7	1
Pressure drop (kPa)	94	113	53	104

4.3 Reduction of Number of Shells

With shell-and-tube exchangers the use of multiple tube passes has four detrimental effects. Firstly, it leads to a reduction in the number of tubes that can be accommodated in the shell because of the need for pass partition lanes. Secondly, the pass partition lanes introduced into the bundle can give rise to increased bundle bypassing and a reduction in the shellside heat transfer coefficient. Thirdly, it gives rise to wasted use of tubeside pressure drop in the return headers. Finally, and most seriously, the use of multiple tube passes can lead to a serious reduction in the effective temperature differences with the subsequent need for multiple shells in series counterflow. However, the use of multiple tube passes may have the major benefit of simplifying the problem of handling thermal expansion effects, either through the use of U tubes or floating heads.

It is often assumed that turbulent flow is best for heat transfer but this is not necessarily true. The use of low Reynolds number flow heat transfer enhancement should be considered in this context. For example, wire matrix inserts are a very effective means of enhancing heat transfer. The heat transfer coefficient, α_t , achieved in a single tube at a Reynolds number of 2000 using such an insert to process a light organic liquid (viscosity of 0.0055 kg/m s (0.55 cP), Prandtl number of 5.9) is calculated⁸ as 1 945 W/m² K. A Reynolds number of approximately 28 000 is needed to achieve this coefficient in a plain tube. The calculated pressure gradient incurred in the tube fitted with the insert would be 3.5 kPa/m and in the plain tube operating at the higher velocity is 4.0 kPa/m. So, the pressure loss incurred by the enhanced tube is lower than that of the plain tube even at fixed length. If the higher plain tube velocity has been bought at the expense of increased tube passes, the plain tube pressure drop could be significantly greater than the enhanced tube pressure drop for the same performance.

These advantages translate into design improvements. By means of illustration, Table 4.2 presents a comparison of three designs developed for lean/rich exchangers on a plant used for the separation of aromatics (benzene/toluene/xylenes). The first design is a plain tubed unit. The second is an enhanced unit handling the duty in the minimum number of shells. The third is an enhanced unit handling the duty with the minimum heat transfer area. Despite the additional mechanical design complexity involved with moving to a single tube pass (that is, installation of shell bellows to handle thermal expansion), the enhanced units could show significant cost savings.

It is clear that both laminar and turbulent flow enhancement devices have a role to play in the processing of low viscosity fluids. It is also clear that the pursuit of turbulence to promote heat transfer in plain tubes can be wasteful.

TABLE 4.2 ILLUSTRATION OF LEAN/RICH EXCHANGER DESIGNS

Case:	Plain Tubes	Wire matrix (Low density)	Wire matrix (High density)
TEMA Type	AEU	AEL	AEL
No. Shells	3	1	2
No. of Passes	2	1	1
Area (m ²)	1092	742	560
Length (m)	6	6	6
Flow length (m)	36	6	12
Tubeside pressure drop (kPa)	138	16	138
Shellside pressure drop (kPa)	94	25	95

There are occasions when the designer may deliberately choose to specify a multi-shell design in order to make more effective use of available pressure drop. With many applications more pressure drop is available than can be used in a single shell. This pressure drop can be utilised in a multi-shell design giving the designer a choice between a single shell design or a series of shells of smaller diameter with lower overall area.

The essence of this problem is effective use of pressure drop, as discussed in Section 4.4. The pressure drop associated with friction in plain tubes is usually relatively low so the amount of pressure drop utilised in the promotion of heat transfer is low. As discussed in Section 4.1, higher heat transfer can be achieved for given pressure drop using enhancement. Consequently, when it comes to the effective use of excess pressure drop the designer can, by the use of tube inserts, often use a single shell of smaller diameter and lower surface area than is possible with a plain tube design.

4.4 Effective Use of Pressure Drop

The effective use of pressure drop is described in Section 4.3 in the context of the number of shells used in a design. However, other considerations are also important in this context. Generally designers seek to maximise the use of available pressure drop. Given a large allowable pressure drop, the designer generally operates at a tube velocity as high as possible. The higher the velocity the higher the heat transfer coefficient.

However, there are constraints on this velocity and it is usual to impose a maximum allowable tube velocity dependent on the nature of the fluid and the tube material. Velocities higher than this allowable maximum could possibly give rise to erosion problems. Corrosive liquids or liquids containing suspended solids lead to lower maximum allowable tube velocities.

As already noted, there is also a balance between achievable velocity and exchanger complexity. High velocities may be bought at the cost of increased number of tube passes and/or increased number of exchanger shells. So, situations exist in which full use of available pressure drop is impossible, impractical or undesirable.

These arguments apply to both plain tubes and tubes with inserts. However, a range of types of tube inserts is available to the designer and for any given type a range of geometries exists. It is therefore often possible to select an insert that makes better use of the allowable pressure drop than is achievable with the plain tube.

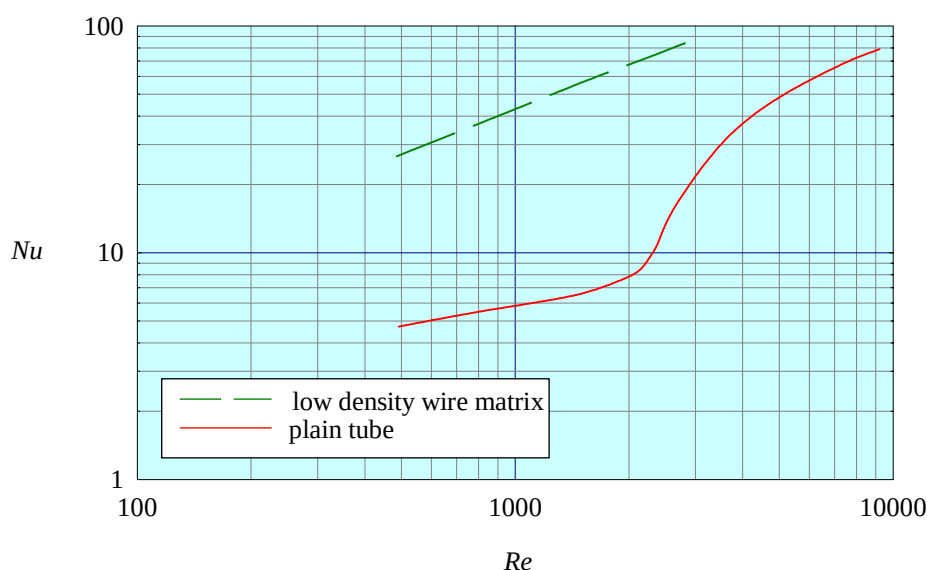
4.5 Operability Improvement

Process plant operability has many aspects, the most important of which are controllability, flexibility and availability. For good controllability heat exchangers must respond to imposed disturbances in a manner that is predictable and readily rectifiable. The nature of the disturbances must be within the design limits of the plant. Plain tubed exchangers operating under fully-developed turbulent flow conditions are generally well behaved. However, those that are close to, or in, the transitional flow region can present control problems as small changes in throughput or temperature can lead to unexpectedly large changes in overall performance.

Flow transition can result from physical property changes in the fluid or from flow rate changes. This can be illustrated by a shell-and-tube exchanger cooling oil. At the entry to the exchanger the oil is hot, its viscosity is low and the flow could be fully turbulent. However, as the oil cools its viscosity increases. If the oil undergoes a large temperature drop the viscosity increases significantly. The consequence of such a change could be that the flow leaving the exchanger is deeply laminar. This change in flow regime presents two problems. Firstly, the tubeside heat transfer coefficient decreases significantly along the length of the tube. Second, the relationship between flow rate and heat transfer coefficient varies substantially along the length of the tube. Consequently, not only is the overall performance of the exchanger difficult to predict but also the response of the exchanger to changes in flow or temperature is uncertain.

The problems posed by flow transition within the exchanger are particularly acute when the temperature driving forces for heat transfer are large and the flow is in the same direction as the buoyancy forces (for example, in heating with vertical upflow). In these circumstances, increasing thickness of the laminar boundary sub-layer can occur. This gives rise with plain tubes to a delay in the transition from laminar to turbulent flow with the 'Critical Reynolds number' being a function of buoyancy forces. The effects are well demonstrated in the experiments reported by Petukhov and Noll¹.

Designers can often improve operability by employing tube inserts. With tube inserts the transition from laminar flow occurs at much lower Reynolds numbers. The relationship between heat transfer coefficient and flow rate is then well behaved (see relationships presented in Appendix A). This is demonstrated for an example shell-and-tube heat exchanger in Sketch 4.2, where the performances with plain tubes and tubes fitted with a low density wire matrix element are compared across a Reynolds number range. Wire matrix inserts act by disrupting the wall boundary layer, they combat the problem of laminarisation and are particularly suitable in applications involving large temperature driving forces.



Sketch 4.2 Example shell-and-tube exchanger performance

Whilst process plants are generally designed on the basis of a specified throughput, they are usually required to operate over a range of conditions. As a consequence, each item of equipment must have good flexibility (that is, must provide good performance over a range of throughputs). Since flow transition can radically reduce performance, plain tubed units operating close to the transitional flow regime should be avoided on flexibility grounds.

4.6 Fouling Mitigation

Modern process plants are expected to operate continuously over long periods (typically for 8 000 hours each year), and unplanned downtime is expensive. Perhaps the commonest cause of such downtime is the requirement to clean exchangers due to the build-up of deposits (that is, fouling). There are many causes and forms of heat exchanger fouling. The commonest forms are

- solids deposition,
- crystalline deposition,
- biological deposition,
- thermal degradation,
- wax formation,
- corrosion.

There is strong evidence to indicate that heat transfer enhancement reduces any fouling that is due to solids deposition. In the case of twisted tape inserts this is brought about through increased velocity and wall shear stress, whereas with wire matrix inserts turbulent fluid motion at the tube wall is believed to hinder solids deposition.

Recent experimental results indicate that inserts can also reduce some forms of biological fouling.

In neither of the above cases can these observations yet be generalised and all new applications in these areas must be considered experimental.

The area in which enhancement has a clear role to play in fouling mitigation is that in which the fouling mechanism is temperature dependent. Examples of such mechanisms are wax deposition (found, for example, during the cooling of some hydrocarbon streams) and thermal degradation (such as occurs during the cracking of crude oil). In these cases wall temperatures can be increased (in the case of cooling) or reduced (in the case of heating) through enhancement. If the change is such that the wall temperature is placed outside the fouling threshold temperatures, it is possible to eliminate fouling. In other cases, the rate of fouling can be reduced.

The benefits of enhancement in this context can only be quantified at the design stage for fluids for which the temperature at which fouling starts to occur is known. An example of such a fluid is a solvent used in the petrochemical industry to extract selectively aromatics from the paraffins and residual naphthalenes present in catalytic reformates. Typically, the process consists of an extraction column, an extraction distillation column and a solvent recovery column. Since the solvent has a high boiling point it flows through both column reboilers before being returned to the extraction column. The process is energy intensive and heat recovery is an important design consideration. In terms of heat recovery, the most important stream is the lean solvent being returned from the solvent regeneration column to the extraction column.

The solvent starts to degrade at a temperature close to the temperature at which the solvent recovery column operates. The products of this thermal degradation ('coke') can cause fouling of the solvent recovery reboiler, and subsequent deposition of coke throughout the whole of the heat recovery train as the solvent is continually recycled around the plant.

Given this degradation danger, control of the reboiler wall temperature is very important. It must be kept below the degradation threshold. It is common practice to use vertical thermosiphon reboilers for such duties. Good heat transfer is achieved within these reboilers once reboiling has been initiated. However, the solvent solution enters the reboiler slightly subcooled. The liquid velocities at the entrance to the tubes are also quite low. The result is quite poor heat transfer, and as a consequence, quite high wall temperatures in the lower parts of the reboiler. The heat transfer in this region can be enhanced through the provision of inserts. These inserts need not extend along the full length of the tube, and can extend across the sub-cooled region with the result that the overall effect on flow resistance is not necessarily large.

4.7 Increased Heat Recovery

Heat recovery is important to the economics of many plants. Tube inserts can be used to allow smaller temperature differences between streams, resulting in increased heat recovery for a given surface area.

Heat recovery networks often have two important characteristics:

- (i) the exchanger heat loads are high, and
- (ii) exchangers located at the heat recovery pinch have close temperature approaches (that is, low temperature differences).

As noted in Section 4.2, with plain tubed exchangers high heat loads generally lead to multiple tube passes. This, in turn, leads to the use of multiple shells. The need for multiple shells then places an economic restriction on the temperature approach and heat recovery level.

If, rather than increasing the number of tube passes to maintain turbulent flow, the designer uses heat transfer enhancement then the need for multiple shells may be avoided. The area benefits associated with tube inserts also affect the heat recovery economics. The result is an economic trade-off (that is, capital cost against energy-use saving), that justifies a closer temperature approach and increased heat recovery.

4.8 Improved Flow Distribution

Maldistribution of fluid across a tube bundle can occur when the nozzle and header momentum are large in comparison with frictional resistance presented by the tubes. Under these circumstances tube inserts can be used to increase the frictional resistance of the tubes to even out the flow distribution.

5. INAPPROPRIATE APPLICATION OF HEAT TRANSFER ENHANCEMENT

The procedure developed and demonstrated in this Item (and incorporated in the ENTRANS computer program) allows the designer to identify where and when the use of tube inserts provides capital cost or exchanger performance advantages. However, even given such advantages there are situations in which tube inserts should not be used. These situations are device-specific and some common limitations are listed below.

- Where the heat exchange equipment needs periodic sterilisation (for example, in many food processing applications),
- where the fluid being handled contains fibrous material,
- where the fluid contains suspended solid particles of a size comparable with the flow blockage due to the insert geometry,
- where suspended solids have a tendency to agglomerate.

Tube inserts have been found to reduce fouling (see Section 4.6) even in situations that are not generally considered to be very dependent on wall temperature (one example is the reduction of some forms of biofouling). Since fouling of tubes fitted with inserts can result in subsequent difficulties in insert removal and tube cleaning, care should be taken in using inserts in any situation in which fouling mitigation may not occur. In these cases prior testing is strongly recommended.

Finally, when considering the use of tube inserts thought should be given to upstream processes and how failures in operation can affect the fluid being fed to the exchanger. For instance, there have been cases in which reactor malfunctions have resulted in polymer-forming materials being fed to exchangers fitted with inserts. Subsequent polymerisation reactions within these exchangers then resulted in fouled surfaces and difficulties in insert removal and exchanger cleaning. In other cases upstream process malfunction has resulted in large suspended solids being fed to such exchangers. The inserts then acted as filters with resultant flow restriction.