Chapter 2

DESIGN CONSIDERATIONS FOR ENHANCED HEAT EXCHANGERS

SUMMARY: This chapter focuses on heat transfer augmentation of tubular heat exchangers and describes existing and prospective applications of tubular heat transfer augmentations to a wide range of industries. Thermal, mechanical and economical considerations of particular importance are also presented.

2.1 Introduction

Enhanced tubes are used extensively in the refrigeration, air-conditioning and commercial heat pump industries while, in contrast, their consideration for use in the chemical, petroleum and numerous other industries is still not standard practice, although increasing. Designing *enhanced* tubular heat exchangers results in a much more compact design than conventional *plain* tube units, obtaining not only thermal, mechanical and economical advantages for the heat exchanger, but also for the associated support structure, piping and/or skid package unit, and also notably reduced cost for shipping and installation of all these components (which often bring the installed cost to a factor of 2 to 3 times that of the exchanger itself in petrochemical applications). The compact enhanced designs also greatly reduce the quantities of the two fluids resident within the exchanger, sometimes an important safety consideration. This chapter describes some of the practical considerations and advantages regarding the use of enhanced tubes and tube inserts in tubular heat exchangers and provides some guidelines for identifying their applications.

2.2 Thermal and Economic Advantages of Heat Transfer Augmentations

There are many thermal advantages of utilizing augmentations that must be weighed against their higher cost relative to plain tubing and their economic benefit on plant operation. For many small increases to production capacity (10 to 30%), the purchase and installation of completely new exchangers cannot be justified economically. However, when the heat exchangers are the "bottleneck" of a unit operation, then augmentations may be the right solution.

The principal advantage of introducing an augmentation is the possibility of substantially increasing thermal duty to meet the needs of new process conditions or production goals. This can be achieved either by:

1. Installing removable inserts inside the tubes,
2. Replacing a removable tube bundle with a new enhanced tube bundle,
3. Replacing the heat exchanger with a new enhanced tube heat exchanger of the same size or smaller.

The first two of these interventions can be completed without any modifications to the heat exchanger itself while all three can be implemented without changes to the original piping connections and to its supports. Hence, these interventions have the benefit of a minimum effect on the operating schedule of the production plant.

What about replacements of existing installed units? As a prime example of this latter point, a removable tube bundle is easily replaced during shutdown by a new enhanced tube bundle. Or, a fixed tubesheet unit can be partially replaced by using the same heads, piping and supports. Tube inserts, on the other hand, can be installed inside the tubes of an existing exchanger during a normally scheduled shutdown, resulting in no lost production. The installation may require that some (or all) of the pass partition plates in the heads be removed to reduce the number of tube passes and thus meet pressure drop limitations when installing twisted tapes, especially for laminar flows. These types of interventions have very high payback ratios and plant operating reliability because of their simplicity and avoid the necessity of purchasing a new larger plain tube unit, which would require costly engineering services and expensive changes to the heat exchanger supports, its foundation and piping *and* also loss of production during these modifications.

What about new units in new plants? For new heat exchangers, a well-optimized plain tube unit is normally the easy way out for heat exchanger designers, even though they unwittingly are often paying a premium of 20-50% in the cost of the unit compared to an enhanced unit for the same service. Another consideration regards difficult applications where space is not available for two or more exchangers in parallel or where weight/bundle removal restrictions on units mounted on structures is a problem, and these are situations where heat transfer augmentations have been used to advantage by well informed heat exchanger designers.

What about cost savings? The cost savings for appropriate applications of enhanced tubes to shell-and-tube heat exchangers in the petrochemical industries typically range from $10,000 to $200,000 per unit or more. Hence, several of these interventions a year easily justifies the engineering cost for evaluating otherwise conventional designs for appropriate use of an enhanced tube, such as a Wolverine Tube Trufin or Turbo-Chil tube with internal and external enhancements (internal helical ribs and external low fins).

What about alloy tubes? When utilizing high alloy tubes in heat exchangers (stainless steel, titanium, nickel alloys, duplex stainless steels, etc.), applying the appropriate augmentation can very significantly reduce their first cost. The augmentation may not only reduce the cost of the tubing, but also those of the heads and tubesheets (smaller diameters, smaller wall thicknesses, fewer tube holes to drill, less alloy cladding material, etc.). Even for conventional carbon steel heat exchangers, if the entire cost of the heat exchanger is included as it should be its total cost to the plant, a more compact, lighter weight enhanced shell-and-tube unit can greatly reduce the cost of shipping and installation. It is often estimated that the installed cost is 2 to 3 times the cost of the heat exchanger itself and hence a smaller enhanced unit will achieve a significant first cost savings when the true total cost is considered.

What are typical prices per foot or meter of enhanced tubing? These are difficult to describe in a simple set of tabular values since their prices are very dependent on the particular tube material involved (primarily related to its hardness and hence resistance to deformation during the enhancement production process) and the wall thickness specified and the base cost of the bare tube. In general, the enhanced tube cost multiplying factor falls as the base tube material cost increases. Price information is readily available by contacting the enhancement's manufacturer (Wolverine Tube Inc.) for an offer. A doubly-enhanced tube version is also often available for the application (enhanced on the tube-side as well as on the shell-side) and it typically is the best thermal and economic choice.

What about heat exchanger cost savings? As a quick measure, a thermal designer is tempted to compare an enhanced tube on a cost per foot basis versus its equivalent plain tube in the same material and wall gauge, which however is equivalent to purchasing a portable PC on its sticker price per kilogram of weight irrespective of performance. For low finned tubes, sometimes it is suggested to compare them on a ($\$/m^2$)/m) basis, i.e. unit cost per meter of surface area per unit length. Since a low finned tube often has about 3 times the external surface area of a plain tube, if it costs 1.5 more per meter than a plain tube (a typical rule-of-thumb value), then the real cost is about one-half that of a plain tube on this surface area basis. A more realistic comparison would be to look at the respective cost per meter of tubing divided by the overall heat transfer coefficient for the optimized units, which gives a cost to performance ratio. This approach includes the entire thermal effect of internal and external heat transfer augmentation and fouling factors in the evaluation. Yet another basis is to compare the total cost of the tubing for each type of unit, since that is what the fabricator actually pays for the tubing. Even so, this is still not a realistic evaluation since a large savings in the exchanger's shell, heads, tubesheets, and fabrication costs are gained by going to a more compact unit. Overall, the best choice is to get competitive bids from the heat exchanger fabricator on the conventional plain tube unit and on the enhanced tube unit, both optimized for the application. Typical savings will be from 15-40% even when the total tubing costs are identical. If we assume that (i) tubing, (ii) all other materials plus fixed costs and (iii) manpower each contribute equally (1/3 each) of the total cost of the heat exchanger, it is easy to see that very significant savings are gained from the second and third category as the heat exchanger gets smaller in size. Thus, the best *simple* economical yardstick to apply to the comparison is that of size reduction, i.e. if an enhanced unit uses 1/3 less tubes it will cost 1/3 less than the conventional unit, including the higher price per meter of the enhanced tubing. This is typically quite close to reality and easy for the thermal designer to evaluate himself.

2.3 Thermal Design and Optimization Considerations

One of the first questions about enhanced tubes to be asked is *When can I use them*? Continuing the discussion above, an *old* rule of thumb says that an augmentation should be considered when that fluid's thermal resistance is *three* times that of the other fluid. However, because doubly-enhanced tubes are readily available, i.e. those augmenting both the tube-side and the shell-side processes, almost any application can benefit thermally, and this old rule of thumb is really now only old technology. Hence, it is important to determine which augmentation(s) are applicable to the situation and then to run some simulated heat exchanger designs to determine the magnitude of the benefit in reduced size (and cost if possible) of the heat exchanger.

An important point to remember is that using an enhancement on one side of the tube will have a *positive* effect on the other side. For instance, in boiling processes an augmentation applied to the heating fluid tube surface will increase the heat flux in the smaller enhanced unit and thus the boiling heat transfer coefficient on the other side of the tube wall (which is positively effected by the larger heat flux) while also reducing the number of tubes that increases the mass velocity (and convective heat transfer) too. In single-phase applications, an augmentation will reduce the size of a new heat exchanger and thus increase the fluid velocity on the other side of the tube. This "free" enhancement is often overlooked unless a thermal design is performed for the enhanced tube unit. This *secondary augmentation* often contributes to a notable fraction of the reduction in unit size.

Another important point to remember...do *not* impose unnecessary or unwitting design restrictions on an enhanced tube heat exchanger. Compared to an optimized plain tube unit for the same application, the enhanced tube exchanger will almost always optimize to a different bundle configuration, such as shorter tube length, fewer tubes, perhaps choice of a different tube diameter, fewer tube passes, fewer bundles in parallel, etc. Thus, do not self-impose unnecessary restrictions on an enhanced unit's design.

On the other hand, many plain tube units, especially *horizontal condensers*, are poorly optimized and end up with small tube length to shell diameter ratios (sure sign of costly designs) because of the maximum tube length limit imposed during construction of most petrochemical processing plants. In these plants, nearly always a maximum length of 20 ft (6.1 m) tube length is imposed, which often has a very negative effect on achieving the real potential thermal performance of large plain tube tube-side and shell-side condensers. On the contrary, enhanced tube units in these cases often optimize out to a shorter length than plain tube units and hence sometimes in these cases the enhanced tube unit is less than *one-half* the size of the plain tube unit or *two* units in parallel are now easily handled by *one* smaller enhanced unit. In these situations, often cost savings exceed 50% by applying an externally low finned tube, an internally microfinned tube or a doubly-enhanced tube.

The positive effect of the increased surface area of the enhancement should not be overlooked. The fouling factor is applied to the *entire* wetted surface area of an augmentation, not its nominal projection. This has the effect of reducing the thermal *resistance* due to the fouling factor; if the enhanced to plain surface area ratio is 3.0 such as for a low finned tube, the thermal resistance using the same fouling factor will be reduced to 1/3 the plain tube value. Even so, the *percent overdesign* for the fouled design compared to the clean design typically increases in an enhanced heat exchanger because the fouling factors remain the same in value while the tube-side and shell-side heat transfer coefficients are greatly increased, thus providing more *fouling insurance*.

The fouling factor also effects fin efficiency. The fouling resistance must be added to that of the heat transfer coefficient on the fin; hence the "effective" heat transfer coefficient on the fin is reduced, which increases the fin efficiency. Thus, the effective surface area of a finned surface increases as fouling increases, which partially compensates for the increase in fouling with time.

Most conventional heat exchanger design programs allow the user to input heat transfer data in one or more of the following manners, providing various possibilities to input information about enhancements to make an enhanced heat transfer design:

1. A fixed value of the heat transfer coefficient;
2. A fixed value to multiply the heat transfer coefficient relative to the internally calculated plain tube value;
3. A pressure drop multiplier term;
4. A single-tube nucleate pool boiling curve;
5. Constants for j-factor and f-factor curves;
6. A design safety factor that can be used as a multiplier for an enhanced heat transfer coefficient.

For intube single-phase processes, it is convenient to input a *fixed value* of the heat transfer coefficient or the *enhancement multiplier* in most cases. Instead, if the option is available, j- and f- factor curves can be entered. When using a multiplier, your program will multiply *its* plain tube coefficient's value, not necessary using the same correlation that was used to find the enhancement multiplier relative to the plain tube value.

Since tube-side and shell-side fouling factors are applied to their respective total wetted surface areas when calculating the overall heat transfer coefficient, for enhanced surfaces a conventional plain tube design software will normally incorrectly apply the fouling factors to the nominal internal and external surface areas of the tube. To properly determine the fouling resistances, the fouling factors must be applied to the actual respective surface areas. One way of doing this in a conventional plain tube design software is to divide the fouling factors by their respective surface area ratios first before inputting them into the program, which results in the same final effect. For example, if an internal helically ribbed tube has an internal area ratio of 1.8 with respect to the nominal area at the base of the ribs, then the tube-side fouling factor should be divided by 1.8 in order to respect the correct application of the fouling factor in calculating the overall heat transfer coefficient.

2.4 Mechanical Design and Construction Considerations

Mechanical Stress Calculations. Burst tests with internal pressure on integral low finned tubes have shown that the plain ends of the tube are the weakest point along the tube because the external helical fins act as reinforcement rings. One can also imagine that the same holds true for internal helical fins or ribs. Choice of the minimum wall thickness under the augmentation has a direct impact on the amount of metal per meter of tubing. Thus, this choice is economically important, especially for expensive alloy materials.

For most applications, the base wall thickness under the augmentation must be used for the mechanical stress calculations defined by various pressure vessel codes, such as that of ASME. For some applications, the ASME code will allow the original plain tube wall thickness before finning/deformation to be used in the minimum wall thickness calculation, rather than the wall thickness under the fins which would be more restrictive and lead to heavier tube walls. One such situation is for vacuum applications in surface condensers. Consult the applicable pressure vessel code for guidance.

Heat Exchanger Fabrication. Enhanced tubes normally have plain lengths at each end that are a little longer than the tubesheet thickness. This allows these tubes to be rolled and/or welded into the tubesheet. The outside diameter over the augmentation (such as a low finned tube) is equal to or slightly less than that of the plain ends. Thus, these tubes can be drawn into the tube bundle during assembly without any problems. Tube inserts normally have pull rings or attachments to install them, fix them in place and to remove them for cleaning.

U-Tubes. Nearly all integral heat transfer augmentations (i.e. those augmentations that are an integral part of the tube wall) can be bent into U's for U-tube heat exchangers. For the minimum-bending radius of a particular type of tube, one should refer to the manufacturer's recommendations (refer to Wolverine Tube Inc. recommendations).

Mean Metal Temperature Differences. Some fixed tubesheet plain tube designs result in very large mean metal temperature differences between the tubes and the heat exchanger shell. This large temperature difference causes unequal thermal expansion (or contraction) and large stresses. Since augmenting one fluid stream of an exchanger almost always shifts the controlling thermal resistance to the other fluid stream, an enhancement can be used to reduce the mean metal temperature difference and avoid using an expansion joint where they are undesirable. For instance, in a feed effluent heater with a single-phase hot gas on the shell-side being cooled by a tube-side fluid, the mean metal temperature difference will be reduced by about 1/3 by use of integral low finned tubes. In addition, the tube length will also usually optimize out to a shorter exchanger and thus further reduce the thermal differential stresses between the shell and tubes.

In any case, for a fixed tubesheet heat exchanger, it is good practice to check the effect of an augmentation on the mean metal temperature difference. When installing inserts in existing fixed tube units, it is good practice to compare the new mean metal temperature difference value against that used for the mechanical design of the exchanger.

2.5 Refrigeration and Air-Conditioning System Applications

The benefits of heat transfer augmentations in these systems are well-known to the industry and their use is the norm rather than the exception. Still, their selection, optimization and proper implementation are critical factors in order to realize the maximum benefit.

For intube boiling (or condensation), the aluminum star inserts that were once widely used have nearly been completely abandoned in favor of internally microfinned tubes. These microfinned tubes provide heat transfer augmentation similar to that of the star inserts but at a fraction of their increase in two-phase pressure drop. Wolverine Tube Inc. is a leading manufacturer of microfin tubes.

With air passing over air-conditioning coils, the newest louvered plate designs for "compact" heat exchangers provide a substantial increase in heat transfer compared to older designs. Thus, the inside coefficient is now often augmented with microfins. Use of an internal augmentation in turn reduces the length and weight of the unit, which reduces the number of louvered external plates required, increases air velocity and thus the air-side coefficient too. For a direct-expansion evaporator operating with the refrigerant in a stratified flow mode inside the tube, the top perimeter of the tube is not wetted by a plain tube while instead the helical microfins lift the liquid film around the top of the tube, making not only better use of the internal surface area of the tube but also the aluminum fins outside around the top of the tube as well.

For evaporation in flooded evaporators, many external augmentations are available commercially, most of which are normally produced in doubly-enhanced versions (i.e. with tube-side enhancement for the chilled water/cooling water/brine). Comparison of these augmentations should be made *not* on their single-tube performances but on their *bundle* performances, including tube-side augmentation for the water and the positive effect of the ID surface area ratio on reducing the fouling resistance.

For chilled water flowing inside tubes, highly efficient internal rib and fin designs provide a good tradeoff between heat transfer and pressure drop. For chilled water on the shell-side, i.e. dx-evaporator water chillers, the best tube selection may be a low finned tube with internal microfins where one pays for more tube metal per meter but will use only about one-half the meters of tubing.

2.6 Refinery and Petrochemical Plant Applications

Several examples of heat transfer augmentation interventions in chemical plants and refineries to look for are:

Single-Phase Exchangers:

1. Use of integral low finned tubes in heat exchangers when the limiting thermal resistance is on the shell-side;

2. Use of tube inserts (wire mesh or twisted tape types) are highly effective in laminar flows inside tubes;

3. Installation of inserts on the tube-side of heat recovery units to increase energy recovery via a larger overall heat transfer coefficient and smaller temperature approaches;

4. Utilization of inserts in oil coolers of compressors and turbines to solve an overheating problem or realize smaller exchangers that are more convenient in package units.

To overcome the fact that laminar flow heat transfer coefficients are almost independent of fluid velocity and performance is difficult to improve using plain tubes even resorting to large pressure drops, either inserts or external low fins may be the simple solution. Properly designed units with tube inserts normally are much smaller in size and have smaller or equal pressure drops as conventional plain tube units.

Reboilers:

1. Substitution of a plain U-tube bundle with a *low finned tube* bundle or a *high performance boiling* tube bundle (such as one of the Turbo-B versions) to increase thermal capacity;

2. Installation of an insert in the heating fluid stream (tube-side) to increase horizontal reboiler heat duties, especially for hot effluent gases in feed effluent heaters;

3. Replacement of a plain tube bundle with an *enhanced tube bundle* that allows a cheaper, lower temperature heating stream to be used if available (such as low pressure steam instead of medium or high pressure steam);

4. For highly fouling, difficult to clean plain tube horizontal thermosyphon and kettle reboilers, substitution with a *low finned tube bundle* with a larger tube pitch and the same external surface area can reduce labor costs for heat exchanger cleaning substantially;

5. For reboiler-condensers in refrigerated process units, boiling and condensing augmentations can be used to decrease the log-mean-temperature-difference of these units and reduce the compression power required, saving as much as 5-10% of the power costs for these units.

6. *Internal microfins* are very effective for horizontal tube-side evaporators over the entire range of vapor qualities and particularly at low mass velocities.

Condensers:

1. Substitution of a plain U-tube bundle of an overhead condenser with a *low finned tube bundle* (or installation of inserts in an existing tube bundle) to increase the overall heat transfer coefficient, which reduces the distillation tower operating pressure and temperature, and saves energy in heating the tower's feed to a lower operating temperature;

2. Using *low finned tubes* as opposed to plain tubes can often reduce the number of shells in multiple shell condensers by a factor of two, especially for condensing multi-component vapors on the shell-side.

3. *Internal microfins* are very effective for horizontal tube-side condensers over the entire range of vapor qualities and particularly at low mass velocities.

Air Coolers:

1. For cooling viscous fluids in air-cooled heat exchangers, tube inserts inside high finned tubes can significantly reduce the number of parallel units and the plot size required (some air cooler manufacturers have already been building units with wire mesh type inserts, for instance);

2. Installing tube inserts can increase the cooling capacity of some existing air coolers during hot summer periods and also may alleviate the need for installation of additional cooling tower capacity in marginal cases.

In the hydrocarbon processing industries, some actual applications include:

1. Coolers (gas oil, glycols, atmospheric resids, phenol, refinery light ends, sulfinol, air-cooled lubricating oil, heavy wax distillates, CO gas, citric acid, fatty acids, Therminol-66 and many other fluids);

2. Heaters/pre-heaters (crude bottoms, acetylene, exhaust gases, polymers, naphthalene, post-critical ethylene, butyl rubber, solvents, styrene, cumene, heavy tar oil and crude oil);

3. Chillers and vaporizers (liquefied gases, refrigerants, light hydrocarbons, xylenes, MDEA, alcohols, ethylene glycol, glycerine);

4. Condensers (alcohols, tower overheads, multi-component vapor mixtures, refrigerants).

Other important applications include adhesives (heating and cooling of various process streams in their production) and plastics (heating and cooling of polymer melts such as PVC, acetate and polystyrene). Other applications are found in vacuum distillation units, lube oil plants and heavy oil production facilities, where many viscous fluids are present and laminar flows persist.

2.7 Air-Separation and Liquified Natural Gas Plant Applications

Plate-fin heat exchangers are widely used in these facilities. While plate-fin exchangers allow very close temperature approaches, their long and tortuous flow paths can have larger pressure drops than shell-and-tube units. Thus, the real gain in overall system efficiency must be looked at. *Enhanced boiling tubes* and *enhanced condensing tubes* are used to advantage in these facilities in their refrigeration system reboiler-condensers, with the heating fluid-side also augmented with fins.

2.8 Applications to Lubricating Oil Coolers

Intube heat transfer augmentations are particularly suited to increasing heat transfer coefficients in lubricating oil coolers. These coolers normally come as part of package or skid mounted units for compressors, turbines, motors, engines, etc. or are mounted on large construction equipment. Hence, a lighter weight, more compact cooler has many economical advantages. Normally an increase in tube-side heat transfer performance is obtained with an insert while meeting the same pressure drop limitations as for the plain tube unit. This is achieved by using fewer tube passes.

2.9 Power Plant Operations

Integral low finned tubes are becoming widely used in power plant main condensers. The external fins for condensing steam must be optimized for the right fin density and height. Use of external fins shifts the controlling thermal resistance to the cooling waterside. Thus, low finned tubes with internal ribs are particularly suitable for this application, especially when utilizing alloys. Corrugated tubes are also beneficial in these applications.

Installation of inserts on the tube-side (i.e. water-side) of an existing steam condenser (plain or low finned tube) increases the overall heat transfer coefficient and the vacuum in the steam chest; thus the steam turbine's power output can sometimes be increased by 0.5-2%, which represents very significant savings in fuel or additional capacity to a power utility at a minimum cost.

Fossil fuel power boilers are designed with internally ribbed tubes by some steam generator manufacturers. The swirl flow created by the ribs keeps the tube wall better wetted in this asymmetrical heat flux environment, which increases the critical heat flux before passing into the film boiling regime. Twisted tape inserts can be installed in tube sections prone to this problem in existing plain I.D. units.

2.10 Geothermal and Ocean-Thermal Power Plant Applications

The viability of geothermal and ocean-thermal power plants depends in part on the use of small temperature approaches in the heat exchangers in order to increase the cycle efficiency. Low temperature approaches result in low log-mean-temperature-differences and thus large heat exchanger surface areas. To minimize the size and cost of these units, typically built with expensive alloys such as titanium or 316-grade stainless steel, heat transfer augmentations are used in both the evaporator and the condenser.

For boiling on the outside of the tube bundle in the evaporator, low finned tubes or high performance boiling geometry tubes (such as Turbo-B) should be used. Since the shell-side is only in contact with the non-corrosive working fluid (a refrigerant or propane), the external enhancement does not have to be a high alloy.

For shell-side condensing, a low finned tube or an enhanced condensing tube such as Turbo-CSL is an appropriate choice. For the former, the fin density (fins/m), fin height and thickness should be optimized, choosing among the geometries commercially available in that particular material (refer to Wolverine Tube product tables).

Enhanced evaporators and condensers enjoy much larger tube-side water velocities, which increases the internal heat transfer coefficient and reduces fouling and scale formation. Table 2.1 shows the results for an actual geothermal condenser optimized using ENHANCED HEAT TRANSFER, a software program written by J.R. Thome and licensed throughout the world from HTRI. The tube length was fixed by the application (portable truck transported unit) to 9.9 m (32.5 ft). The number of tubes (and shell) was significantly reduced in size. The design has a higher factor of reliability because the ratio of the clean overall heat transfer coefficient to the fouled value is much larger, 1.62 compared to 1.34, respectively.

Table 2.1. Geothermal Condenser Design Case Study with 316L Stainless Tubing.

Tube Type	Plain	19 fpi Low Fin	28 fpi Low Fin
Heat Duty (MW)	12.1	12.1	12.1
LMTD (°C)	10.0	10.0	10.0
Tube-Side Water Flow Rate (kg/s)	489	489	489
R-11 Flow Rate in Shell (kg/s)	68	68	68
Condensing Pressure (bar)	1.65	1.65	1.65
Tube Nominal O.D. (mm)	19.05	19.05	19.05
Tube I.D. (mm)	17.45	14.63	15.75
Tube Length (mm)	9900	9900	9900
Water Fouling Factor (m^2K/W)	0.000176	0.000176	0.000176
Fin Height/Thickness (mm)	No fins	1.5/0.3	0.94/0.3
Water Velocity in Tubes (m/s)	1.19	2.14	1.97
Number of Tubes Required	1720	1363	1280
Tube-Side Heat Transfer Coeff. (W/m^2K)	5790	9713	9034
Shell-Side Heat Transfer Coeff. (W/m^2K)	3178	5831	6393
Overall Heat Transfer Coeff. (W/m^2K)	1333	1691	1799
Ratio of $U_{o,finned}/U_{o,plain}$	1.0	1.29	1.35
Decrease in Required Length	0%	-22%	-26%
Ratio of $U_{o,clean}/U_{o,dirty}$	1.34	1.63	1.62

2.11 Applications in the Food Processing Industries

In the food processing industries, heat transfer augmentations are of particular benefit to pasteurizing temperature-sensitive foodstuffs and for heating or cooling viscous fluids, such as vegetable oils. Non-Newtonian fluids, such as power law fluids, can be augmented using tube inserts for laminar flows or integral tube enhancements, such as ribbed tubes, for turbulent flows.