1 of 1

Performance Comparisons of Enhanced Tubes with Discrete and Wavy Disruption Shapes*

B. Arman and T.J. Rabas

Energy System Division
 Argonne National Laboratory
 Argonne, IL 60439

The submitted manuscript has been authored by a contractor of the U. S. Government under contract No. W-31-109-ENG-38. Accordingly, the U. S. Government retains a nonexclusive, royalty free license to publish or reproduce the published form of this contribution, or allow others to do so, for U. S. Government purposes.

RECEIVED
 AUG 26 1993
 OSTI

ABSTRACT

This paper presents comparisons of the friction factors and heat-transfer coefficients obtained with enhanced tubes with transverse discrete and almost transverse wavy two-dimensional disruptions. Both experimental data and numerical predictions were used for the comparisons. For the latter a two-layer turbulence model incorporated in a body-fitted, finite-volume method was used.

The disruption shape, discrete or wavy, depends on the manufacturing process. If an extrusion process is used, discrete disruptions (ribs) of various profiles are obtained that are separated from each other by a flat or unaltered inside diameter. If a spirally indenting process is used, a wavy profile is obtained with a continuously varying inside diameter between two adjacent disruption peaks. These disruptions are transverse or almost transverse to the tube axis and separated by a distance that exceeds the reattachment length.

Based on these comparisons, the following conclusions are obtained: (1) the disruption shape is not an important correlating parameter for discrete disruptions, (2) only the friction factor is influenced by the shape for wavy disruptions, and (3) there are major differences between both the friction-factor and heat-transfer performance of discrete and wavy disruptions with the same maximum disruption height and spacing. However, the most important finding is that the groove radius of spirally indented tubes should be increased because of the substantial reduction of the friction factor but only a small decrease in the thermal performance. Additional comparisons of predicted results were made to obtain a fundamental understanding of the influence of these different shapes.

NOMENCLATURE

D	maximum inside tube diameter
$e_{\max}$	maximum disruption height
$\langle e \rangle$	mean disruption height defined by Equation 1
$e(x)$	local disruption height
f	friction factor, $f = \Delta p / (0.5 \rho U^2)$

h	heat transfer coefficient
k	thermal conductivity
Nu	Nusselt number, $Nu = hD/k$
p	pressure
p	pitch
Pr	Prandtl number
r_1	groove radius
r_2	circular sector radius, Equations A1 and A2
R	maximum tube radius
Re	Reynolds number
U	mean velocity
x	axial distance
w	disruption width

Greek Letters

α	helix angle
β	contact angle of circular sector, Equation A1
ρ	density
τ_w	local wall-shear stress

Subscripts

p	smooth tube
---	-------------

INTRODUCTION

The turbulent, forced-convective heat-transfer performance can be increased by a factor of two to three with disruptions of various shapes on the inside surfaces of heat-exchanger tubes. A commonly used geometry consists of disruptions that are transverse or nearly transverse to the tube axis and with a spacing equal to or greater than the reattachment length. This geometry is one of the most used for single-phase forced-convection applications such as power-plant condensers in the U.S. (Rabas et al., 1990), the U.K. (Rowe, 1983), and the former Soviet Union (Katsman and Kishinevsky, 1983; Nikolaev et al., 1982; Anisimova et al., 1982). This tube type is also used for feedwater-heater trains of desalination plants (Houle and Buhig, 1973; Inoue et al., 1975). The enhancement mechanism characterized by these geometry constraints is called separation and reattachment (Rabas, 1989).

The disruption shape of these enhanced tubes is dependent on the manufacturing process. If a spirally indenting process is used, a wavy disruption profile is obtained with a continuously varying inside diameter between two adjacent disruption peaks. The wavy profiles

* Work supported by the U.S. Department of Energy, Assistant Secretary for Conservation and Renewable Energy, Office of Industrial Technologies, Office of Industrial Processes, Advanced Industrial Concepts Division, under contract W-31-109-Eng-38.

MASTER

depend on the shape of the roller, the tube material, and the tube wall thickness. If an extrusion process is used, discrete disruptions (ribs) of various profiles are obtained that are separated from each other by a flat or unaltered inside diameter. Figures 1a and 1b show typical spirally indented and extruded tubes. The spirally indented tubes are the most used for the above mentioned applications although tubes with discrete disruptions are widely used in the refrigeration industry.

Figure 1 a) Spirally Indented Tube; b) Extruded Tube

The tube-geometry parameters that govern the thermal-hydraulic performance of these tubes with the separation and reattachment mechanism are the disruption maximum height, $e_{\max}$, pitch, p , and shape (see Figure 1). However, the major focus of previous investigations was to determine the effect of the disruption height and spacing and to neglect the effect of the shape. As a result, there are many prediction methods that adequately account for these two parameters provided that the tubes used to generate the data base were made by the same manufacturing process and have essentially the same disruption shape.

The next section of this paper will present a review of the literature that addresses the effect of the shape on the thermal-hydraulic performance of tubes with discrete and wavy disruption shapes.

PERFORMANCE COMPARISONS

The effect of the shape on the performance of tubes with the separation and reattachment mechanism has not been extensively investigated. However, there is sufficient data and information to reach some major conclusions. Comparisons of the existing performance data will be presented for tubes with different discrete shapes, different wavy shapes, and discrete and wavy shapes with the same maximum disruption height and pitch.

Discrete Shapes

Only the two investigations of Nunner (1956) and Hijikata and Mori (1987) made comparisons of experimental data for different discrete disruption shapes of enhanced tubes: rectangular, arc, and sine shapes of the discrete type. The general trend is that both the friction-factor and heat-transfer values decrease as the curvature of the disruption increases; however, the heat-transfer reduction is less than that of the friction factor. Hijikata and Mori (1987) showed that differences in the local heat-transfer measurements existed in the downstream recirculation regions but not in the boundary-layer development regions after the reattachment. It is apparent that the influence of the shape on the heat-transfer and friction-factor values decrease as the pitch increases or as the ribs are moved further apart.

Of the numerous empirical prediction methods, only one (Ravigururajan and Bergles, 1985) was discovered that considers the effect of discrete disruption shapes — rectangular, triangular, and circular (wire inserts). This shape correction was included in the friction-factor correlation but the mean heat-transfer coefficient was assumed to be independent of the shape. Based on these shape

options, the friction-factor prediction method appears to be restricted to just discrete disruptions; however, it was also recommended for wavy surfaces.

There were two recent publications (Arman and Rabas, 1992a, 1992b) that investigated the effect of different shapes, but only of the discrete type using a numerical prediction method. Arman and Rabas (1992a) showed that the thermal-hydraulic performance was not significantly changed with different disruption shapes with the same base dimension. The shapes considered were sine, arc, semicircle, and trapezoid. The friction factors decreased when the shape changes in the following order: sine, trapezoid, semicircle, and arc. For the heat transfer, the trapezoid rather than the sine shape yielded the highest value. At a Reynolds number of 39,000, the maximum variations of the friction-factor and heat-transfer values for the four shapes were about 16% and 7%, respectively. These variations were less at a lower Reynolds number. Based on these numerical predictions, it appears that the thermal-hydraulic performance are not strongly shape dependent because of these small changes.

Arman and Rabas (1992b) also compared the thermal-hydraulic performance for rectangular disruption shapes with width-to-height ratios ranging from 0.5 to 2.0. There was a small decrease of the friction factor with an increase in the width-to-height ratio and this difference increases with an increasing Reynolds number. The maximum deviation was about 9% at a Reynolds number of about 49,000. The mean heat-transfer coefficient decreased slightly with an increase in the width-to-height ratio for low Prandtl number fluids (air). However, the trend is more complex for higher Prandtl number fluids; it can either increase or decrease with an increase in the width-to-height ratio depending on the magnitude of the Prandtl number and the rib spacing. The maximum variation in the thermal performance, about 15%, occurred for $Pr = 21.5$ and $p/e_{\max} = 10.0$, the highest Prandtl number considered and for the closest spacing. This finding is consistent with previous findings in that the influence of the rib on the total performance increases as the Prandtl number increases (Rabas and Arman, 1992) and as the ribs are moved closer together. Because of the small deviations in both the friction factors and heat-transfer coefficients, the rib width does not appear to be a necessary correlating parameter for this particular discrete disruption shape.

Wavy Shapes

Only one publication, Kalinin et al. (1991), presented comparisons of experimental data obtained with different disruptions shapes formed by a spirally indenting process. They measured the friction factors and heat-transfer coefficients for 16 different symmetric profiles using water as the test fluid. The shape of the profiles were characterized by a dimensionless parameter, $\langle e \rangle / e_{\max}$, defined as

$$\frac{\langle e \rangle}{e_{\max}} = \frac{1}{(e_{\max}/p)} \int_0^p e(x) dx \quad (1)$$

The range of $\langle e \rangle / e_{\max}$ was considerable, varying from about 0.3 to 0.6. For discrete disruptions, $\langle e \rangle / e_{\max}$ is almost always less than about 0.1. A surface profile of $\langle e \rangle / e_{\max} = 0.6$ is very different from a discrete disruption. For an equivalent rectangular rib ($e = e_{\max}$) with a height of e and $p/e_{\max} = 10$, the width-to-height is $6e_{\max}$ with a spacing between the ribs of $4e_{\max}$. This profile begins to resemble internal grooves with a very wide rib.

Kalinin et al. (1991) discovered that there was a strong dependence of the hydraulic resistance on the shape or $\langle e \rangle / e_{\max}$ but the thermal performance was almost independent of the shape. In particular, the friction factor was reduced by 25 - 40% when $\langle e \rangle / e_{\max}$ was increased from 0.3 to 0.6. Figure 2 taken from this work shows the type of variation that is obtained for the friction factor with this shape change for $e_{\max}/D = 0.04$ and $p/e_{\max} = 10$.

Kalinin et al. (1991) claimed that the pressure drop is reduced with the wider shapes in a similar manner to the pressure-drop

reduction for flow through an orifice. For an orifice, the pressure drop is reduced with a larger radius of curvature and with an increased length of the orifice because of the reduced flow contraction and resulting mean velocity. This analogy is not totally correct because of the interactions of adjacent disruptions and because the disruption heights of enhanced tubes are typically only an order of magnitude larger than the laminar boundary-layer thickness whereas an orifice extends far into the turbulent core.

Figure 2 Variation of Friction Factor Ratio with Disruption Shape (Kalinin et al. 1991); $e_{max}/D = 0.04$ and $p/e_{max} = 10$

Kalinin et al. (1991) also claimed that indented tubes should be designed with the maximum possible $\langle e \rangle / e_{max}$ value. They argued that this configuration leads to the lowest pressure drop and yet does not cause any measurable reduction in the thermal performance.

Discrete and Wavy Shapes

Comparisons of friction-factor and heat-transfer data are now presented for different shapes — discrete and wavy. Figures 3 and 4 show the friction-factor and heat-transfer values, respectively, for a tube geometry with essentially the same maximum disruption height, $e_{max}/D = 0.02$ and $p/e_{max} = 20$. The test fluid was water (Pr in the 5 to 6 range). These geometry values and fluid were picked because they permit comparisons of experimental data from six different sources: one for a discrete shape (Webb et al., 1971) and five wavy or spirally indented shapes (Li et al., 1982; Gupta and Raja Rao, 1979; Mehta and Raja Rao, 1979; Withers, 1980). Table 1 lists the tube geometry for these six tubes. A rectangular rib with a width-to-height ratio of about 0.5 was selected by Webb et al. (1971).

Table 1 Tube Geometries Considered for Comparisons

Source	D(mm)	e_{max}/D	p/e_{max}	Shape
Webb et al. (1971)	36.83	0.020	20	rectangular, $w/e=0.52$
Li et al. (1982)	17.81	0.024	17.52	spirally indented
Mehta and Raja Rao (1979)	15.90	0.02	19.84	"
Gupta and Raja Rao (1979)	25.00	0.02	20.00	"
Withers (1981)	20.45	0.024	19.46	"
Withers (1981)	18.74	0.016	22.07	"

Figure 3 shows that the measured friction-factor values for the tube with rectangular discrete disruptions are substantially higher than the friction factor values obtained with the tubes made by the spirally indenting process. Also note that the friction factor for the transverse ribbed tube continuously increases and reaches an asymptotic value with increasing Reynolds number whereas the friction factors for the spirally indented tubes continuously decreases with an increasing Reynolds number. There are some deviations in the friction-factor values for the four different spirally indented tubes because the maximum disruption height and pitch values and the measuring procedures used for each experiment were not identical. The friction-factor values for the spirally indented tubes are about 60% less at a

Reynolds numbers of 10,000 and about a factor of three less at a Reynolds number of 80,000.

Figure 3 Variation of Friction Factors with Reynolds Number; $e_{max}/D = 0.02$ and $p/e_{max} = 10$

Figure 4 Variation of Mean Nusselt Numbers with Reynolds Number

Also shown in Figure 3 are numerical predictions of the friction factors for the discrete surface and for a wavy-sine and wavy-circular surface profiles. For the details of these numerical prediction methods, see Arman and Rabas (1991, 1992a, and 1992c). The two wavy surfaces that were selected for comparisons are shown in Figure 5. Note the good agreement between the experimental and numerical predictions for both the discrete and wavy surface types. Almost no difference existed between the predicted friction factors for the two profiles shown in Figure 3. The general perception is that the wavy-circular profile is more representative of a spirally indented tube. This profile is generated with sectors of circles of different radii contacting at a point with a common slope and is the same characterization used by Kalinin et al. (1991) for the indented tubes considered in some of their investigations. The Appendix discusses this particular wavy profile in more detail.

Figure 4 compares the Nusselt numbers for the rectangular rib surface and the five spirally indented tubes. Note that the experimental values obtained by Webb et al. (1971) are again higher than that obtained with the spirally indented tubes. Also note the excellent agreement between these experimental and the numerically predicted heat-transfer coefficients, the latter being described in detail by Rabas and Arman (1992) and Arman and Rabas (1992c).

Figure 4 shows small reductions in the thermal-performance values with the spirally indented tubes from that obtained with the rectangular ribs. Also note that the magnitude of these reductions do not appear to be Reynolds number dependent. These differences are significant and can not be neglected; however, they are not accounted for with any of the published prediction methods. The reasons for the scatter of the Nusselt numbers values for these indented tubes are the slightly different tube geometries, Prandtl numbers, and experimental procedures and equipment.

Figure 4 also compares the numerically predicted heat-transfer coefficients with the data obtained with the spirally indented tubes. Again, both a wavy-sine and wavy-circular profiles shown in Figure 5 were used. Note the reasonable agreement between the predictions and the experimental results and the wavy profiles used with the numerical model did not have any significant effect on the predicted values.

Figure 5 Wall Profiles of Tubes with $p/e_{\max} = 20$

Based on these mean data, the following conclusions can now be made concerning the influence of the disruption shape for tubes with the separation and reattachment mechanism:

- 1) The disruption shape is not an important correlating parameter for discrete disruptions.
- 2) The friction factor but not the heat transfer is influenced by the shape for wavy disruptions.
- 3) There are major differences between both the friction-factor and heat-transfer performance of transverse discrete and wavy disruptions with the same maximum disruption height and spacing.

The last portion of this paper will supply some of the fundamental reasons for the second and third conclusions listed above using local values of the wall shear stress, wall static pressure, and heat-transfer coefficients obtained with a numerical modeling program. The first conclusion listed above — the reasons why discrete disruption shapes do not influence the pressure drop and the thermal performance — was already addressed by Arman and Rabas (1992a, 1992b). The second conclusion will be addressed first, that is when and why different wavy profiles have a significant effect on the friction factor but not the heat-transfer coefficient. This will be followed with an explanation for the major differences between both the friction-factor and heat-transfer performance of discrete and wavy disruptions with the same spacing and maximum disruption height.

LOCAL PERFORMANCE COMPARISONS

It is first shown with the predicted results that the wavy profile can influence the friction factor but not the thermal performance in the manner described by Kalinin et al. (1991). Finally, the reasons for the differences between the pressure-drop and heat-transfer performance of discrete and wavy disruptions are addressed. A non-orthogonal, body-fitted, finite-volume numerical code is used for the present numerical simulations. Details of the numerical code is given in Arman and Rabas (1992c). The turbulence model used is the two-layer model of Chen and Patel (1988).

Influence of the Groove Radius

A wavy surface profile must first be selected to represent that of a spirally indented tube. The selected profile for this analysis consists of two circular sectors that connect with a common slope. This representation of a wavy surfaces was previously used by Kalinin et

al. (1991) and also was recommended for this analysis by Thors (1993) of Wolverine Tube, Inc. Figure 6 shows three wavy profiles with groove radius to tube diameter ratios of $r_1/D = 0.01, 0.10$, and 0.25 . The groove radius, r_1 , is being the middle circular sector formed by the indenting tool. The equations used to generate these profiles are presented in the Appendix. Comparisons of the mean and local performance will now be presented for these three groove radii.

Figure 6 Wall Profiles of Tubes Considered

Consider first the mean performance. Figures 7 and 8 show the friction factors and heat-transfer coefficients, respectively, for the same disruption height but different pitch values and the three groove radii. The disruption-tube diameter ratio, $e_{\max}/D$, is 0.02 and the fluid is water ($Pr = 5.72$). Note that substantial differences can exist with the friction factors but almost no differences exists between the heat-transfer coefficients for each pitch value. These findings are consistent with the experimental findings of Kalinin et al. (1991). However, note that the differences decrease as the pitch increases. This finding can be explained by the fact that the variations of $\langle e \rangle / e_{\max}$ decrease with increasing $p/e_{\max}$ values as shown in the Appendix. These figures clearly demonstrate that the energy efficiency can be significantly increased through the use of larger groove radii, especially for operation in the higher Re range. For typical cooling-water applications, the Re values are always in this range or about $50,000$. The variations in the friction-factor values with the different groove radii decrease as the disruption height increases; as a result, the maximum benefit are obtained with smaller disruptions.

Figure 7 Variation of Friction Factors with Reynolds Number; $e_{\max}/D = 0.02$

Consider next why there is a significant reduction of the friction factor but no major change of the heat-transfer performance as the groove radius is increased. Comparisons of the local wall pressure, shear stress, and heat-transfer coefficient distributions will be used to obtain these explanations. Conditions selected for the comparisons are as follows: the disruption height-to-diameter ratio is 0.02 , the Reynolds and Prandtl numbers are about $40,000$ and 5.5 , respectively, and the pitch is $10e_{\max}$. The four groove radius-diameter ratios are used: $0.01, 0.10, 0.17$ and 0.25 . First, the reason for the friction-factor reduction is explained.

The wall-pressure variations for the four different groove radii are shown in Figure 9. The entire wall pressure field is affected by

the wave shape. The pressure drop around the wave crest increases sharply with a decrease in the groove radius. This pressure gradient is caused by the fluid acceleration and the magnitude of this reduction is a measure of the form-drag loss. Note that the magnitude of this wall-pressure drop is reduced with a larger groove radius or with a more contoured shape. There is a greater pressure recovery away from the crest for the tube with the larger groove radius.

Figure 8 Variation of Mean Nusselt Numbers with Reynolds Number; $c_{\max}/D = 0.02$

More difficult to explain is the reason why there is no major reduction in thermal performance with the larger groove radii. It will be shown that the heat-transfer on the upstream side of the disruption is increased to compensate for the reductions at the peak (crest) region and in the downstream recirculation region. Before comparing distributions of the local heat-transfer coefficients for different groove radii, it is very helpful to analyze the influence of the groove radius on the wall shear stress.

Figure 9 Axial Variation of Wall-Pressure Drops with $c_{\max}/D = 0.02$ and $p/c_{\max} = 10$

Figure 10 shows the normalized wall shear-stress distributions for the four groove radii. There is only one recirculation zone for all four of the shapes. The upstream zero value of the wall shear stress is the location of the reattachment point, at $x/c_{\max}$ equal to about 1 to 1.5. With an increase of the groove radius, this figure clearly shows that the wall shear stress becomes more uniform and the size of the recirculation region is reduced. The size of the recirculation regions are $5.68c_{\max}$, $5.01c_{\max}$, $4.55c_{\max}$ and $3.71c_{\max}$ for groove radii of 0.01D, 0.10D, 0.17D and 0.25D, respectively.

Of particular interest is the change in the behavior of the shear stress at the front half of the disruption. For this region of positive shear stress ($x/c_{\max}$ in about the 1 to 6 range), the maximum moves upstream from the wave crest and its magnitude becomes smaller with increasing groove radius. This also suggests that the location and magnitude of maximum heat transfer will move accordingly. The locations of minimum wall-shear stresses in the recirculation region will move further downstream with increasing groove radius. This implies that the heat transfer peaks within the recirculation region will also move further downstream. Also note that the tube with groove radius of 0.25D has a very strong second wall-shear stress minimum just before the reattachment point.

Figure 10 Axial Variation of Wall-Shear Stresses with $c_{\max}/D = 0.02$ and $p/c_{\max} = 10$

Figure 11 shows that distributions of the local heat-transfer coefficient for the same four groove radii. First consider the recirculation region. The high heat-transfer enhancement in the recirculation region is the result of kinetic energy generated in the shear layer due to the separation process and convected to the surface near the reattachment point (Mori and Daikoku, 1972). Numerical predictions (Rabas and Arman, 1992 and Arman and Rabas, 1992c) suggest that the location of this maximum heat transfer is located between the point of maximum negative wall shear stress and the reattachment point. The location of maximum heat transfer will move further downstream with increasing groove radius and is located between the shear stress minimum and the reattachment point, closer to the shear stress minimum. However, the movement of the heat-transfer peak is a function of Reynolds and Prandtl numbers (Arman and Rabas, 1992c). Figure 11 shows that the tubes with a larger groove radius will not achieve as high a contribution to the total heat transfer from the recirculation region. However, the size of the recirculation region is reduced with larger groove radius so the impact of this effect is minimized.

Now consider the wave crest region. Figure 11 shows that the local Nusselt number distributions show a maximum for each profile and that the location moves further upstream from the wave crest with an increasing groove radius. This is similar to the maximum of the wall-shear stress. The location of the heat-transfer maximum corresponds exactly with the shear-stress maximum for the tube with the smallest groove radius (0.01D); however, the location moves increasingly further upstream from the location of the shear-stress maximum with increasing groove radius. For the tube with groove radius of 0.25D, the location of maximum is moved about to the reattachment point and the second shear-stress minimum. The dividing free shear layer between the recirculation zone and the core region will impact on the second wave crest. Thus this increased turbulence level causes the high heat-transfer performance. With an increase in the groove radius, Figure 11 clearly shows that the heat-

transfer reduction in the smaller recirculation region is compensated by the increasing contribution from the wave crest region.

Figure 11 Axial Variation of Nusselt Numbers with $c_{\max}/D = 0.02$ and $p/c_{\max} = 10$

Wavy and Discrete Shapes

Comparisons of the local performance characteristics of a wavy and discrete disruptions are now presented for enhanced tubes with $c_{\max}/D = 0.02$, $p/c_{\max} = 10$, and Reynolds and Prandtl numbers of about 40,000 and 5.72, respectively. A discrete disruption can be considered as a limiting case of a wavy surface with a very small groove radius. As a result, much of the previous discussion on wavy surfaces is applicable for these comparison.

Figure 12 compares the wall shear stress for a trapezoidal rib (base width equals $2c_{\max}$ and a base angle of 68.2 degrees) and a wavy surface with a groove radius of 0.01D. There are three major features of the ribbed tube worthy of noting:

- 1) a negative and a very large positive peak near the front and top windward faces, respectively
- 2) larger downstream recirculation region
- 3) a small upstream recirculation region

When the shear stress is negative, which is a characteristic of recirculation flow regions, the dominant flow type controlling the heat-transfer enhancement is separating and reattaching rather than boundary-layer. Figure 12 shows that the controlling enhancement mechanism for the ribbed tube is separation and reattachment because the negative shear stress exists for most of the axial distance between two adjacent disruptions.

These features are in sharp contrast to that shear-stress profile obtained with the wavy surface. For this case, a typical boundary-layer type flow exists for the majority of the axial distance similar to that in a smooth tube. However, the magnitude is greater than the smooth-tube value in the front half of the tube because of the mean flow acceleration for most of the front half of the disruption.

The friction factor variation with the Reynolds number in turbulent flow has two limiting characteristics:

- 1) independent of the Reynolds number when the form drag is controlling — separating and reattaching flows fields

- 2) a negative slope (slope of about 0.25) when the frictional contribution is controlling — boundary-layer flow fields

For discrete disruption shapes, the friction factor is independent of the Reynolds number for most of the range as is clearly shown in Figure 3. However, the decrease in the friction factor with Reynolds number for the spirally indented tubes suggests the boundary-layer flow is more dominant than separating and reattaching flows. The local shear stress variations shown in Figure 12 demonstrate that this is indeed the case — separating and reattaching dominates with discrete disruptions and boundary-layer flow characteristics dominate with discrete disruption profiles.

Figure 12 Axial Variation of Wall-Shear Stresses for Trapezoidal Rib and Wavy Tubes

Figure 13 shows that there also are dramatic differences in the distributions of the local heat-transfer coefficients. With the discrete disruption, most of the enhancement occurs in the vicinity of the disruption and in the reattachment region and almost no enhancement is obtained upstream of the disruption. The superior local thermal performance of wavy profiles in this region is the explanation why the mean thermal performance (the integrated average for the entire axial distance) is comparable to that obtained with discrete disruptions.

Figure 13 Axial Variation of Nusselt Numbers for Trapezoidal Rib and Wavy Tubes

CONCLUSIONS

The most important conclusion of this investigation is that the groove radii currently being used by manufacturers of spirally indented tubes are too small. A larger groove radius would significantly reduce the pressure drop but only slightly reduce the heat-transfer performance. As a result, tubes made with a larger groove radius will be more energy efficient. This conclusion was previously reached by Kalinin et al. (1991) with experimental data and is further supported with the results from this investigation.

Some other conclusions regarding the influence of the disruption shape obtained by comparisons of data and numerical results are as follows:

- 1) The shape is not a important correlating parameter for discrete disruptions — disruptions with base widths that are less than about twice the maximum height.
- 2) The shape is a necessary correlating parameter of the friction factor but not the heat-transfer performance for wavy disruptions.
- 3) There are major differences between both the friction-factor and the heat-transfer performance values obtained with discrete and wavy disruptions with the same maximum height and pitch. The differences in the thermal performance are not large about 20 to 30%, but certainly can not be neglected.

REFERENCES

- Anisimova, O.S., et al., 1982, "A Comparative Investigation of the Resistance to Corrosion of Smooth and Profiled Brass Tubes," *Thermal Engineering*, Vol. 29, pp. 452-456.
- Arman B., and T.J. Rabas, 1992a, "Disruption Shape Effects on the Performance of Enhanced Tubes with the Separation and Reattachment Mechanism," *Enhanced Heat Transfer*, ASME HTD-Vol. 202, 67-76.
- Arman B. and T.J. Rabas, 1992b, "The Influence of the Rib Width on the Performance of Tubes with the Separation and Reattachment Enhancement Mechanism," *Fundamentals of Forced Convection Heat Transfer*, ASME HTD-Vol. 210, pp. 159-166.
- Arman B., and T. J. Rabas, 1992c, "Influence of Prandtl Number and Effects of Disruption Shape on the Performance of Enhanced Tubes with the Separation and Reattachment Mechanism," Argonne National Laboratory, ANL/ESD-16.
- Arman B., and T.J. Rabas, 1991, "Prediction of the Pressure Drop in Transverse, Repeated-Rib Tubes with Numerical Modeling," *Fouling and Enhancement Interactions*, ASME HTD-Vol. 164, pp. 93-99.
- Chen, H.C., and V.C. Patel, 1988, "Near-Wall Turbulence Models for Complex Flows Including Separation," *AIAA J.*, Vol. 26, pp. 641-648.
- Hijikata, K., and Y. Mori, 1987, "Fundamental Study of Heat Transfer Augmentation of Tube Inside Surface by Cascade Smooth Surface-Turbulence Promoters," *Wärme-und Stoffübertragung*, Vol. 21, pp. 115-124.
- Gupta, R.H., and M. Raja Rao, 1979, "Heat Transfer and Frictional Characteristics of Spirally Enhanced Tubes for Horizontal Condensers," *Advances in Enhanced Heat Transfer*, ASME, San-Diego, pp. 103-114.
- Houle, J.F., and W.T. Buhrig, 1973, "Performance of Freeport, Texas, VTE/MSF Plant," *4th International Symposium on Fresh Water from the Sea*, Vol. 1, pp. 313-318.
- Inoue, S., T. Tsuruta, and S. Funhashi, 1975, "Multiple Effect VTE Desalination Plant Experiences in Operation and Performance 500 Ton/Day Plant at Innoshima Shipyard," *Heat Transfer Japanese Research*, Vol. 4, pp. 1-19.
- Kalinin, E.K., G.A. Dreister, N.V. Paramonov, A.S. Myakochin, A.I. Tikhonov, S.G. Zakirov, E.S. Levin, L.S. Yanovsky, 1991, "Comprehensive Study of Heat Transfer Enhancement in Tubular Heat Exchangers," *Experimental Thermal and Fluid Science*, Vol. 4, pp. 656-666.
- Katsman, D., and Y. Kishinevsky, 1983, "Spiral, High Strength, Corrosion-Resistance Tube Application in PWR Power Plant Condensers," ASME Paper 83-JPGC-Pwr-17.
- Li, H.M., K.S. Ye, Y.K. Tan, and S.J. Den, 1982, "Investigation of Tube-Side Flow Visualization, Friction Factors and Heat Transfer Characteristics of Helical-Ridging Tubes," *Proceeding of the Seventh International Heat Transfer Conference*, Hemisphere Publishing Co., Washington, D.C., Vol. 3, pp. 75-80.
- Mehta, M.M., and M. Raja Rao, 1979, "Heat Transfer and Frictional Characteristics of Spirally Enhanced Tubes for Horizontal Condensers," *Advances in Enhanced Heat Transfer*, San Diego, pp. 11-22.
- Mori, Y., and T. Daikoku, 1972, "Effect of 2-Dimensional Roughness on Forced Convective Transfer," *Bulletin of the JSME*, Vol. 15, pp. 1581-1590.
- Nikolaev, G.V., V.V. Nazarov, G.V. Grigorev, and Y.N. Bogolyubov, 1982, "Comparative Studies of a 200 KT's-2 Condenser with Smooth and Rolled Tubes," *Energomashinostroenie*, Vol. 1, pp. 15-16.
- Nunner, W., 1956, "Heat Transfer and Pressure Drop in Rough Tubes," A.E.R.E. Lib/Trans. 786 (1958).
- Rabas, T.J., 1989, "Selection of the Energy-Efficient Enhancement Geometry for Single-Phase Turbulent Flow Inside Tubes," *Heat Transfer Equipment Fundamentals, Design, Applications and Operating Problems*, ASME HTD-Vol. 108, pp. 193-204.
- Rabas, T.J., and B. Arman, 1992, "The Influence of the Prandtl Number of the Thermal Performance of Tubes with the Separation and Reattachment Mechanism," *Enhanced Heat Transfer*, ASME HTD-Vol. 202, 77-88.
- Rabas, T.J., R. Merring, R. Schaefer, R. Lopez-Gomez, and P. Thors, 1990, "Heat-Rate Improvements Obtained with the Use of Enhanced Tubes in Surface Condensers," EPRI Condenser Technology Conference, Boston, MA.
- Ravigururajan, T.S., and A.E. Bergles, 1985, "General Correlations for Pressure Drop and Heat Transfer for Single-Phase Turbulent Flow in Internally Ribbed Tubes," *Augmentation of Heat Transfer in Energy Systems*, ASME HTD-Vol. 52, pp. 9-20.
- Rowe, M., 1983, "Power Plant Condensers - Recent CEGB Experience," *Condensers: Theory and Practice*, The Institution of Chemical Engineers, Symposium Series No. 75, pp. 113-134.
- Thors P., 1993, Personal Communication.

Webb, R.L., E.R.G. Eckert, and R.L. Goldstein, 1971, "Heat Transfer and Friction in Tubes with Repeated-Rib Roughness," *International Journal of Heat and Mass Transfer*, Vol. 14, pp. 601-607.

Withers, J.G., 1980, Tube-Side Heat Transfer and Pressure Drop for Tubes Having Helical Internal Ridging with Turbulent/Transitional Flow of Single-Phase Fluid, Part 1, Single-Helix Ridging," *Heat Transfer Engineering*, Vol. 2, pp. 48-58.

APPENDIX

The purpose of this Appendix is to present the geometry variation expressions selected to represent spirally-indented or wavy tube profiles. These surfaces consist of two circular sectors that connect with a common slope. Let β be the contact angle of the two circles, r_1 the external groove radius, and r_2 the radius of the other circular sector. It is easy to show the following:

$$p = 2 [r_1 \sin(\beta) + r_2 \sin(\beta)] \quad (A1)$$

$$e_{\max} = r_1 [1 - \cos(\beta)] + r_2 [1 - \cos(\beta)] \quad (A2)$$

$$\frac{p}{e_{\max}} = \frac{2 \sin(\beta)}{[1 - \cos(\beta)]} \quad (A3)$$

The three independent variables are the disruption pitch (p), maximum height ($e_{\max}$), and the groove radius (r_1). The tangent contact angle, β , is calculated implicitly with Equation A3 and r_2 is then calculated with either Equations A1 or A2. Knowing these quantities, $e(x)$ can be calculated for any value of x in a straightforward manner and will not be discussed.

The ratio $\langle e \rangle / e_{\max}$ defined in Equation 1 can be determined for different values of $e_{\max}/D$, $p/e_{\max}$ and r_1/D for this wavy profile type. Table A1 shows the variations of $\langle e \rangle / e_{\max}$ with typical $e_{\max}/D$, $p/e_{\max}$ and r_1/D values. The important point is that small $e_{\max}/D$ and $p/e_{\max}$ values and/or large changes in r_1/D are needed to obtain significant variations in the values of $\langle e \rangle / e_{\max}$.

Table A1 $\langle e \rangle / e_{\max}$ Variation with $e_{\max}/D$, $p/e_{\max}$, and r_1			
$e_{\max}/D$	$p/e_{\max}$	r_1/D	$\langle e \rangle / e_{\max}$
0.01	10.0	0.01	0.355
		0.10	0.593
		0.25	0.829
	20.0	0.01	0.339
		0.10	0.399
		0.25	0.498
	40.0	0.01	0.335
		0.10	0.350
		0.25	0.375
0.02	10.0	0.01	0.341
		0.10	0.460
		0.25	0.659
	20.0	0.01	0.335
		0.10	0.365
		0.25	0.415
	40.0	0.01	0.334
		0.10	0.341
		0.25	0.358
0.04	10.0	0.01	0.335
		0.10	0.394
		0.25	0.493
	20.0	0.01	0.334
		0.10	0.349
		0.25	0.374
	40.0	0.01	0.334
		0.10	0.337
		0.25	0.343

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

**DATE
FILMED**

11 / 9 / 93

END

100