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Disruption Shape Effects on the Performance of Enhanced Tubes with the Separation and Reattachment Mechanism*

B. Arman and T. J. Rabas

Energy Systems Division
Argonne National Laboratory
9700 S. Cass Avenue
Argonne, IL 60439

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ABSTRACT

A non-orthogonal, body-fitted numerical code is used to determine the thermo-hydraulic performance of enhanced tubes with transverse, periodic sine-, semicircle-, arc-, and trapezoid-shaped disruptions. The turbulence closure was achieved with a two-layer turbulence model.

It is shown that there is a tradeoff of the heat-transfer and pressure-drop performances when the disruption shape becomes more contoured; that is, both the heat transfer and the pressure drop increase. The local heat transfer is strongly dependent on the shape in the vicinity of the disruption but it is less dependent in the downstream recirculation region and in the boundary layer development zone. With increasing pitch, the effect of the shape on the heat-transfer performance becomes less important. The pressure drop is more dependent on the disruption shape and it continues to decrease when the disruptions become less contoured because of the reduced form drag which is by far the major contribution to the total pressure drop.

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It is shown that there is a tradeoff of the heat-transfer and pressure-drop performances when the disruption shape becomes more contoured; that is, both the heat transfer and the pressure drop increase. The local heat transfer is strongly dependent on the shape in the vicinity of the disruption but it is less dependent in the downstream recirculation region and in the boundary layer development zone. With increasing pitch, the effect of the shape on the heat-transfer performance becomes less important. The pressure drop is more dependent on the disruption shape and it continues to decrease when the disruptions become less contoured because of the reduced form drag which is by far the major contribution to the total pressure drop.

NOMENCLATURE

c_p	specific heat
D	tube inside diameter
e	rib height
f	friction factor
h	heat transfer coefficient
$h(x)$	local heat transfer coefficient
k	turbulent kinetic energy
k	thermal conductivity
m	total flow rate
Nu	Nusselt number, $Nu = hD/k$
$Nu(x)$	local Nusselt number, $Nu(x) = h(x)D/k$
p	pressure
p	disruption or rib pitch
Pr	Prandtl number
r	radial distance
Q	total heat added per pitch
Re	Reynolds number based on tube diameter, $Re = \rho UD/\mu$
St	Stanton number, $St = Nu/(RePr)$

T	temperature
u	axial velocity
U	average velocity
w	disruption width
x	axial distance
y	normal distance from the wall
y^+	wall coordinate, $y^+ = y\sqrt{\tau_w/\rho\nu}$

Greek Letters

ϵ	turbulent energy dissipation
η	Nusselt number ratio, Nu/Nu_p
μ	viscosity
ν	diffusivity
ρ	density
τ_w	wall shear stress
Δp	pressure drop
ΔT	temperature rise
ϕ	disruption contact angle (Figure 1)

Subscripts

p	plain tube
$trap$	trapezoid-shaped disruption

INTRODUCTION

An important category of enhanced tubes contains disruptions nearly transverse to the flow direction and with a spacing greater than the reattachment length. This geometry was proposed and/or used for many single-phase forced-convection applications, and is characteristic of an enhancement mechanism called separation and reattachment (Rabas, 1989).

The tube geometry parameters that control the thermal-hydraulic performance of tubes with the separation and reattachment mechanism are the disruption height, pitch, and shape. In general, there is a wide range of disruption geometry parameters that can be obtained; the major factors are the manufacturing method and/or the tubing material. The effects of the height and pitch of the disruption have been adequately determined by experiments. There are many prediction methods that do an adequate job to account for these effects provided that all the tubes used to generate the data base were made by the same manufacturing process and have essentially the same disruption shape. However, these prediction methods do not yield acceptable results when used to predict the performance of tubes made by different manufacturers and/or manufacturing processes. The shapes of the

disruptions are dependent on the manufacturing process and can vary from gentle wavy profiles to ribs with rectangular shapes.

The purpose of this paper is to demonstrate that the influence of different disruption shapes on the heat-transfer and pressure-drop performance can be predicted with numerical modeling methods. Figure 1 shows the four disruption shapes that are considered in this investigation: sine, semicircle, arc, and trapezoid.

Figure 1 Different Disruption Shapes Considered in Investigation

The first part of the paper will present a review of previous experimental investigations dealing with the influence of the disruption shape on the performance of enhanced heat-exchanger tubes characteristic of the separation and reattachment mechanism. The second part presents a review of previous, pertinent numerical investigations. A brief description of the turbulence model and the numerical method, and subsequent validation of the program is presented in the third part using data for a smooth tube and enhanced tubes with arc- and semicircle-shaped disruptions. The paper concludes with a comparison of the thermal-hydraulic performance for the disruption shapes shown in Figure 1.

EXPERIMENTAL INVESTIGATIONS

There are a few investigations that experimentally determined the effect of disruption shape on the heat-transfer and pressure-drop performance: Nunner (1956), and Hijikata et al. (1987). Also, a comparison of data from various sources with different manufacturing methods clearly displayed differences in the pressure-drop performance. Each is now discussed.

Nunner (1956)

Nunner (1956) measured the heat-transfer and pressure-drop performance of four tubes that contained transverse disruptions of different shapes. This was the first publication known to the authors that determined the differences in the performance caused by the disruption shape. For all four tubes, the disruption heights were the

Figure 2a Heat Transfer Performance Variations with Different Disruption Shapes (Nunner, 1956)

Figure 2b Friction Factor Variations with Different Disruption Shapes (Nunner, 1956)

same but two different p/e ratios and three different shapes were used. The test fluid was air. For a fixed p/e , Figure 2 shows that the Nusselt number and friction-factor values decrease with the disruption curvature; however, the reductions of the heat-transfer values are less than the reductions in measured friction-factor values.

Hijikata et al. (1987)

Hijikata et al. (1987) measured the local heat-transfer coefficients for different transverse disruptions in two experiments: rectangular channels with air, and tubes with water. Three different disruption shapes — square, sine, and arc — were installed in the rectangular channel. The same disruption height and pitch were used for all three shapes. Figure 3 shows that the thermal performance is strongly dependent on the shape in the recirculation region bounded by the disruption and reattachment point but is essentially independent of the shape downstream of the reattachment point. The symbols just above the x/e axis mark the locations of the reattachment point for the different shapes. Also note that the reattachment point moved closer to the rib as the Reynolds number increased from 25,000 to 55,500 for the arc-shaped disruption.

Figure 3 Local Heat-Transfer Coefficient Variations with Different Disruption Shapes in a Rectangular Passage (Hijikata et al., 1987)

The enhanced tubes contained periodically placed transverse disruptions of different shapes: two different arc shapes and a wing or asymmetric shape. The local turbulent fluctuations, pressure drop values, and Nusselt numbers were compared for these three tubes. A

Figure 4 Local Heat-Transfer Coefficient Variations with Different Disruption Shapes Inside an Enhanced Tube at $Re = 20,000$ (Hijikata et al., 1987)

comparison of the local heat-transfer coefficients is presented in Figure 4. Note that the arc shape leads to substantially higher heat-transfer coefficients than that obtained with the asymmetric shape for the same disruption heights. However, the wing-shaped disruption exhibited about a 23% reduction of the pressure drop.

Rabas et al. (1988)

Even for tubes of the same family such as spirally indented or rope tubes, the disruption shape affects the pressure-drop prediction. Figure 5 taken from Rabas et al. (1988) shows the friction-factor enhancement ratios at $Re = 40,000$ as a function of the severity ratio defined as $e^2/(pD)$ for 60 different tubes from eight different sources. Note that the data fall into three groupings based on the manufacturer and/or shape resulting from the indenting process: wavy (Ravigururajan and Bergles, 1985; Withers, 1980; Li et al. 1982; Yamada and Nishijima, 1980; and Young et al. 1975), semicircular with small pitch (Mehta and Raja Rao, 1988; and Gupta and Raja Rao, 1979), and semicircular with large pitch (Ganesan and Raja Rao, 1982).

Figure 5 Friction-Factor Enhancement Ratio Variations for Spirally Indented Tubes (Rabas et al., 1988)

There exists sufficient experimental data — mean and local — to demonstrate that the disruption shape is a significant parameter affecting the pressure drop and to a smaller extent the heat-transfer performance of enhanced tubes with the separation and reattachment mechanism. However, the shape is incorporated into only one prediction method that is recommended for the separation and attachment mechanism — that of Ravigururajan and Bergles (1985) — and this correction is only for the friction factor prediction. Also, this

correlation contains just a few shape options and no flexibility exists to evaluate a wide variety of different shapes.

The above discussion demonstrated the need for research to determine how the disruption shape influences the thermal-hydraulic performance of enhanced tubes with the separation and reattachment mechanism. The method selected is a numerical analysis. The major reasons for this selection are: (1) the recent progress made in numerical modeling, (2) the improvement of computer hardware capability, and (3) the relatively simple analysis — that is, a two-dimensional rather than three-dimensional analysis is needed. It was felt that other methods such as a classical analysis and recorelating of the data would not reveal the fluid-mechanics and heat-transfer details as well as the numerical method.

NUMERICAL INVESTIGATIONS

Numerical modeling has not been overly successful in predicting the heat transfer with turbulent flow inside enhanced passages. The underlying physical reason for the lack of the high-quality predictions is the complex flow field that exists as shown in Figure 6. The flow field is under the influence of adverse, favorable, and zero pressure gradients. It is well-known that the current popular engineering turbulence models suffer under the condition of varying pressure gradients (Rodi, 1991). It is also very difficult to employ a simple classical correlation approach because of the very complex nature of the flow and scalar-transport fields. Hence, the prediction capability of pressure drop, heat-transfer performance, and fouling propensity of these surfaces needs to be improved.

Figure 6 Turbulent Flow Structure within an Enhanced Tube with the Separation and Reattachment Mechanism

There have been some recent attempts to numerically predict the heat-transfer and pressure-drop performance of enhanced tubes with transverse, repeated-rib disruptions. More active areas of research have been laminar-flow and turbulent-flow investigations of just single disruptions as opposed to fully-developed flow inside an enhanced tube with large length to diameter ratios. A brief review of the germane investigations is now presented.

Hijikata et al. (1987)

Hijikata et al. (1987) presented a very short description of a numerical simulation of a fully developed flow in a rectangular channel (aspect ratio of six) with twelve pairs of disruptions, 10 mm height (1/5 the channel height), at intervals of 150 mm ($p/e = 15$). Arc-shaped disruptions were selected. Measurements of the flow field — velocities and turbulence properties — were also taken for comparison with the numerical predictions.

An improved $k-\epsilon$ model was used that prevents the dissipation rate of the turbulent energy and the vorticity from becoming unbounded at the wall. New wall functions were selected to equal zero at the wall and join with the results of the $k-\epsilon$ model at the edge of the log law region. Very few additional details of this two-layer model were given.

Comparisons of the predicted and measured results were then made. Acceptable agreement was obtained for the turbulent-energy profiles and the mean heat-transfer coefficients but not for the mean velocity profiles. The authors claimed that the major reasons for the discrepancies were the use of the logarithmic velocity profile near the

reattachment point and neglecting the turbulent-energy generation due to anisotropy of the turbulence.

Fodemski and Collins (1988)

Fodemski and Collins (1988) evaluated the fluid-flow and heat-transfer characteristics in a rectangular passage (4.0 aspect ratio) with one surface containing transverse, square ribs with a $p/e = 7.125$ and $e/D = 0.11$. The bottom of the enhanced surface was heated (constant temperature) and the coupling between the convective heat transfer to air and conduction within this solid was also considered. This computer simulation of the temperature field was carried out over ten ribs. A unique feature of this simulation was the consideration of the full conjugate problem.

The Harwell FLOW3D computer code was used. A standard $k-\epsilon$ model and standard wall functions with a finite volume method was employed with a fully developed velocity profile. To accommodate a fully developed flow field, cyclic boundary conditions were invoked. The predicted temperature field was in reasonable agreement with holographic interferometry measurements.

Arman and Rabas (1991, 1992)

Arman and Rabas (1991, 1992) investigated the effects of the disruption pitch and height on the pressure-drop and heat-transfer performance of rectangular-shaped, transverse disruptions. The disruption height-to-tube-diameter ratios were 0.01, 0.02, and 0.04, and the pitch-to-height ratios were 10, 20, and 40. The enhancement geometries were identical to that used by Webb et al. (1971) in their benchmark experiments on transverse rectangular disruptions.

The numerical model used a finite volume method with cyclic (periodic) boundary conditions in longitudinal direction. The two-layer turbulence model of Chen and Patel (1988) was employed for closure. The pressure-drop predictions were in good agreement with the experimental data. They argued that the use of logarithmic velocity profiles was in error for this geometry. An extensive comparison of the measured and numerically predicted heat-transfer coefficients for the five tubes and four Prandtl number fluids used by Webb et al. (1971) demonstrated that very satisfactory agreement was obtained. The average deviation between the measured and predicted results was 15 percent and the numerical method underpredicted the experimental results with the maximum deviation occurring at the high Re range but never exceeding about 31 percent.

The numerical modeling effort of Arman and Rabas (1991, 1992) was successful in predicting the thermo-hydraulic performance of periodic, transverse repeated-rib enhanced tubes with the separation and reattachment enhancement mechanism; however, the computer code was restricted to rectangular disruption shapes. The next section briefly describes a new computer code that uses the same turbulence model but with a new, colocated, body-fitted numerical procedure. This new code is then used to demonstrate that numerical analysis is an effective research tool to investigate the shape effects on the thermal-hydraulic performance of enhanced tubes with transverse disruptions and with pitch to height ratios greater than the reattachment length. The particular shapes that will be studied are shown in Figure 1.

NUMERICAL METHOD

Non-Orthogonal Curvilinear Code Description

This section of the paper briefly describes the modeling approach that was selected; however, a detailed description is given by Arman and Rabas (1992). The governing conservation laws yield the continuity equation, the momentum equations, and the energy equation. The transport equations for k and ϵ are used for the closure of the equation set. These governing equations are presented in many publications; therefore, they are not repeated here. The conservation equations coupled with the $k-\epsilon$ equations are the common starting point for almost all the current engineering numerical modeling efforts. The turbulence model employed is a two layer turbulence model that divides the flow domain into two regions: (1) a near-wall region that includes the sublayer, the buffer layer, and a small part of

the turbulent core and (2) the remainder of the fully developed turbulent core region. The standard $k-\epsilon$ model is used in the core region while a one-equation model is used in the near-wall region. The details of the implementation of the turbulence model is given by Chen and Patel (1988).

The non-orthogonal, body-fitted numerical program is based on the finite-volume method of Patankar and Spalding (1972). The algebraic equations are discretized by using a non-staggered, colocated mesh network. The diffusive terms are discretized by the second-order central differencing while the convective terms are discretized by the parabolic/upwind composite difference scheme of Zhu (1991). The velocity-pressure coupling is achieved through the use of the continuity equation and applying the SIMPLEC algorithm (Van Doormaal and Raithby, 1984). The resulting algebraic equations were solved by using a line-by-line, three-diagonal matrix algorithm that sweeps in both directions.

The boundary conditions at the wall are the usual no-slip conditions: $u = v = w = 0$, and there is no need to specify the wall boundary condition for ϵ . A constant wall-heat-flux boundary condition was selected for the solution of the energy equation. The axisymmetry conditions are used at the tube centerline.

The fully developed flow field is periodic in the longitudinal direction, hence periodicity conditions are used at the inlet and at the outlet of the flow domain that is equal to one disruption pitch. The fully developed periodic flow condition requires identical inlet and outlet profiles for the velocities, the turbulent kinetic energy, and the turbulent energy dissipation. The driving force for the flow is a constant pressure difference that exists between the inlet- and outlet-pressure distributions. As with the pressure profiles, the temperature profiles must be identical at the axial increments of one pitch except for a constant value displaced by an amount of $\Delta T = Q/mc_p$. For more information on the periodicity conditions and the fully developed flow constraint, see Patankar et al. (1977) and/or Arman and Rabas (1992).

During the numerical runs, the first near-wall grid point was located well within the laminar sublayer, $y^+ = 0.1$. Both iteration and grid dependency tests were performed. The iterations were continued until the normalized mass residual (total mass residuals divided by the total mass flow rate) fell below 10^{-5} for the flow-field iterations. The convergence criteria for the temperature field was also 10^{-5} for the normalized energy residual (the total heat-balance residual divided by the total heat input). However, the maximum number of iterations was set at 20,000 for the flow-field calculations and 30,000 for the temperature-field calculations. The typical underrelaxation factors of 0.5 and/or 0.3 were used for both velocities and pressure iterations. A typical run time was in the range of 6 to 24 hours on an IBM RS/6000-320 workstation, and it was dependent on the number of iterations and the grid density.

Comparison with Experimental Data

Predicted friction-factor and heat-transfer values using the new computer code are compared with experimental results. Figure 7 shows that an excellent agreement is obtained between the predictions and the smooth-tube friction-factor correlation of Techo et al. (1965) and the Stanton number prediction method of Petukhov and Popov (Petukhov, 1970). The Techo et al. (1965) friction factor correlation was selected because it is an explicit form of the classical correlation called PKN (Prandtl-Kármán-Nikuradse) (Kakaç et al., 1987). With extensive experimental measurements, its predictions agree within $\pm 2\%$. Both the friction factors and the heat-transfer coefficients were overpredicted by about 1.2% and 3.0%, respectively.

A comparison of the measured and predicted friction factor and heat-transfer values is next presented for an arc-shaped disruption. The experimental data was obtained by Hijikata and Mori (1987) and the test fluid was water. Two tube geometries were selected for this comparison: (1) $e/D = 0.02$ and $p/e = 44.4$ and (2) $e/D = 0.04$ and $p/e = 22.2$. A width to height ratio of three, $w/e = 3.0$, was selected for both tube geometries although the exact value was not specified in the paper. Figure 8a shows that the numerical friction factors are overpredicted with a maximum deviation of about 26 percent. Figure

8b shows that the Nusselt numbers are underpredicted with a maximum deviation of about 17 percent.

Figure 7 Comparison of the Numerically Predicted Friction Factor and Stanton Number Values with Those Obtained with Commonly Used Correlations

Figure 8a Comparison of the Numerically Predicted Friction Factors with Measured Values Obtained by Hijikata and Mori (1987) for an Arc-Shaped Disruption

Figure 8b Comparison of the Numerically Predicted Nusselt Numbers with Measured Values Obtained by Hijikata and Mori (1987) for an Arc-Shaped Disruption

A similar comparison was made with the data obtained by Nunner (1956) for semicircular transverse ribs with $e/D = 0.04$ and $p/e = 20$. The test fluid for these experiments was air. Figures 9a and 9b show that satisfactory agreement is obtained between the numerically predicted and experimental values.

Figure 9a Comparison of the Numerically Predicted Friction Factors with Measured Values Obtained by Nunner (1956) for a Semicircular Disruption

Figure 9b Comparison of the Numerically Predicted Nusselt Numbers with Measured Values Obtained by Nunner (1956) for a Semicircular Disruption

RESULTS

In this section of the paper, the effects of the disruption shape on the pressure-drop and the heat-transfer performance of enhanced tubes are discussed. Four different disruption shapes were selected: sine, semicircle, arc, and trapezoid.

Wall Shear Stress and Friction Factors

Figure 10 shows the normalized local wall shear stresses (i.e., skin-friction coefficients) for a sine-shaped disruption. The reattachment-point location — the point of zero shear stress — moves toward the rib with an increasing Reynolds number. A similar finding was obtained for the other three disruption shapes. This finding is in agreement with that observed by Hijikata et al. (1987) and shown in Figure 4. Note that the size of the upstream recirculation zone increases with an increasing Reynolds number. Figure 10 also shows that the upstream recirculation region is much smaller than the downstream recirculation region. There is a sharp increase in the wall shear stress on the leading face of the rib and there is a flow reversal on the rib just immediately after this pike. The strength of this flow reversal increases with an increasing Reynolds number.

Figure 10 Normalized Wall Shear Stress Distributions for a Sine-Shaped Disruption

Figures 11 and 12 compare the wall shear stress for four different shapes at Reynolds numbers of about 9,400 and 39,000, respectively. The wall shear-stress patterns in the vicinity of the rib are strongly shape dependent. The wall shear stress of the trapezoidal rib exhibits

Figure 11 Comparison of the Wall Shear Stress Variations for the Smaller Reynolds Number

Figure 12 Comparison of the Wall Shear Stress Variations for the Larger Reynolds Number

two maxima at the leading and trailing top corners. Also there is no flow reversal on the rib. The other three shapes show flow reversals on the contoured trailing faces. These figures show that the size of upstream recirculation region is not strongly dependent on the shape

of the disruption; however, the size of the downstream recirculation region is more dependent on the disruption shape.

Table 1 presents the reattachment-point locations that are defined as the distance from the end of the disruptions to the point of zero wall shear stress and the friction factors for the different disruption shapes, for two different spacings ($p/e = 10$ and 20), and for two different Reynolds numbers. It shows that both the reattachment lengths and friction factors decrease as the shape becomes less contoured — trapezoid, sine, semicircle, and arc. The influence of the shape becomes more pronounced with decreasing pitch and increasing Reynolds number.

Table 1 Reattachment Lengths and Friction Factors					
Shape	p/e	Reattachment Lengths		Friction Factor Ratios, f/f_{trap}	
		Reynolds Number, Re		Reynolds Number, Re	
		9,400	39,000	9,400	39,000
sine	20	4.75	3.88	0.993	1.019
semicircle	20	4.69	3.25	0.968	0.959
arc	20	4.41	2.81	0.959	0.936
trapezoid	20	4.94	3.72	1.000	1.000
sine	10			0.989	1.017
semicircle	10			0.962	0.964
arc	10			0.948	0.885
trapezoid	10			1.000	1.000

Local Pressure Distributions

Figure 13 shows the local pressure-drop distributions at the wall, the centerline, and the radial average for a sine-shaped rib. The reference pressure is the centerline pressure at the inlet. Note that there is a large radial pressure gradient even for a fully developed

Figure 13 Pressure-Drop Distributions for a Sine-Shaped Disruption; Re = 9,370

Figure 14 Comparison of the Pressure Distributions for the Smaller Reynolds Number

flow field. The condition of a radially uniform pressure distribution certainly does not exist as for the case of a plain or unenhanced tube. Note the very sharp drops of the wall pressure at the disruption and the recovery that follows. There is a slight increase in the centerline pressure just in front of the disruption followed by a decrease. The average pressure drop decreases at the disruption and is also followed by some pressure recovery.

The wall pressure drops are shown in Figures 14 and 15 for Reynolds numbers of 9,400 and 39,000, respectively. Note that the entire wall pressure field is affected by the disruption shape. The pressure drop at the rib increases for the more contoured shapes. There is a greater pressure recovery away from the rib when the rib is less contoured and this is the major reason why the overall pressure drop decreases with a decrease in the surface contour.

Figure 15 Comparison of the Pressure Distributions for the Larger Reynolds Number

Heat Transfer Enhancement Level

Consider next the heat-transfer performance of the sine-shaped disruption for two Reynolds numbers that is shown in Figure 16. Note that the enhancement levels increase in the vicinity of the disruption but decrease in the boundary-layer development region downstream of the reattachment point. Again, this pattern was obtained for all shapes. This tradeoff is the reason the enhancement level increases to a maximum value with increasing Reynolds numbers. The maximum level depends on the disruption height, pitch, shape, and the Prandtl number (Arman and Rabas, 1992). It is also apparent that the contribution from the downstream recirculation region increases with increasing Reynolds number and the magnitude of heat-transfer maximum in this region also increases. The heat-transfer maximum located in the recirculation region moves toward the rib with an increase in Reynolds number. These points of maxima are located well within the recirculation region and are

Figure 16 Heat Transfer Enhancement Level for a Sine Shaped Disruption

between the reattachment points and the points of the maximum shear stresses.

The effect of the disruption shape on the local Nusselt numbers is shown in Figures 17 and 18 at Reynolds numbers of about 9,400 and 39,000, respectively. Note that the variations caused by the different shapes are mostly concentrated in the region of the disruption. It is readily apparent, then, that the influence of the disruption shape on the thermal performance should decrease as the pitch increases. The heat-transfer enhancement levels in the recirculation zone, in the boundary layer development zone, and in the front recirculation zone increase when the surface contour of the rib becomes more contoured. Also, the point of the maximum heat transfer in the downstream recirculation zone moves toward the rib as the rib surface becomes more contoured.

Figures 17 and 18 show that the local heat-transfer performance in the rib region is very complex. There are three maxima for the semicircle, arc, and trapezoid shapes and two for the sine shape. For the lower Reynolds number, the magnitudes of the middle maxima are the smallest for the semicircle-, arc-, and trapezoid-shaped ribs; however, the magnitudes become the largest of the three maxima at the higher Reynolds number.

Figure 17 Comparison of the Heat Transfer Enhancement Levels for the Smaller Reynolds Number

Figure 18 Comparison of the Heat Transfer Enhancement Levels for the Larger Reynolds Number

The difference in the overall thermal performance resulting from the different shapes is displayed in Table 2. This table shows the mean enhancement level, (Nu/Nu_p) , and enhancement level ratio with respect to the trapezoid shape as the base for p/e values of 10 and 20. In general, there is a bigger shape effect—a departure from unity—for the smaller p/e value. This is expected because the effect of the shape on the local heat-transfer coefficient is concentrated in the region of the disruption and has a lesser effect in the boundary layer development region downstream of the reattachment point.

Table 2 -- Comparison of the Mean Nusselt Numbers

Shape	p/e	$Re=9,400$		$Re=39,000$	
		Nu/Nu_p	η/η_{trap}	Nu/Nu_p	η/η_{trap}
sine	10	2.1674	0.945	2.2203	0.958
sine	20	1.8419	0.974	1.8848	0.988
semicircle	10	2.1266	0.927	2.2143	0.955
semicircle	20	1.7920	0.947	1.8347	0.961
arc	10	1.9826	0.864	2.1158	0.913
arc	20	1.7118	0.905	1.7849	0.935
trapezoid	10	2.2941	1.000	2.3183	1.000
trapezoid	20	1.8906	1.000	1.9083	1.000

CONCLUSION

The purpose of the paper is to demonstrate that numerical modeling techniques are sufficiently advanced to predict the thermohydraulic performance of enhanced tubes with transverse disruptions with different shapes. A non-orthogonal, body-fitted numerical code was used to explore the importance of four different shapes -- sine, semicircle, arc, and trapezoid -- that are somewhat representative of commercially available enhanced tubes.

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