A STUDY OF HEAT TRANSFER AND PRESSURE LOSS WITH AIR FLOW THROUGH A WIRE SCREEN ROUGHENED ANNULAR CHANNEL

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CHAPTER I

INTRODUCTION

Most heat exchanger applications require that the largest possible rate of heat be transferred from the exchanger surface with minimum pressure loss. One method of increasing the rate of heat transferred from a surface is to increase the surface area. This method usually involves adding fins to the surface. Due to space requirements, fabrication techniques, or other factors, it is not always possible to add fins. For such cases, it may be necessary to increase the heat transfer rate by other means such as turbulence promotion.

As the development of nuclear reactors progressed, it was found that the use of finned surfaces on the fuel elements generally are not acceptable. Since the fuel element in a reactor forms the inner portion of an annular space, it is not always possible to add fins to the element due to space limitations. Also, due to the high temperatures encountered in the reactor, materials which would withstand these temperatures are required. These materials usually have low thermal conductivities and are not as effective in the process of transferring heat as other materials. It was found that the additional mass added by the finning materials results in neutron absorption.

For the above reasons, other methods of increasing the heat transfer rate were considered. One method was that of increasing the boundary layer turbulence of the fluid by roughening the surface. In general, high heat transfer rates accompany increases in boundary layer turbulence.

The effect of roughened surfaces on heat transfer and friction factor has not been completely analyzed for all surfaces. Recent interest by the nuclear power generation industry in this field has stimulated much research in flow of low Prandtl number fluids in roughened annular spaces.

Results of earlier work by Ratcliffe and Henning (1, 2)*, with screen wire roughness on surfaces in convection and boiling heat transfer, coupled with the practical need for more information on roughened surfaces in annular flow, evoked the present study. The purpose of this study was to examine the effect of screen wire roughness on smooth tubes for flow in an annular space.

Much work has been done in the study of roughened surfaces in annular flow. The recent book edited by Bergles and Webb (3) presents some of the more recent work done with enhanced surfaces in convection heat transfer. Among the more common surface enhancing techniques being considered are spines, ribs, threads, and sand grain roughness. The studies of White and Wilkie (4) and of Puchkov and Vinogradov (5) were of particular interest to this study. White and Wilkie examined the effects of multi-start ribbed surfaces in annular flow, while Puchkov and Vinogradov worked with triangular, spiral, and sawtooth spoilers in annular flow. Various methods of evaluating the performance of roughened surfaces are used, and some of the more noted correlations are discussed by Norris (6).

The purpose of the present study was to evaluate the heat transfer and friction factor characteristics of screen wire roughness for flow of air (at room temperature) through an annular space in the Reynolds number range of $5 \times 10^4 - 1.4 \times 10^5$. A performance criteria was also applied to the experi-

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*Numbers in parentheses designate references in the List of References, page 27.

mental results and direct comparisons were made with the work of Puchkov and Vinogradov.

Three mesh sizes of copper alloy screen were arbitrarily chosen from the wide selection available for this study. The sizes were 10 x 10, 20 x 20, and 30 x 30 meshes to the inch. The present study focused on screen wire as a method of surface augmentation because of:

1. Low initial cost of screen;
2. Ease of forming to contours;
3. Availability of sizes;
4. Availability of materials which are easily welded or soldered.

Used as a surface roughening material, screen wire would not require costly machine work on smooth surfaces and there would be retention of the mechanical strength of the original tube. For these reasons, it was felt that screen wire warranted study as a surface roughening material on tubes.

CHAPTER II

EXPERIMENTAL PROGRAM

Experimental Test Apparatus

A partially dimensioned drawing of the test apparatus used in this investigation appears in Fig. 1 on the following page. The forced air supply was obtained by using a centrifugal blower with a 2 h.p., 240 volt three phase motor which operated at a maximum speed of 3450 r.p.m. The blower was capable of delivering 570 cfm of air through 10 inches of water pressure drop. A slide damper was used on the inlet of the blower to regulate the flow rate.

An 8 inch in diameter duct was attached to the blower exit and a 5 inch nozzle was installed in the duct. The nozzle was calibrated by a pitot tube traverse of the nozzle exit plane at different flow rates. The average velocity of the nozzle was determined at each flow condition, and flow rates were then determined. These flow rates were correlated to corresponding static pressure drops across the nozzle.

A transition from the 8 inch duct to the 3 inch inside diameter of the outer annular portion was achieved by a funnel shaped section. The outer tube of the concentric annulus was 3 inches inside diameter with 1/4 inch thick walls. It was a section of ABS hard tubing with overall length of 106 1/2 inches. The inner tube was a 92 inch length of 3/4 inch I.D. (7/8 inch O.D.) type K copper tubing. The copper tube was supported in two locations with three 6-32 screws, 1 3/4 inches in length which were threaded through the outer tube walls. The distance between the annular inlet and test

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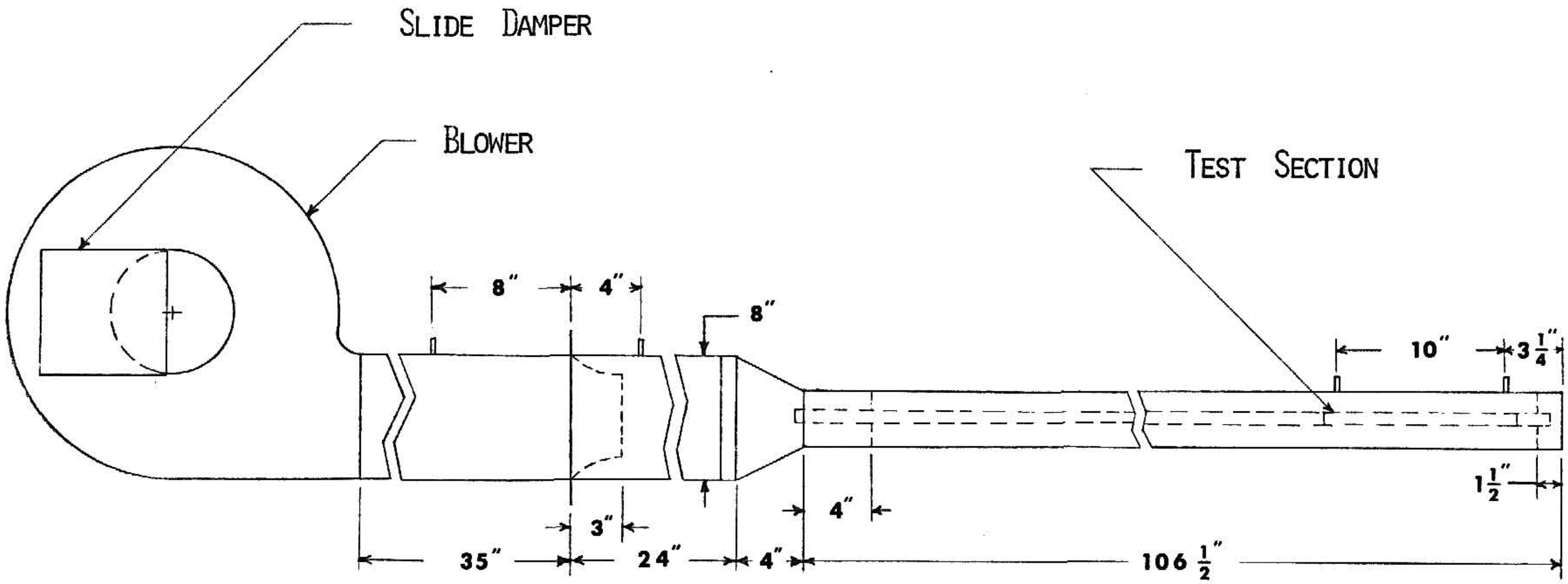

FIGURE 1. EXPERIMENTAL APPARATUS WITH TEST SECTION

section beginning was 92 inches or approximately 43 hydraulic diameters. The distance from the second support to the test section was 48 inches or approximately 22 hydraulic diameters. This corresponded to recommendations (7) for fully developed turbulent flow in concentric annuli and allowed room for damping disturbances introduced by the support. A spiral of copper wire was also attached to the first few inches of the inner tube of the annulus and a screen mesh trip was installed in the first few inches of the annular section to help promote turbulent flow.

Pressure taps were located across 10 inches of the test section and were used to determine pressure drop across the test section. Two micromanometers were used to measure the pressure drop across the test section and the nozzle. A Micarta end plug was used on the end of the test section to support it. End support was achieved by using 3 6-32 screws threaded into the outer pipe and advanced to the support plug.

The test section was heated by a commercial high watt density heater (12 inches long and 1/2 inch O.D.) which was rated at 2 kilowatts at 240 volts. The power was supplied by a variable A.C. transformer with a range of 280 volts and 28 amperes. Power input to the heater was measured with a voltmeter and ammeter with a scale accuracy of ±.5 volts and ±.05 amperes respectively. Surface temperatures of the test section were measured with four thermocouple junctions. These thermocouple wires were connected to a selector switch. A single ice bath reference junction was used and thermocouple readings were recorded on a recording potentiometer.

A 3 foot long insulated mixing chamber (not shown in Fig. 1) was used with limited success to obtain a uniform bulk exit temperature. The mixer was not used on all flow settings, since it created a rather large pressure drop and could only be used for lower capacity readings.

Experimental Test Sections

A detailed sketch of a test section is shown in Fig. 3. Three actual heated test sections were fabricated and were denoted as A, B, and C. A fourth section denoted by the letter S was also made. This section did not contain provisions for heating but was used only to obtain the isothermal friction data. The sections were made from the same 3/4 inch type K copper tubing as the inside tube of the annulus. They were cut to a length of 11 1/2 inches and thermocouples were located as shown below in Fig. 2.

Thermocouple Junctions

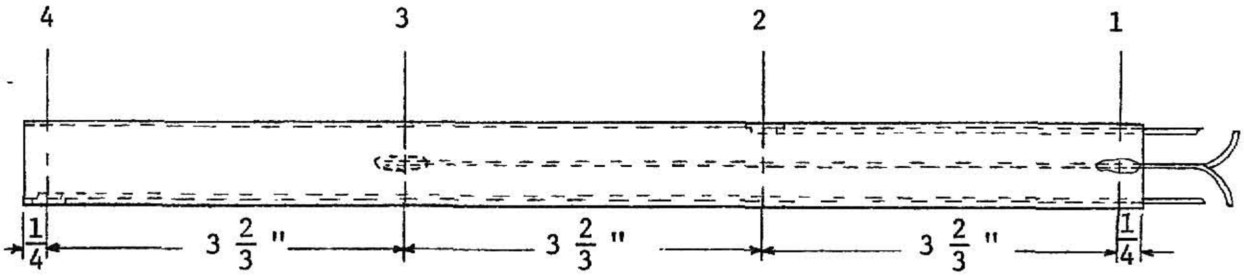

Figure 2. Test Section with Thermocouple Junction Locations

The four junctions were equally spaced along the cylinder starting 1/4 inch from each end and spiraled around the tubes in 90 degree increments so that each respective "side" of the tube was represented. Since the flow was axial through an annular space, it was felt that this would be a sufficient number to determine an accurate value for the surface temperature. Chromel-Alumel thermocouple wire with Fiberglass insulation was used to obtain temperature readings. The wire was type K with limits of error of ±4°F over the temperature range in which readings were taken. The thermocouple wires were

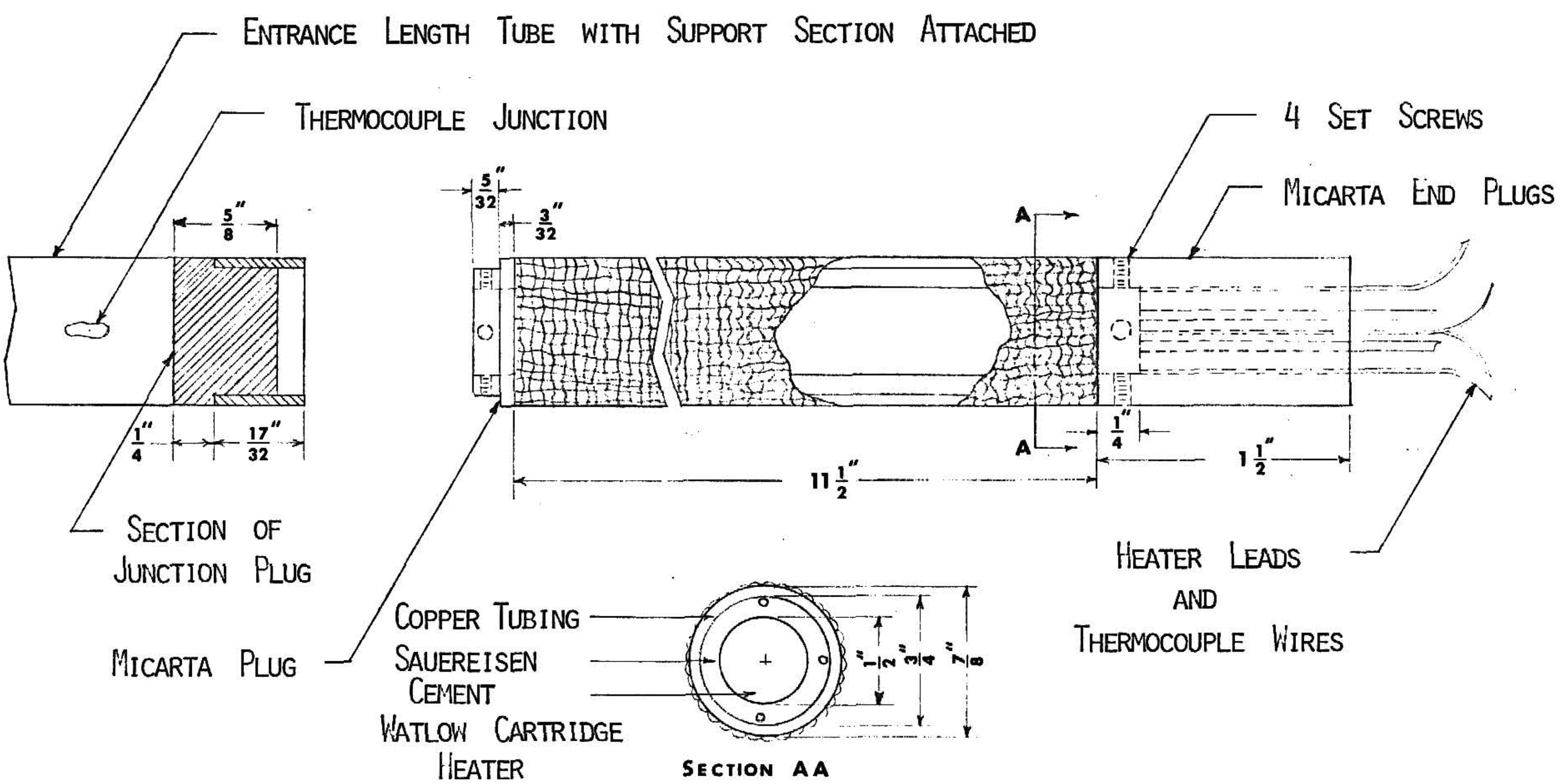

FIGURE 3. TEST SECTION ASSEMBLY

inserted into the test section from one end and pushed up through 2 small drilled holes and bent over to meet each other on the surface in a preground trough. The trough and wires were then covered with high temperature soft solder. This high temperature soldering was the main reason for use of fiberglass insulated thermocouple wires. After the junction was secured, the excess solder was carefully filed off until the surface was again smooth and then the sections were placed in a lathe and polished with emery cloth. Next, as seen in Fig. 3, a 1/8 inch thick cylinder of commercial high thermal conduction cement was poured into the tube around a greased heater blank to provide an inner base for protecting the thermocouple wires and a high contact surface for the 1/2 inch O.D. heater. After drying, the blank heater was removed and the actual heater was installed. Section A-A in Fig. 3 shows the thermocouple wires embedded in the cement ring.

Smooth or bare tubes A, B, and C were fabricated in this manner without the screen covering. These tubes were then tested and removed for the screening process.

The screening process entailed coating the tubes with a thick layer of active solder paint. The tubes were then carefully fitted with the corresponding screen mesh sizes and the screen was tightly tied to the tubes. The heater was inserted into the test section and the surface temperature was allowed to rise well above 450 degrees Fahrenheit in still air. The solder melted and bonded the wire securely to the surface of the tube.

To attain full length heating of the test section, the heater was allowed to protrude 1/4 inch from one end of the test section as seen in Fig. 3. The Micarta end plugs were then bored to accept the heater and 4 set screws were threaded through to the heater ends to hold it in place. The Micarta plug, on the right in Fig. 3, was used to support the end of the test section as mentioned earlier and visualized in Fig. 1. The Micarta plug on the left end of the test section in Fig. 3 was machined to mate into the copper ring on the end of the inner tube of the annular section. This copper ring was also attached to a Micarta plug which extended 2 inches into the copper tubing portion of the annular space. The combination of copper ring and Micarta end plugs provided support and thermal insulation for the test section end. The thermocouple junction located on the annular section about 1/2 inch from the Micarta plug was used to measure end loss.

Experimental Test Procedure

There were 6 actual runs on each tube in both smooth and screened conditions. Flow rate was controlled by the blower damper and was determined from the measured pressure drop across the nozzle. These 6 conditions of flow were as closely repeated as possible on each tube to keep measurement points compatible. After setting the flow rate, the variable transformer was adjusted to supply sufficient power to bring thermocouple number 2 close to 210°F.

When the recording potentiometer indicated that a condition of steady state surface temperature had been attained, junctions 1 through 5 were recorded, current and voltage were recorded, and test section pressure drop was read from the micromanometer. The nozzle exit temperature was also recorded with a thermometer which was extended into the exit of the nozzle through the side of the duct.

When first heating the test sections without any flow, a scan of the temperatures was taken and all junctions recorded a similar rise in temperature, indicating that there were no junction hot spots. Also, before heating, all junctions were scanned and were usually found to be within ±.5% of room temperature.

The flow conditions on the fourth run allowed the exhaust mixer to be attached since a decreased amount of air was being allowed to flow through the section. A calibrated thermometer was placed in the exhaust of the mixer and the temperature was recorded. This was repeated for the fifth and sixth runs.

CHAPTER III

EXPERIMENTAL RESULTS AND COMPARISONS

Consideration of Parameters

Results of the experiment are shown in Fig. 4 and Fig. 5. Figure 4 is a log log plot of Stanton number (St) versus Reynolds number (Re) and Fig. 5 gives a log log graph of friction factor (f) versus Reynolds number. The value used for the Stanton number was determined from

$$St = \frac{h}{\rho VC} = \frac{q(A_{cs})}{A_s \rho Q C(T_s - T_B)} \tag{1}$$

where

q = power input to heater = (I)(E)(3.413) ;

A_{cs} = cross sectional area of the annular space

$= \frac{\pi}{4} (d_2^{\,2} - d_1^{\,2})$;

A_s = surface area of a smooth test section.

The expressions ρ and C represent density and specific heat of the air at the bulk temperature. Q represents the flow rate in cfh. The value T_s was the surface temperature of the test section which was arrived at using the following averaging technique:

$$T_s = \frac{T_1 + 2T_2 + 2T_3 + T_4}{6} \tag{2}$$

Reference to Fig. 2 shows that junctions 2 and 3 represented twice as much surface area as 1 and 4. It was felt that a heavier weighting should be given to these temperatures and that this weighting should be a function of

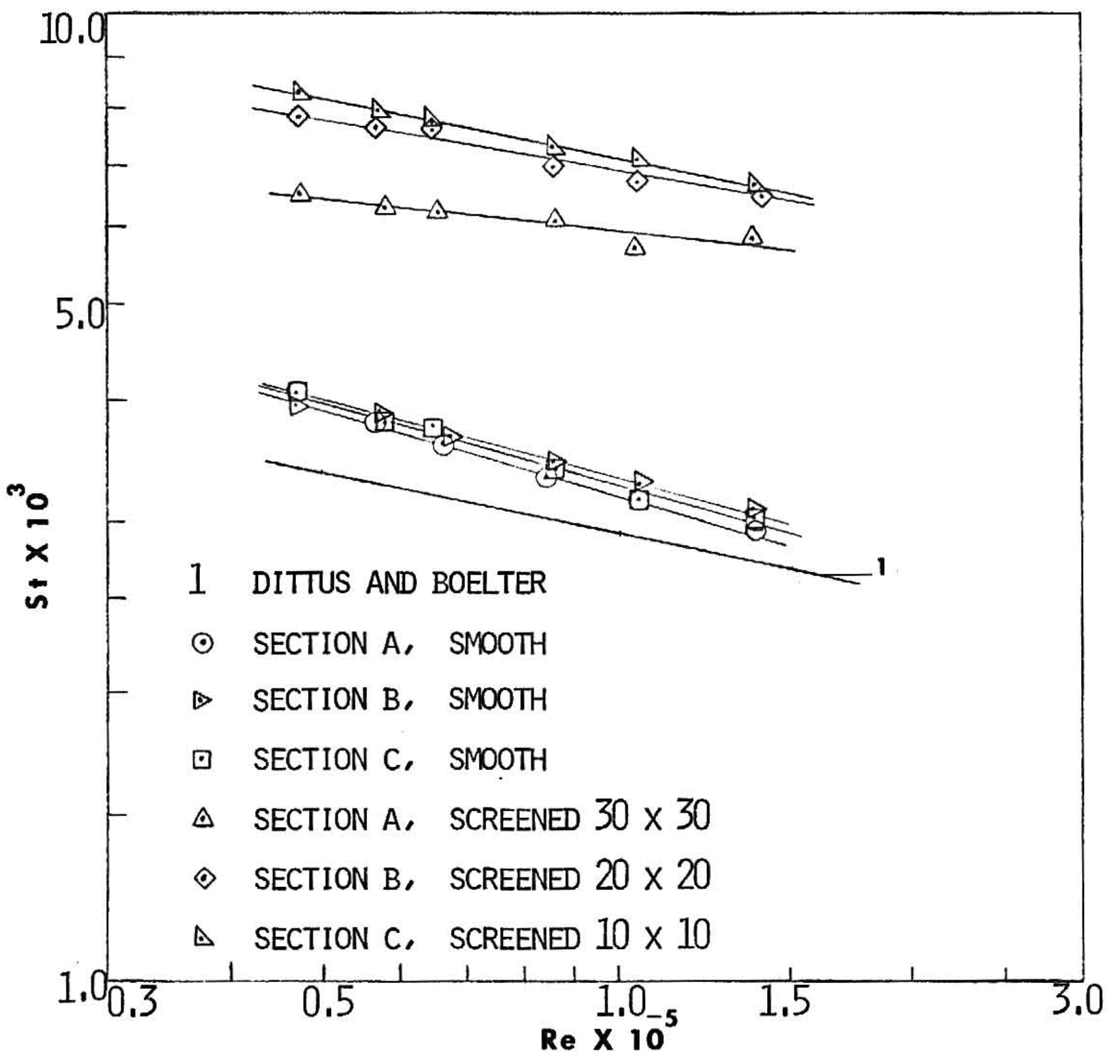

FIGURE 4. COMPARISON OF STANTON NUMBER RESULTS FOR SMOOTH AND SCREENED TEST SECTIONS

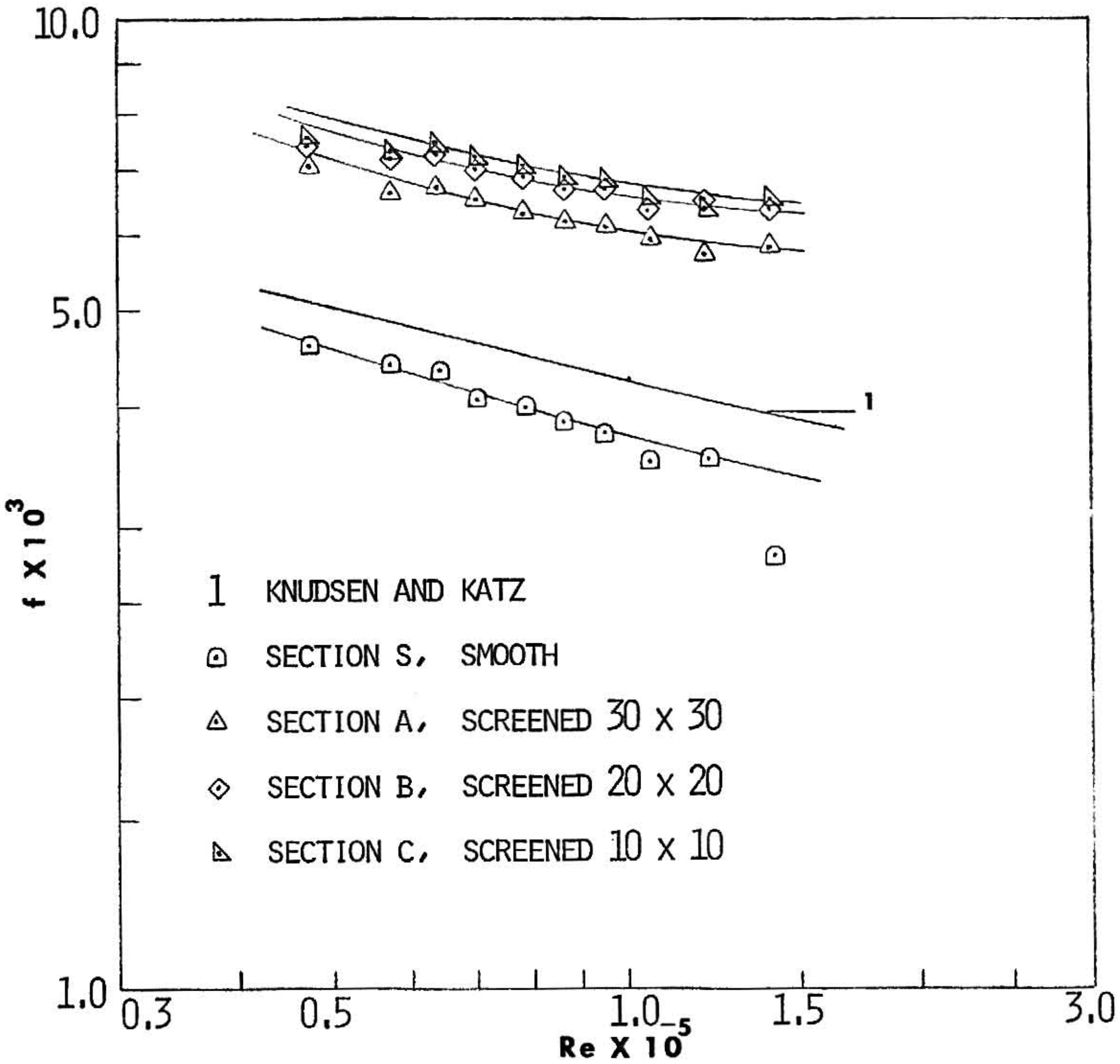

FIGURE 5. COMPARISON OF FRICTION FACTOR RESULTS FOR SMOOTH AND SCREENED TEST SECTIONS

the surface area represented. The value for the bulk temperature T_B was obtained from

$$T_B = T_{Bi} + \frac{\Delta T_B}{2} \tag{3}$$

where:

T_{Bi} = Bulk temperature at the nozzle exit;

$\Delta T_B = \frac{q}{\rho QC}$.

The friction factor was determined from the Fanning equation (2)

$$f = \frac{\Delta PD}{2\rho V^2 L} \tag{4}$$

where, V = velocity of air in the annular space;

ΔP = pressure drop across the test section in inches of H_2O;

L = distance between pressure taps over the test section;

D = Hydraulic diameter, $d_2 - d_1$.

Values for the Reynolds number were determined through the relations

$$Re = \frac{VD\rho}{\mu} \tag{5}$$

where μ represents the dynamic viscosity of the air at bulk temperature conditions.

Presentation of Results

Reference to the curves in Fig. 4 show experimental curves for sections A, B, and C both smooth and screened. The Dittus-Boelter equation (7)

$$St = .023\ Re^{-.2}\ Pr^{-2/3} \tag{6}$$

is shown to allow comparison with theoretical smooth tube values. All experimental smooth tube curves lie close together and slightly higher than the theoretical values. At Re of 1×10^5, the increase in St number is about 85% for tube A from smooth conditions to roughened conditions, 106% for Section B, and 115% for tube C. These figures have little value without comparison to friction increases.

From Fig. 5, comparison between measured isothermal smooth tube S and screened tube A results shows an increase in f of 62%, between S and B 77%, and between S and C 82%. In each instance, the percentage increase in St was greater than the percentage increase in f. On Fig. 5, a plot is also made of the equation

$$f = .076\ (Re)^{-0.25} \qquad (7)$$

which was suggested by Knudsen and Katz (7) for plain annuli. They state that actual data may vary as much as 35% from equation (7). Experimental smooth values fall within 15% of this curve at Re of 10^5. Table 1 contains the actual experimental results for the plotted quantities in figures 4 and 5.

Comparison with Published Data

A log log plot for values of Nusselt number versus Re is shown in Fig. 6, where,

$$Nu = Re\ Pr\ St. \qquad (8)$$

Figure 6 shows screened and smooth tube experimental values for those sections tested, and values found by Puchkov and Vinogradov (5) in their work with triangular surface spoilers (s/d = .0373) and spiral spoilers (s/d = .972). (Their values were taken directly from the curves in their paper.) Puchkov and Vinogradov also present a curve for smooth tubes which corresponds to eq. 6 (Dittus - Boelter).

It should be noted that all present experimental values of Nu fall above those of Puchkov and Vinogradov.

TABLE 1

Experimental Results - Stanton Number, Nusselt Number, and Friction Factor Data for Smooth and Screened Test Sections

Reynolds Number ($X10^{-5}$)	Stanton Number ($X10^{3}$)	Nusselt Number	Reynolds Number ($X10^{-5}$)	Isothermal Friction Factor ($X10^{3}$)
	Test Section A		Smooth	
1.377	2.94	286.6	(See test section S data below for isothermal friction factor for smooth tubes).	
1.048	3.15	233.73		
.849	3.30	198.36		
.662	3.59	168.26		
.573	3.79	153.75		
.474	4.02	134.91		
	Test Section B		Smooth	
1.384	3.06	299.84		
1.052	3.25	242.06		
.860	3.41	207.63		
.677	3.62	173.51		
.574	3.86	156.87		
.474	3.92	131.55		
	Test Section C		Smooth	
1.381	3.02	295.28		
1.052	3.14	233.87		
.860	3.37	205.19		
.650	3.73	171.65		
.581	3.76	154.67		
.474	4.03	135.24		
	Test Section S		Smooth	
			1.407	2.82
			1.208	3.53
			1.052	3.51
			.952	3.77
			.860	3.85
			.784	4.00
			.700	4.07
			.639	4.32
			.574	4.41
			.474	4.60

TABLE 1 -- Continued

Reynolds Number ($X10^{-5}$)	Stanton Number ($X10^{3}$)	Nusselt Number	Reynolds Number ($X10^{-5}$)	Isothermal Friction Factor ($X10^{3}$)
	Test Section A		Screened	30 x 30
1.373	5.80	563.81	1.396	5.84
1.044	5.67	419.10	1.208	5.73
.864	6.05	370.08	1.052	5.90
.662	6.20	290.59	.957	6.10
.581	6.22	255.86	.860	6.19
.478	6.46	218.62	.784	6.32
			.700	6.50
			.639	6.73
			.574	6.60
			.474	7.03
	Test Section B		Screened	20 x 20
1.404	6.43	639.16	1.396	6.42
1.052	6.62	493.07	1.208	6.50
.860	6.87	418.03	1.052	6.37
.650	7.50	345.15	.952	6.66
.573	7.52	305.07	.860	6.68
.478	7.72	261.26	.784	8.87
			.700	7.19
			.639	7.24
			.574	7.16
			.474	7.37
	Test Section C		Screened	10 x 10
1.377	6.59	642.47	1.396	6.50
1.052	6.98	519.88	1.208	6.37
.860	7.20	438.39	1.052	6.56
.650	7.66	352.51	.952	6.80
.573	7.84	318.06	.860	6.86
.478	8.23	278.52	.784	7.04
			.700	6.98
			.639	7.43
			.574	7.21
			.474	7.49

A log log plot of f vs. Re appears on Fig. 7. This experimental data has also been compared to that of Puchkov and Vinogradov. Their method of obtaining smooth friction data is not known, but eq. (7) by Knudsen and Katz falls directly on their values. It should also be noted that their curves lie above all experimental results of the present study.

To compare the present experimental results to those of Puchkov and Vinogradov, an efficiency parameter (η) was defined as:

$$\eta = \frac{Nu/Nus}{f/fs} , \tag{9}$$

where the subscript s denotes smooth test sections. This parameter was then plotted vs. Re in Fig. 8. The smooth tube values, Nus and fs, for the present study were the experimental values determined in this study. The smooth tube values for the work by Puchkov and Vinogradov were the theoretical values determined from equations 6 and 7. In calculating η, all values were taken directly from figures 6 and 7. Care was taken in choosing those values for s/d from Puchkov and Vinogradov which would result in maximum η values for their surfaces. (Points were calculated at Reynolds numbers of .5, 1, and 1.5×10^5.)

The present results would compare even more favorably if the values for efficiency were calculated using Puchkov and Vinogradov's smooth section curves. Since these correlate directly with theoretical equations this technique of comparison would not be at all unsuitable.

Consideration of Error

Error due to heat loss by radiation to the surroundings was calculated using the equation

$$q_r = \frac{\sigma A_1 (T_s^4 - T_2^4)}{\frac{1}{E_1} + (A_1/A_2)(\frac{1}{E_2} - 1)} , \tag{10}$$

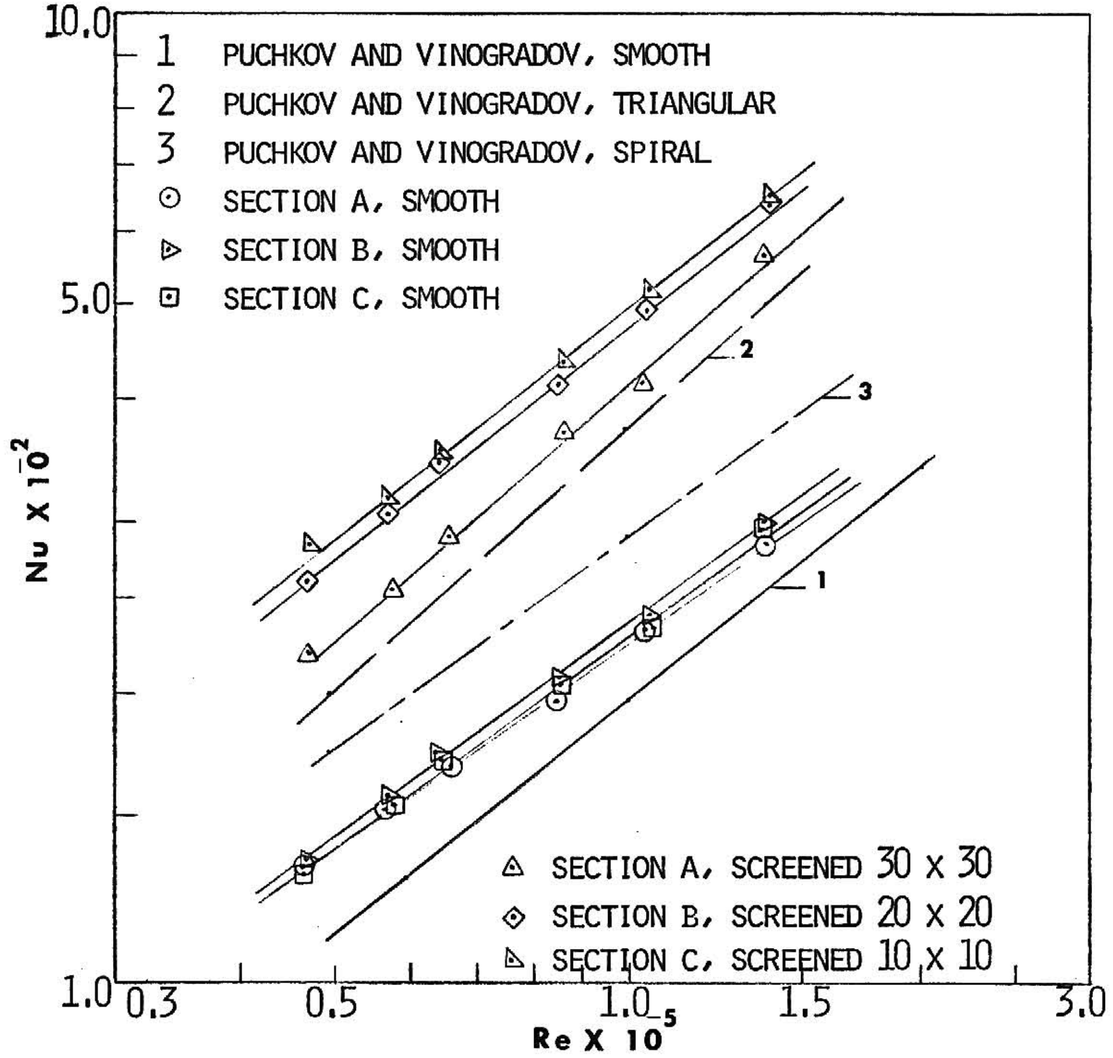

FIGURE 6. COMPARISON OF NUSSELT NUMBER RESULTS FOR SMOOTH AND SCREENED TEST SECTIONS

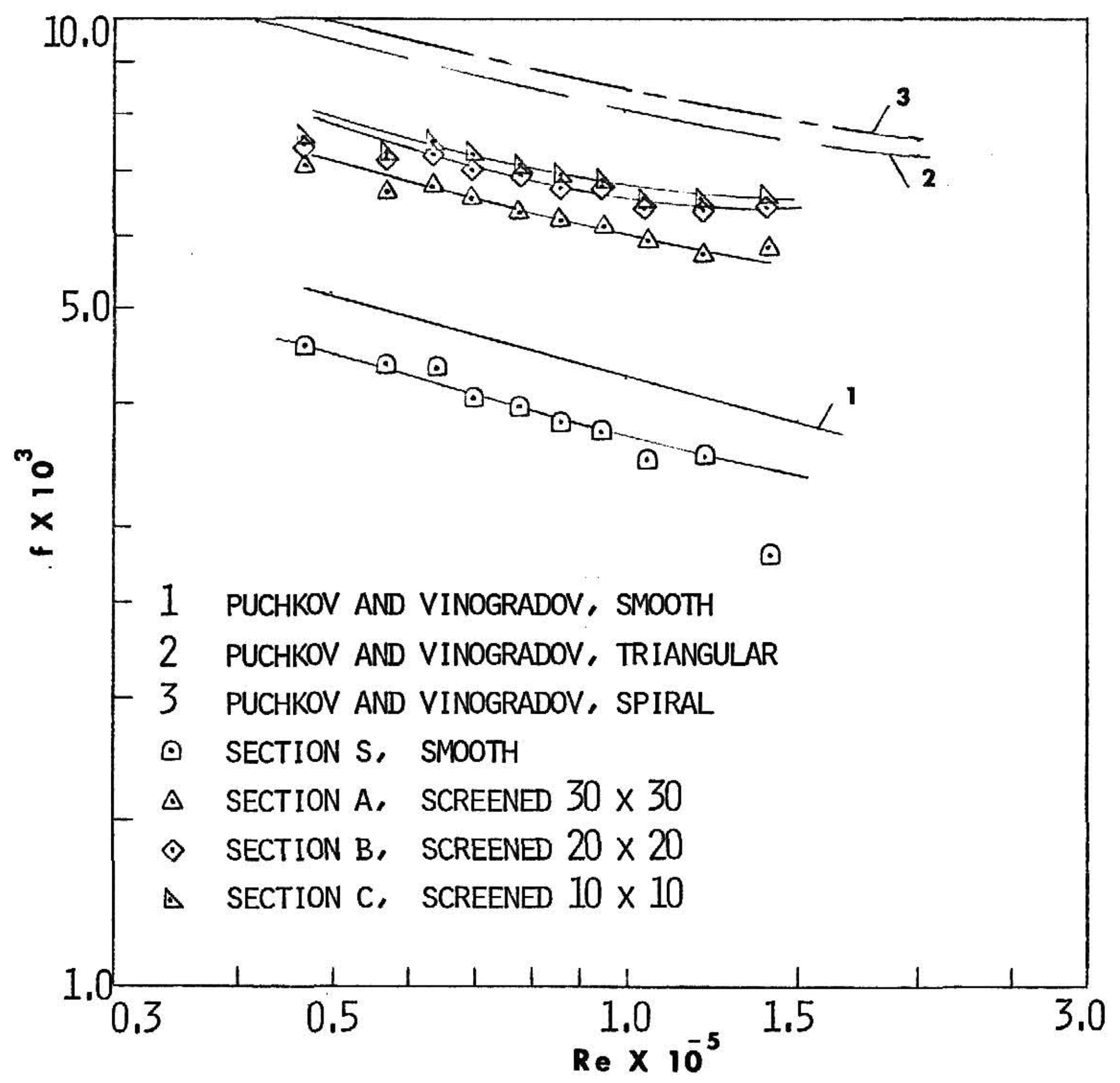

FIGURE 7. COMPARISON OF FRICTION FACTOR RESULTS FOR SMOOTH AND SCREENED TEST SECTIONS

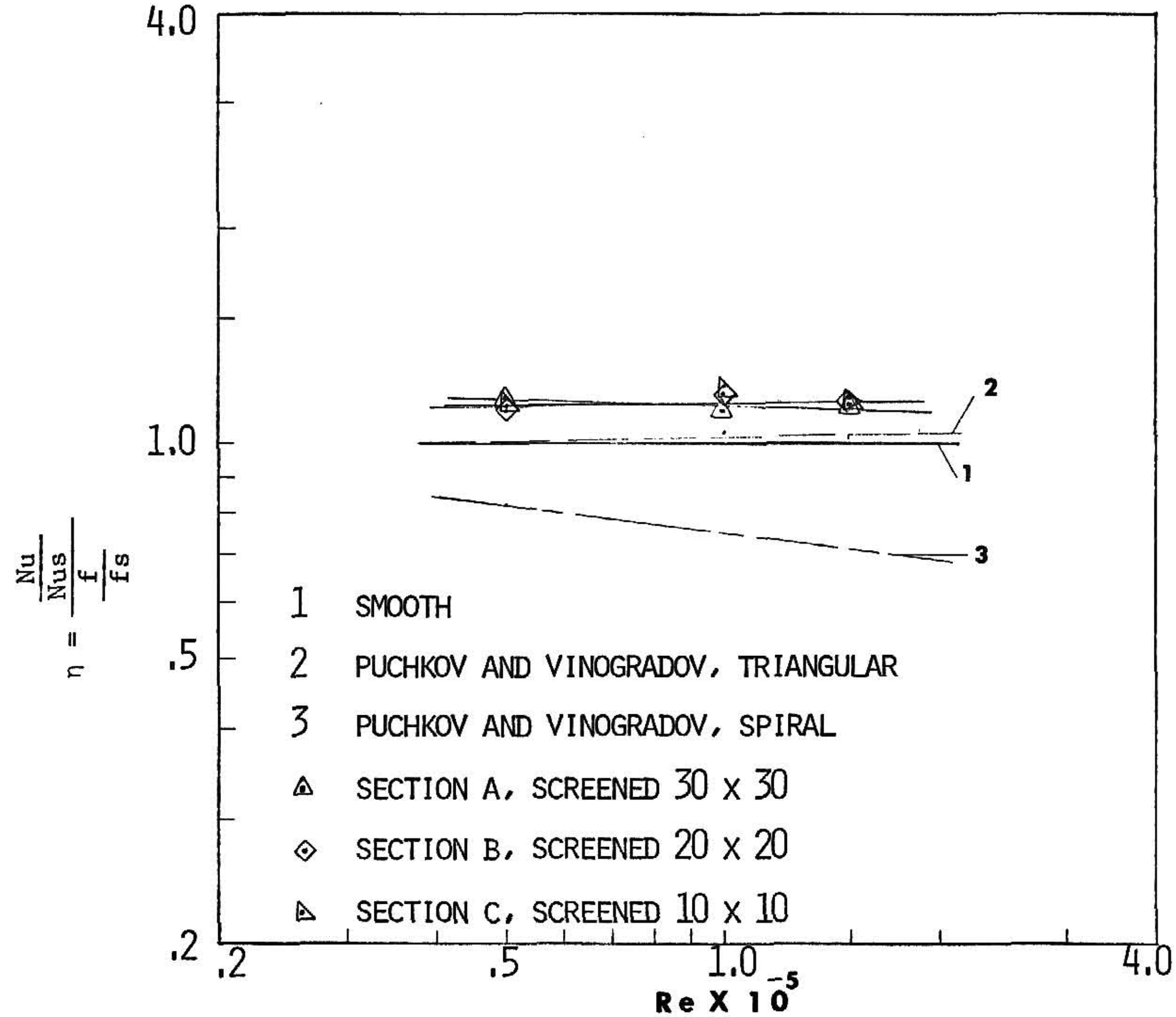

FIGURE 8. EFFICIENCY RESULTS FOR SCREENED TEST SECTIONS WITH SMOOTH TEST SECTION REFERENCE

where $\sigma = .1714 \times 10^{-8} \frac{\text{Btu}}{\text{hr.sq.ft.°R}}$, E_1 and E_2 are the emissivity for the copper and black ABS tubing. Those values were taken to be .4 and .93 respectively. T_2 was set equal to T_B, and A_1 and A_2 are the surface areas of the inner and outer annular surfaces.

Loss due to radiation was calculated to be 16.8 $\frac{\text{Btu}}{\text{hr}}$ or 5.2% of the smallest power input. For screened tubes, the emissivity was taken as .85 and maximum heat loss due to radiation was calculated to be 35.7 Btu/hr or 6.9% of the smallest power input to screened cylinders. All sections were run at a fairly constant surface temperature which resulted in the radiation term remaining constant at the respective smooth or screened tube value. Values for T_5, tabulated in Appendix A, indicate that there was no end loss. As planned, the support section acted as a good thermal insulator.

Errors in values for St, Nu, Re and f were estimated to be 8.4, 10.7, 6.7 and 12.8 percent respectively. These values were obtained through consideration of extreme values in each quantity and are considered to be maximum in value. Since comparisons were made with the present study's own smooth tube data, the inaccuracy due to error is felt to be minimal.

Error due to improper screen wire attachment was very hard to predict. Great care was exercised in attachment and few if any mesh holes were filled with solder. A few locations on both the 20 x 20 and 30 x 30 mesh tubes were found not to have good contact. These areas were not large, however, and it was not felt that they affected the results sufficiently to warrant reconstruction. If corrections had been made, the measured values for St and Nu should have increased.

The technique for arriving at the surface temperature (eq. 2) was felt to be accurate. From Appendix A, reduced data for thermocouple junctions 1-4 shows that junctions 1 and 2 had higher values than did 3 and 4. The reason for this was believed to be a combination of junctions 1 and 4 being located on the ends, and a possible variation of the heat transfer coefficient along the test section. In actual calculations, a change in T_s of 5°F results in only a 3.6% decrease in the value of St for conditions of low heat input and flow rates on smooth tubes. A change in T_s of 5°F results in a 4.1% decrease in the value of St for large Re on 10 x 10 mesh screened section C. It should also be noted that a very large temperature change of any one junction would be required to cause a change in T_s of this magnitude. Since constant surface temperatures were held on all runs, the error due to a shift, as considered above, would be constant for all tubes.

An attempt was made to obtain a heat balance on the system, but it was found that the bulk temperature rise was too small to measure accurately in many cases. For example, with a bulk temperature rise of 2°F, measurements with an inaccuracy of 1°F would result in errors of 50%.

CHAPTER IV

CONCLUSION AND RECOMMENDATIONS

The effect of screen wire roughness on enhancing heat transfer for tubes in annular flow was studied. Screened tubes exhibited an increase in St of as much as 115% over values for smooth tubes. Friction factor data exhibited a smaller percentage increase in value. An efficiency parameter (η) was defined and a comparison was made, in Fig. 8, of the results of this study with those of Puchkov and Vinogradov. All screen mesh roughened surfaces rated higher than those tested by Puchkov and Vinogradov using this performance criteria.

Maximum values of St were exhibited by roughened tube C with 10 x 10 mesh screen. This tube also exhibited the largest increase in f. Upon applying the efficiency criteria, there was little difference in the three screened tubes. An equal heat transfer performance recommendation would have to be given for all three screen meshes based upon these comparison techniques.

Recommendations for Further Study

It would be of interest to apply other performance criteria to the present study. Comparison methods are discussed by Norris (6). White and Wilkie (4) made detailed studies of multi-start ribbed surfaces and it would be of interest to draw comparison between their work and the present study. Applications of other criteria were beyond the scope of this study. The "Hall

Transformation" (8) is recommended for future studies as a method of transforming rough tube values to those of equivalent smooth tubes. Using this technique, comparison of the present study's results could be made with those of Wilkie.

In future studies, sufficient heated test section length should be provided so that thermocouple junctions lie only in the region of the fully developed thermal boundary layer. Values of Nu should be lower in this region and have a constant value along the section for each flow and heating condition.

Recommendations, for increased heated test section length, should include provisions for increasing the bulk temperature rise to values which could be accurately measured. A heat balance could then be made to provide another check on the accuracy of the experiment.

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APPENDIX

APPENDIX A

REDUCED DATA FOR SMOOTH AND SCREENED TEST SECTIONS

Voltage (volts)	Current (Amperes)	Average Surface Temperature (in °F)	T_1 (°F)	T_2 (°F)	T_3 (°F)	T_4 (°F)	T_5 (°F)	Flow Rate (CFM)	Test Section Pressure Drop (inches of water)
	Test Section A	Smooth	Thermocouples Calibrated ±.5% Room Temperature						
74.8	2.68	205.5	211	218	205	176	84	360	.386
67.5	2.45	207.0	214	219	205	180	84	274	.220
61.5	2.25	202.8	210	214	201	177	82	222	.151
56.0	2.03	201.0	208	211	199	178	84	173	.1
55.0	1.98	207.0	214	217	205	184	84	150	.078
53.2	1.94	217.2	225	227	215	194	84	124	.058
	Test Section B	Smooth	Thermocouples Calibrated ±.5% Room Temperature						
74.2	2.67	198.5	208	212	196	167	84	362	.373
67.1	2.43	201.34	212	214	198	172	85	275	.182
61.5	2.23	199.5	210	211	196	173	85	225	.128
56.5	2.05	200.34	210	211	197	176	86	177	.085
54.9	1.97	205.84	216	217	202	181	87	150	.063
51.0	1.85	211.84	222	222	208	189	89	124	.045
	Test Section C	Smooth	Thermocouples Calibrated ±1.0% Room Temperature						
74.5	2.68	198.17	204	210	198	169	81	361	.337
67.2	2.44	202.67	210	215	201	174	81	275	.190
61.6	2.24	198.34	205	209	197	173	83	225	.136
56.1	2.04	197.5	205	207	196	174	83	170	.088
55.0	2.00	207.0	215	217	205	183	84	152	.062
52.0	1.89	210.5	219	220	208	188	85	124	.051

APPENDIX A -- Continued

Voltage (volts)	Current (Amperes)	Average Surface Temperature (in °F)	T_1 (°F)	T_2 (°F)	T_3 (°F)	T_4 (°F)	T_5 (°F)	Flow Rate (CFM)	Test Section Pressure Drop (inches of water)
	Test Section A	Screened 30 x 30		Thermocouples Calibrated ±.5% Room Temperature					
104.8	3.68	201.33	212	219	199	160	81	359	.546
91.8	3.24	199.5	210	215	197	163	82	273	.301
81.8	2.89	194.17	204	208	192	161	83	226	.213
73.2	2.6	196.84	206	209	195	167	83	173	.127
70.1	2.49	203.0	213	216	200	173	84	152	.097
65.2	2.33	207.0	217	219	204	179	85	125	.069
	Test Section B	Screened 20 x 20		Thermocouples Calibrated ±.5% Room Temperature					
108.9	3.81	197.5	206	216	192	163	84	367	.518
100.0	3.5	208.17	220	226	202	173	83	275	.294
87.5	3.07	196.67	207	211	192	167	84	225	.201
78.0	2.74	193.0	204	204	189	168	82	170	.130
74.7	2.63	198.17	209	208	195	174	83	150	.097
69.2	2.44	199.83	210	210	196	177	84	125	.071
	Test Section C	Screened 10 x 10		Thermocouples Calibrated ±.5% Room Temperature					
114.2	4.02	208.84	217	223	209	172	83	360	.541
100.9	3.55	204.5	214	219	204	171	83	275	.315
91.1	3.21	201.67	211	214	200	171	84	225	.219
83.8	2.95	207.34	217	220	205	177	83	170	.140
78.8	2.78	206.67	216	219	204	178	85	150	.110
71.2	2.54	201.0	210	212	198	176	85	125	.079

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VITA

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Candidate for the Degree of

Master of Science

Thesis: A STUDY OF HEAT TRANSFER AND PRESSURE LOSS WITH AIR FLOW THROUGH A SCREEN WIRE ROUGHENED ANNULAR CHANNEL

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A STUDY OF HEAT TRANSFER AND PRESSURE LOSS
WITH AIR FLOW THROUGH A WIRE SCREEN
ROUGHENED ANNULAR CHANNEL

by

RODNEY L. WEBRING

B. S., Kansas State University, 1971

AN ABSTRACT OF A THESIS

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

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Name: Rodney L. Webring Date of Degree: 1972

Institution: Kansas State University Location: Manhattan, Kansas

Title of Study: A STUDY OF HEAT TRANSFER AND PRESSURE LOSS WITH AIR FLOW THROUGH A WIRE SCREEN ROUGHENED ANNULAR CHANNEL

Pages in Study: 31 Candidate for Degree of Master of Science

Major Field: Mechanical Engineering

Scope and Method of Study: Recent applications of heat exchangers has required maximum possible performance from the system. Much of the research effort in heat transfer has centered on methods of enhancing the surface of the exchanger to increase the turbulence in the hydrodynamic boundary layer and increase the rate of heat transfer from the surface. Many surface alteration techniques have been tested and many are left to be tested.

The use of enhanced surfaces in annular flow in nuclear reactors stimulated interest in the present study. The object of this thesis was to determine the heat transfer and friction loss characteristics for air flowing through an annular space over a screen wire roughened heating element.

Three smooth test sections were constructed and tested in an annular flow apparatus over a Reynolds number range of $5 \times 10^4 - 1.4 \times 10^5$. The tubes were then covered with 10 x 10, 20 x 20, and 30 x 30 meshes to the inch copper alloy screen. The sections were again tested and heat transfer and pressure loss data were recorded.

Findings and Conclusions: Comparisons were made between the screened and smooth tubes and the screened sections exhibited a maximum increase in Stanton number of 115% over smooth tube values with a corresponding maximum increase in pressure drop of only 82%. An efficiency parameter was defined and results were compared with published work on other surface enhancing

techniques. It was found that the wire screen surfaces exhibited higher efficiency than the other surfaces in the comparison.