EXPERIMENTAL STUDY OF HEAT TRANSFER AUGMENTATION BY SCREEN WIRE ROUGHNESS ON CYLINDERS IN CROSS FLOW

by

JOHN THOMAS RATCLIFFE

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Approved by:

Robert L. Gorton

Major Professor

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CHAPTER I

INTRODUCTION

In heat exchanger design there is an ever-present stimulus to seek economy in operating cost and in size. Because of the opportunity to realize such economy, there is a continuing search for ways to increase heat transfer rates from tubes. Two methods of increasing the heat transfer rates are available to manufacturers: increase the effective surface area of the tubes, or alter the surface characteristics to increase the surface heat transfer conductance.

The most widely accepted method has been to increase surface area. This is particularly so for the situation of heat transfer from the external surface of tubes where area increases are easily accomplished by annular finning. However, in some instances finning is impractical. For example, nuclear power plants are being designed to operate at higher temperatures, requiring the use of high alloy steels in their tubing. These metals are difficult to fabricate and, due to poor thermal conductivity, are relatively inefficient as fins. Another example is in compact heat exchangers, where size consideration dictates closely spaced tubes and eliminates the use of long fins. Shorter fins may be used, but, in the limit as fins become shorter, this is equivalent not to increasing surface area, but to altering the tube surface characteristics.

Neal and Hitchcock (1)* performed an experiment on low profile longitudinal fins on tubes in crossflow. The results of their study showed increases

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*Numbers in parentheses designate references at the end of the thesis. (page 25) in heat transfer by as much as 40%. This increase was attributed to not only a larger surface area but primarily to an increase in turbulence. The low fins acted as trips, which provided a turbulent boundary layer and increased heat transfer rates. These low profile fins, because of their effect on the boundary layer, can be considered a form of surface roughness.

Rather surprisingly, in the search of the literature, no other studies of the effect of surface roughness on heat transfer from tubes in crossflow were located. This search included the current heat transfer publications and the recent (1970) compilation of references in "Augmentation of Convective Heat and Mass Transfer" by Bergles and Webb (2). This lack of information is perhaps understandable when it is remembered that the usual applications allow use of finned surfaces.

Because of the increasing number of applications which require improved heat transfer rates from unfinned tubes in crossflow, it was decided to select a method of surface alteration and to determine its potential for heat transfer enchancement. Wire screen was selected as a roughness element because it is readily available in a variety of mesh sizes and wire diameters and because it has a potential for application in practical heat exchangers.

The purpose of this study was to determine the heat transfer and drag characteristics of tubes in crossflow, tube surfaces having been altered by application of three different mesh wire screens.

CHAPTER II

EXPERIMENTAL PROGRAM

The following chapter describes the experimental equipment used, the test section, and the procedure followed during the experiment.

Test Apparatus

Figure 1 shows a schematic diagram of the entire test assembly used in this experiment. It consists of five main groups; air movement and velocity measurement group, power input and measurement group, temperature measurement group, test section support group and drag measurement group.

Air Movement and Velocity Measurement

A ten horsepower direct current motor was used to drive the centrifugal blower. The motor speed was used for coarse adjustment of air flow. Fine adjustment of air flow was made by a slide damper on the inlet of the blower. The blower was connected to a 4 x 4 x 8 foot plenum chamber by a 12 inch round duct. There were baffles and screens inside the plenum chamber to dampen fluctuations from the blower. Air from the plenum chamber exited by a 14 inch round duct. A standard concentric round to flat oval transition was used to connect the 14 inch round duct to a 6 x 12 inch flat oval duct. The flat oval duct was 12 feet long. A wire screen was placed 5.5 duct diameters downstream of the transition section to produce a symmetric and flat velocity profile in the duct.

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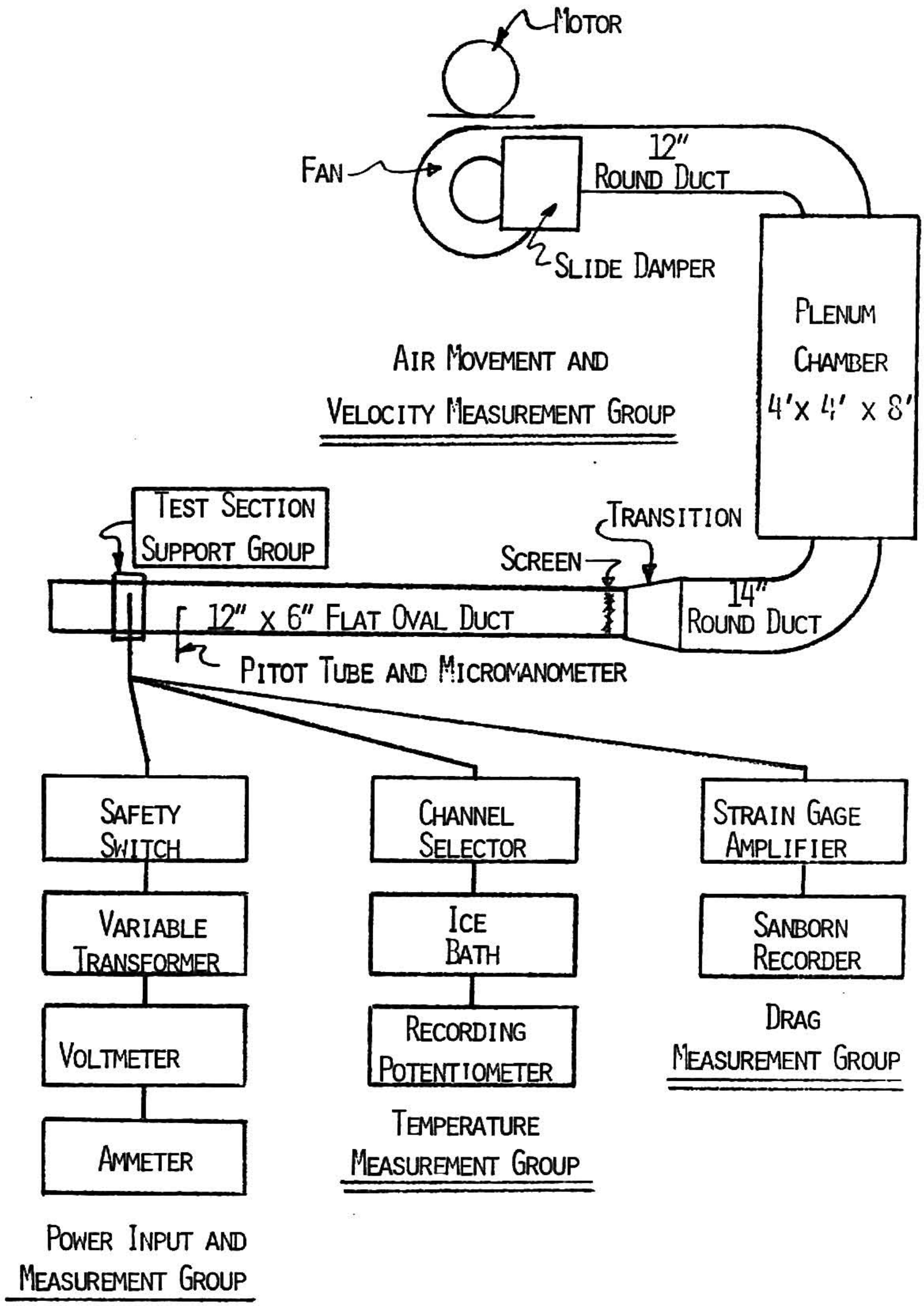


Figure 1. Schematic of Experimental Assembly

The velocity of the air was measured by a pitot-static tube in conjunction with a Meriam micromanometer. Velocity measurements were taken at the geometric center of the duct 7.64 duct diameters from the exit (8 inches upstream of the test section).

Power Input and Measurement

A 0-230 volt 10 amperes variable transformer supplied the power for the test section heater, a 500-watt Watlow cartridge heater. A volt meter and ammeter were used to determine the voltage drop across the heater and the current supplied to it. Power input was determined from voltage and current measurements. A safety switch was installed at the mouth of the duct. If the air movement stopped, power to the test section heater was automatically shut off.

Temperature Measurement

Test cylinder temperature was measured by use of six copper-constantan thermocouples, five on the test cylinder and one on an end plug. The thermocouples wires were brought out to a channel selector. A single ice water junction was used. The leads from the cold junction were connected to a recording potentionmeter.

Test Section Support

The support system was designed to allow rapid change of test cylinders. Figure 2 shows an isometric of the test support system. A 2 x 10 inch hole was cut into the top of the duct 4.92 duct diameters from the exit. The 12-1/8 inch by 6-1/3 inch support bracket was made of welded 1-1/4 inch steel angle and bolted to the top of the duct by four 1/4 inch bolts. Four studs were welded to the bracket for securing the support plate.

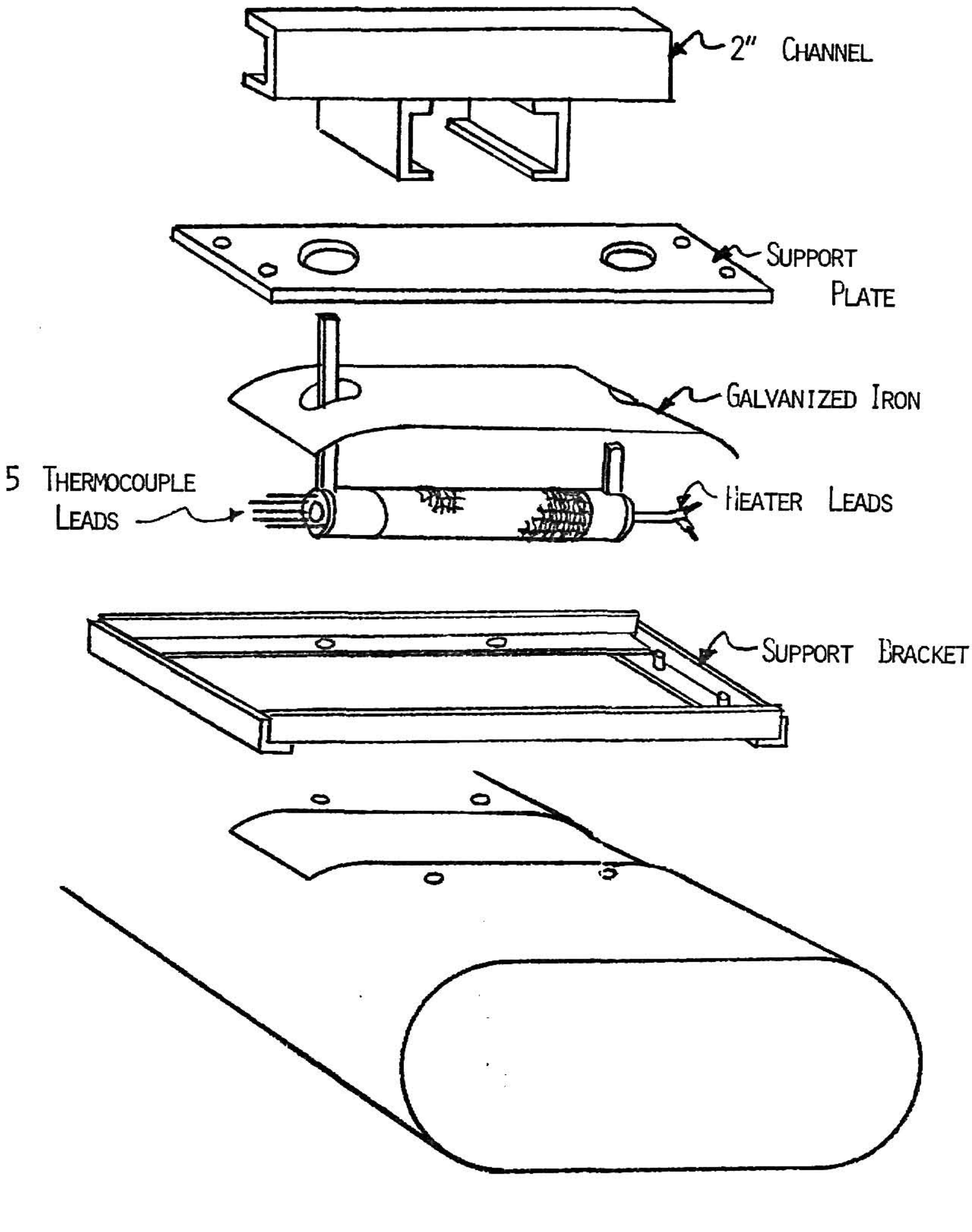


FIGURE 2. ISOMETRIC OF THE TEST SUPPORT SYSTEM

The 3-1/2 x 12 inch support plate was cut from a 3/16 inch steel sheet. A piece of 24 gauge galvanized iron was cut to 3-1/2 x 10-1/4 inches and formed to the contour of the top of the duct. The galvanized iron was attached to the bottom of the support plate. Two 1/4 inch holes were drilled in the support plate and galvanized iron to allow passage of the support arms. The support arms were 1/8 inch x 1/4 inch steel bars. Washers were brazed to the ends of the support arms and screws were used to secure the test section to the support arms. The other end of the support arms were bolted to a 2 inch channel. The 2 inch channel was spaced from the support plate by another piece of 2 inch channel. The free length of the support arms was 5 inches.

Drag Measurement

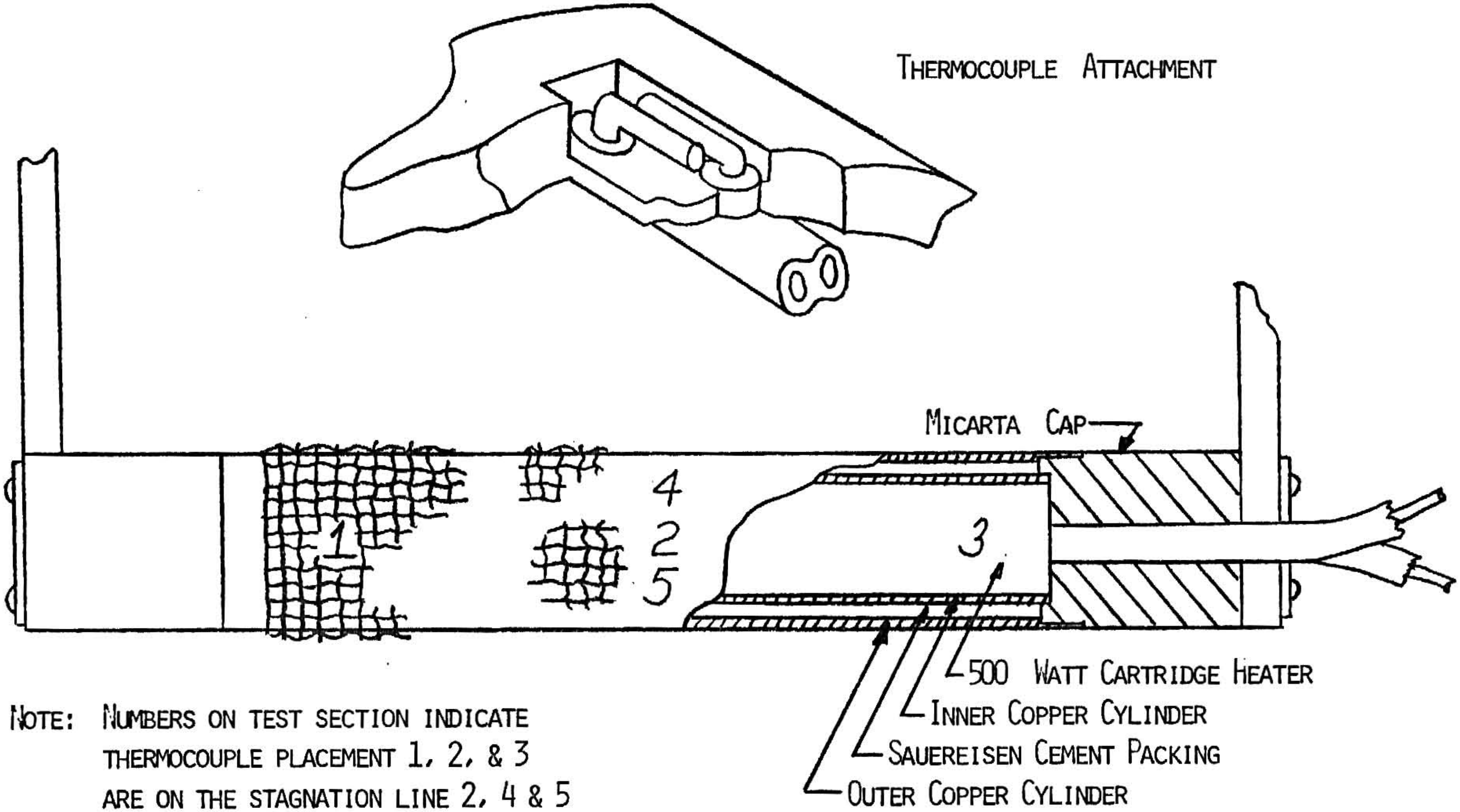
Two 4-active arm strain gage bridges were used to measure drag. The bridges were attached as close to the 2 inch channel as possible. Each bridge was attached to a Sanborn strain gauge amplifier. Amplifier output was recorded on a twin channel Sanborn Recorder.

Test Section

The test section assembly is shown in Fig. 3. Four different test cylinders were constructed. Three cylinders were instrumented with thermocouples.

Before screen wire was attached to the instrumented cylinders the bare cylinders were tested to obtain smooth cylinder heat transfer results, presented in Chapter III. Smooth cylinder drag results were obtained from tests run on the fourth cylinder.

For each test piece the outer cylinder was a 1-1/8 inch OD copper pipe cut to a 5-3/8 inch length. A 1/4 inch section at each end was machined



NOTE: NUMBERS ON TEST SECTION INDICATE THERMOCOUPLE PLACEMENT 1, 2, & 3 ARE ON THE STAGNATION LINE 2, 4 & 5 ARE ON THE CENTER LINE 120° APART.

FIGURE 3. TEST SECTION ASSEMBLY

to 1.0625 inch, ID. End caps were made of 1-1/2 inch thick micarta and machined to fit the cylinder.

Five thermocouples were attached to the surface of each cylinder. Three thermocouples were placed along the stagnation line; two were 3/4 inch from each end of the cylinder and one was on the center line. The remaining two thermocouples were also on the center line, 120° apart. Thermocouples were attached in the following manner: two 1/16 inch holes 3/16 inch apart were drilled at each thermocouple location, ceramic tube insulators were epoxied in each hole to prevent the bare wires from touching the cylinder, and a shallow slit was cut into the outer surface of the cylinder so that the thermocouple wires would lay flush with the surface. The wires were threaded through the insulators and the ends were laid in the slits and soldered in place. The excess solder was then filed off, leaving a smooth surface. The thermocouple leads from inside the cylinder were threaded through 1/16 inch diameter holes that had been drilled in the end cap.

The inner cylinder was a 13/16 inch OD thick walled copper pipe. The inside diameter of the inner cylinder was machined to 0.75 inches. This perimitted a 500 watt, 5 inch, commercial heater to be used interchangeably in the different test cylinders. The space between the inner and outer cylinders was filled with high thermal conductivity Sauereisen Cement, No. 29. The end cap through which the thermocouple wires had been threaded was cemented in place when the Sauereisen was placed between the cylinders. Thermocouple leads were thereby cemented in place.

Three copper wire screen sizes were used, 10 x 10, 20 x 20, and 30 x 30 (meshes to the inch). A 4-7/8 x 3-5/8 inch piece of wire screen was wrapped around a cylinder and centered. The screen was secured to the cylinder by tightly wrapping the cylinder screen combination with insulated resistance wire. A butane torch was used to heat the cylinder, while solid core solder was applied. Soldering flux was applied liberally to permit the solder to flow freely, leaving only the points of contact between the cylinder and screen soldered. Special care was taken to prevent the solder from filling the screen mesh.

Experimental Procedure

Data was taken in two parts, heat transfer data and drag data. Drag measurements were made after all the heat transfer work was completed. This enabled support drag to be reduced by removal of heater and thermocouple leads.

Heat Transfer Procedures

Preliminary steps included fixing the ice bath, standardizing the recording potentiometer with a Leeds and Northrup Millivolt Potentionmeter and zeroing the Merian micromanometer.

The blower motor was turned on and the slide damper set for a given flow rate. The air velocity was measured with the pitot-static tube. The pitot tube was moved to the duct wall and power was applied to the test section heater. Power was adjusted to bring the outside surface of the cylinder to approximately 200°F. After temperature equilibrium was reached, voltage, current and thermocouple indications were recorded. The flow velocity was checked and the temperature of the air stream was recorded.

This procedure was followed for all data runs. Preliminary steps were repeated every time the test cylinder was changed.

Drag Procedure

The strain gage amplifiers were zeroed and the support arms individually calibrated. Calibration set-up is shown in Fig. 4. Calibrated scale weights were placed in the bucket and scale deflections were recorded. After calibration the amplifiers were rechecked for zero.

A test cylinder was placed in the duct and drag measurement over the range of Reynold's numbers were made. The amplifiers were checked for zero after each cylinder was run. If any zero shift was detected the amplifier was rezeroed and the cylinder retested.

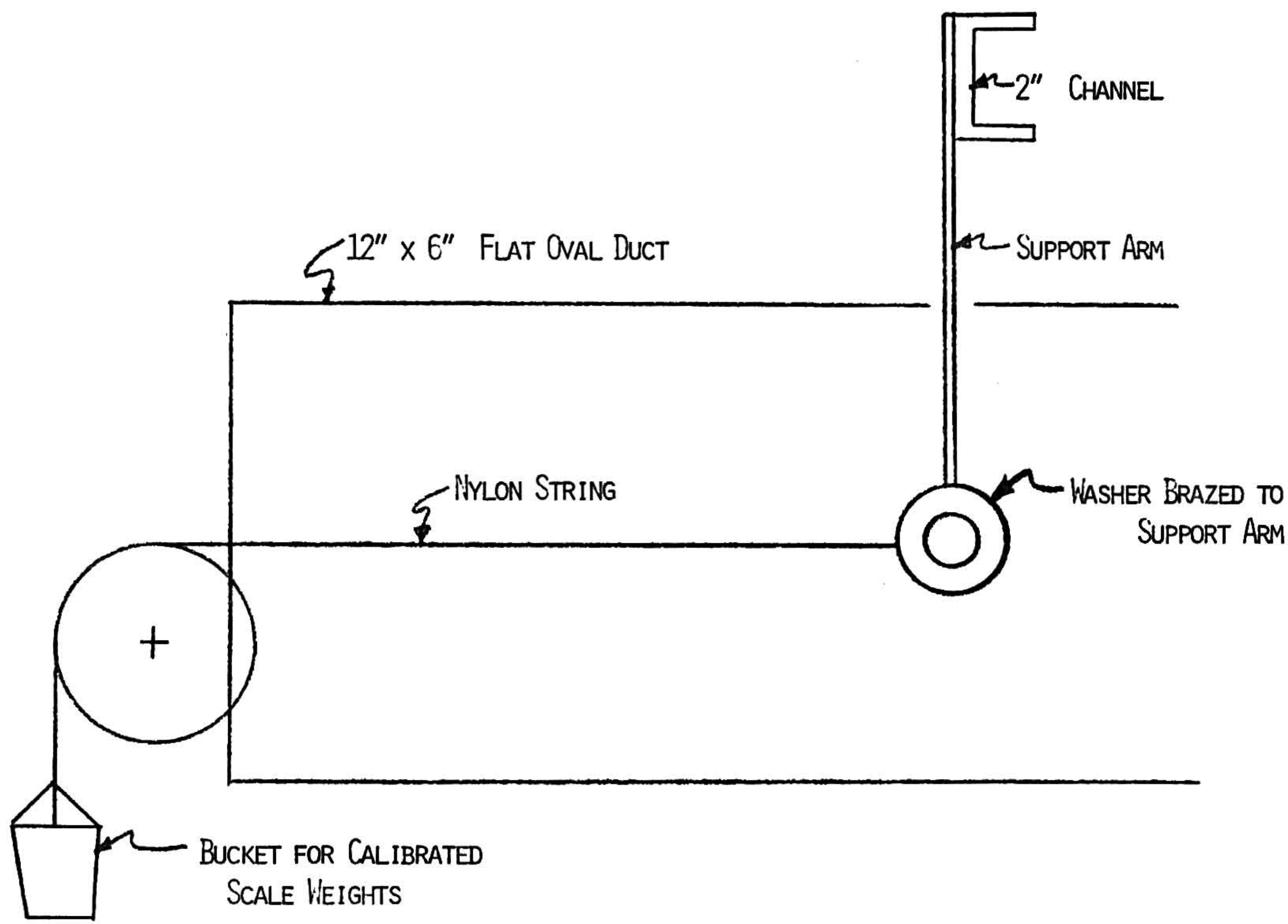


FIGURE 4. STRAIN GAGE CALIBRATION SETUP

CHAPTER III

EXPERIMENTAL RESULTS & DISCUSSION OF RESULTS

To provide a check on the validity of the results of the experiment, results from smooth cylinders were compared to the established literature correlation (3). This comparison is shown in Fig. 5, a log log plot of Nusselt number (Nu) versus Reynolds number (Re). For the experimental results reported here Nu was calculated from

$$Nu = \frac{hD}{K} \tag{1}$$

D was taken as the smooth cylinder diameter. K, the thermal conductivity was taken at the average film temperature (4). The surface conductance (h) was calculated from

$$h = \frac{Q}{A(T_{cyl} - T_{air})} \tag{2}$$

Where Q = Power dissipated from cylinder;

A = smooth cylinder area to be covered by screen;

T_{cyl} = arithmetic average of the five measured surface temperatures;

T_{air} = temperature of the moving air.

Re was calculated from

$$Re = \frac{VD}{\nu} \tag{3}$$

D was the same value used in equation 1. The Kinematic viscosity (ν was taken at the average film temperature (4).

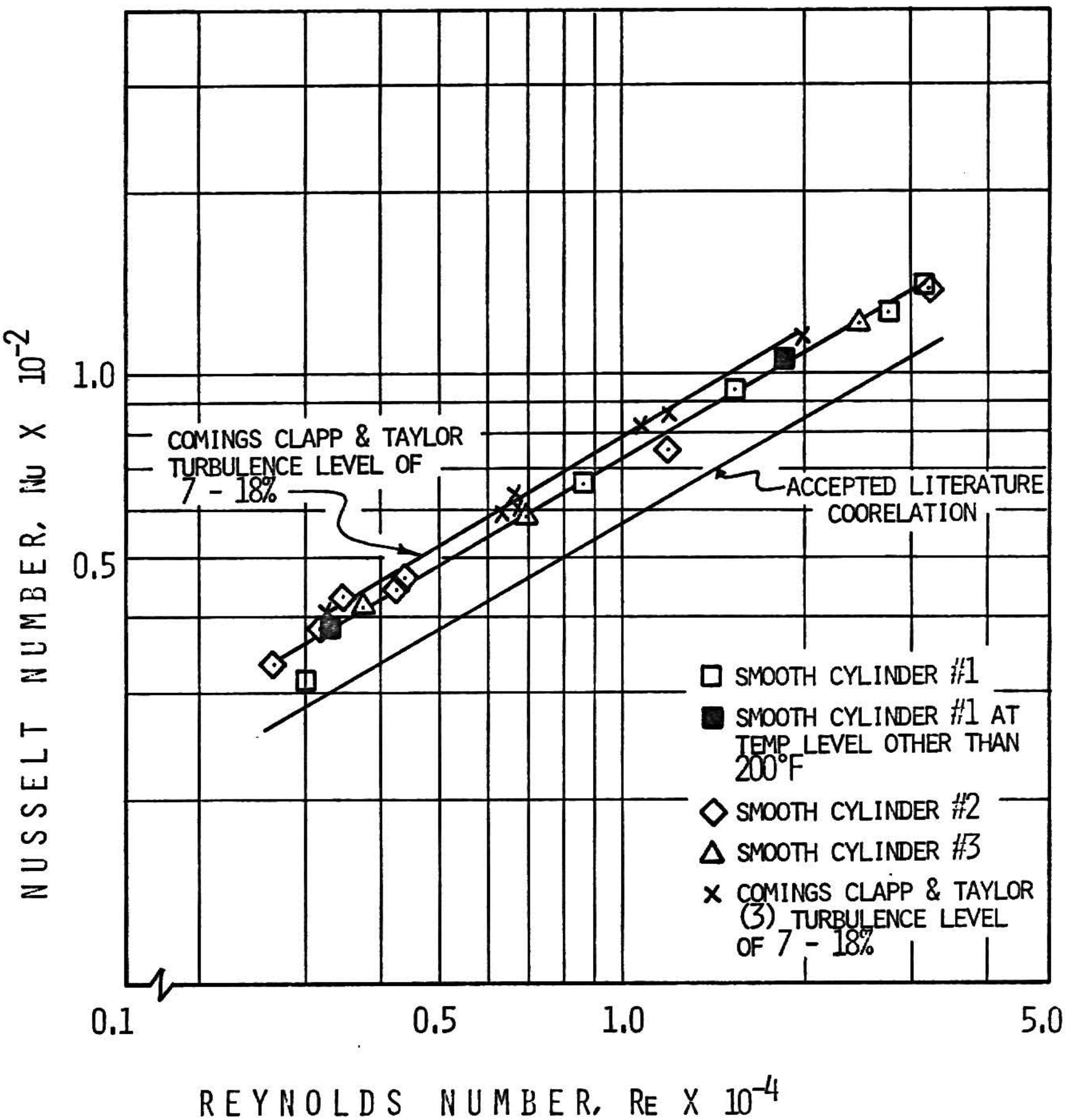


Figure 5. Smooth Cylinder Nusselt Number Results Compared to Published Results

Velocity (V) was calculated from

$$V = 67.5\sqrt{\Delta P} \tag{4}$$

The velocity pressure (ΔP) was in inches of water read from the Meriam micromanometer.

The experimental points lie on a line 22% higher than the accepted correlation line. The maximum experimental uncertainty in Nusselt number was estimated to be 6%. This difference between experimental values and correlation curve values can be explained by differences in free steam turbulence levels. The literature curve is for turbulence levels of 1 - 2% (5). Higher turbulence levels yield higher Nu. For example, data taken at turbulence level of 7 - 18% by Comings, Clapp and Taylor (3) show increased Nu by 40%.

In addition to verification of the literature curve Figure 5 shows there is negligible difference in results from the three smooth cylinders. The smooth cylinder data was taken with average cylinder temperature of 200°F, with the exception of runs which were made at temperature levels of 225 and 150°F. The results demonstrate that temperature effect was negligible.

Figure 6 is a log log plot of the experimental values of Nu versus Re for the three screen fitted cylinders. The smooth cylinder line represents the experimental results taken from Fig. 5 for comparison. The curves for all three screen sizes shown an increase in Nu, particularly at high Re. The greatest increases in Nusselt Numbers are obtained from the 10 x 10 mesh covered cylinder. At a Re of 32,000 Nu was increased by 82% over the smooth cylinder value. If this increase were to be accomplished by a diameter increase instead of by use of the 10 x 10 screen the tube diameter would have to be 1.82 times the tested diameter. At the low end of the range the curves approach the Nu value obtained for the smooth cylinders.

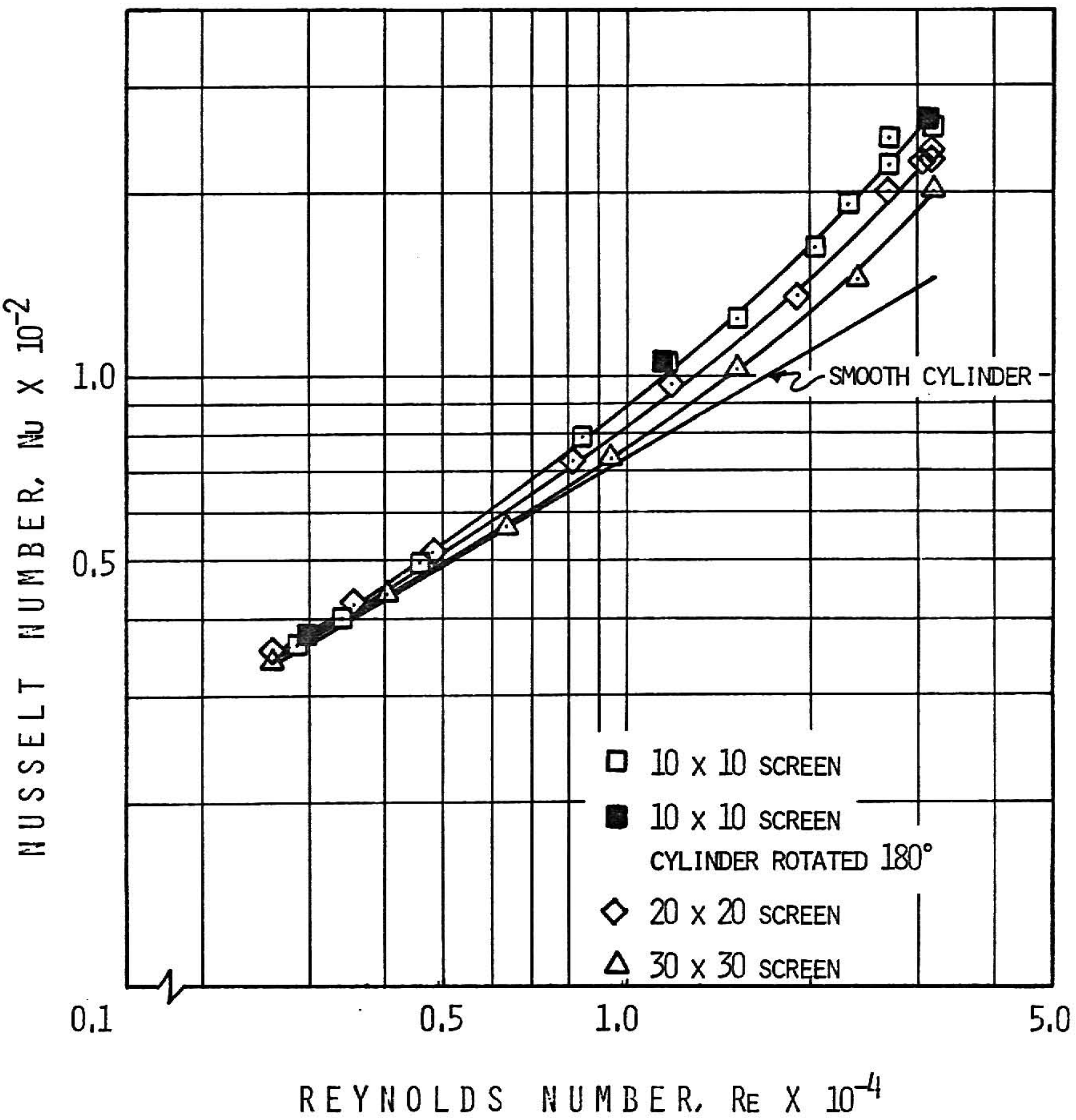


FIGURE 6. COMPARISON OF NUSSELT NUMBER RESULTS: SMOOTH CYLINDER AND CYLINDERS WITH SCREEN

Three runs were made with the coarse (10 x 10 mesh) screen with the cylinder turned 180°. There was no effect in results due to the changed position.

The area term in equation 2 was taken to be the screen covered area of the cylinder. This left a 1/4 inch band at each end. This area was neglected in the computations because it touched only the insulating end plugs and not the heated, cement filled annular region. Heat was transferred to the band by lateral conduction only. The cross sectional area of the band was decreased by machining to decrease this conduction.

It was expected that the addition of surface roughness would increase drag on the cylinder as well as increasing the heat transfer rate.

Figure 7 shows a log log plot of experimental drag coefficient (C_D) versus Re. C_D calculated from

$$C_D = \frac{\text{Drag}/A}{1/2\rho V^2} \tag{5}$$

Drag was measured by the strain gage system described in Chapter II. V was the same value as used in equation 3. A was the cylinder projected area, diameter times length. The drag data was taken with unheated cylinders so the density, ρ, was taken at free stream temperature. Results for the smooth experimental cylinder lie on a line with C_D approximately 1.66. Literature shows C_D for a smooth cylinder to be from 1.0 to 1.2 (4, 5). This difference between these values and the experimental value was probably due to the drag on the support pieces. Since support drag was present in all runs during the experiment, comparison for C_D for screened cylinders to smooth cylinders is possible. The magnitude of C_D is known to be incorrect, but increases in C_D due to the screen covering are detectable by this method.

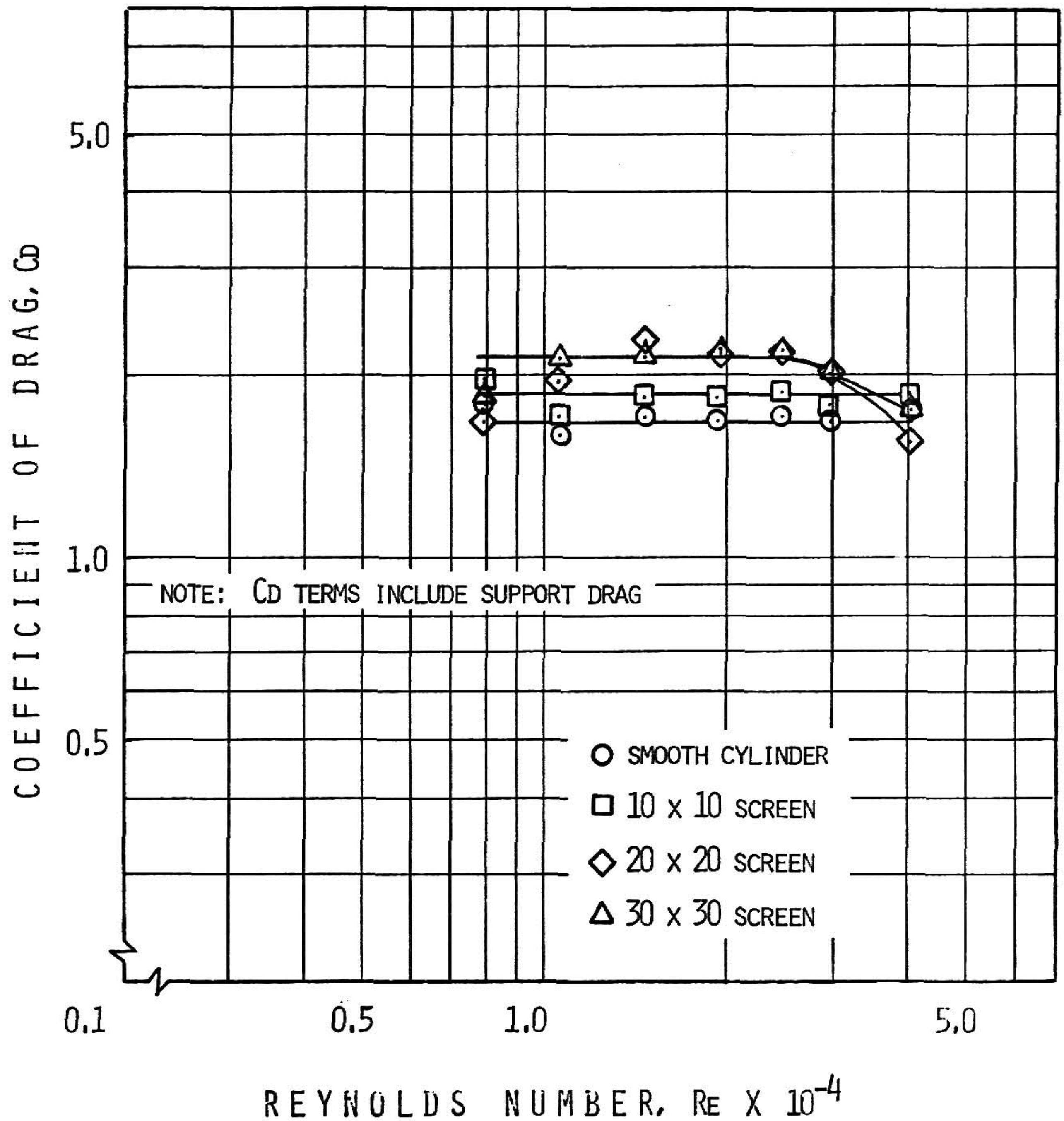


FIGURE 7. COMPARISON OF COEFFICIENT OF DRAG RESULTS, SMOOTH CYLINDER AND CYLINDERS WITH SCREEN

It was surprising to find that for the majority of the range tested, the 10 x 10 mesh gave the least drag of the screened cylinders. It would be expected that, since the coarse screen had the largest increase in Nu, it would also have the highest drag. At a Re of 40,000 the 10 x 10 screen had the highest C_D and the 20 x 20 and 30 x 30 had respectively lower values. Unfortunately, the heat transfer data does not extend into this range. It is not known if the Nusselt Number curves of Fig. 6 will continue upward in the higher Reynolds number range. There was not much difference between the values of C_D for the 20 x 20 and 30 x 30 mesh screened cylinders for Re below 25,000.

The break in the graph for the smaller two screen sizes at higher Re was not unexpected. Fage and Warsap (6) showed this effect on circular cylinders with varying roughness. Their work indicates that if the Re ranges were extended, the 10 x 10 mesh screen and the smooth cylinder curves would also bend downward.

Major concern in heat exchangers is to increase Nu without causing correspondingly higher increases in C_D. To evaluate the relationship of Nusselt number and drag, an efficiency term η was calculated and plotted against Re. These results are shown on the graph in Figure 8. The efficiency term was calculated from

$$\eta = \frac{\dfrac{Nu_{screen}}{Nu_{smooth}}}{\dfrac{C_{D\ screen}}{C_{D\ smooth}}}$$

where Nu screen and Nu_{smooth} were read from the curves of Fig. 6. The drag terms, $C_{D_{screen}}$ and $C_{D_{smooth}}$, were determined by reading C_D's from Fig. 7. From this formula the smooth cylinder always has an efficiency of 100%. The

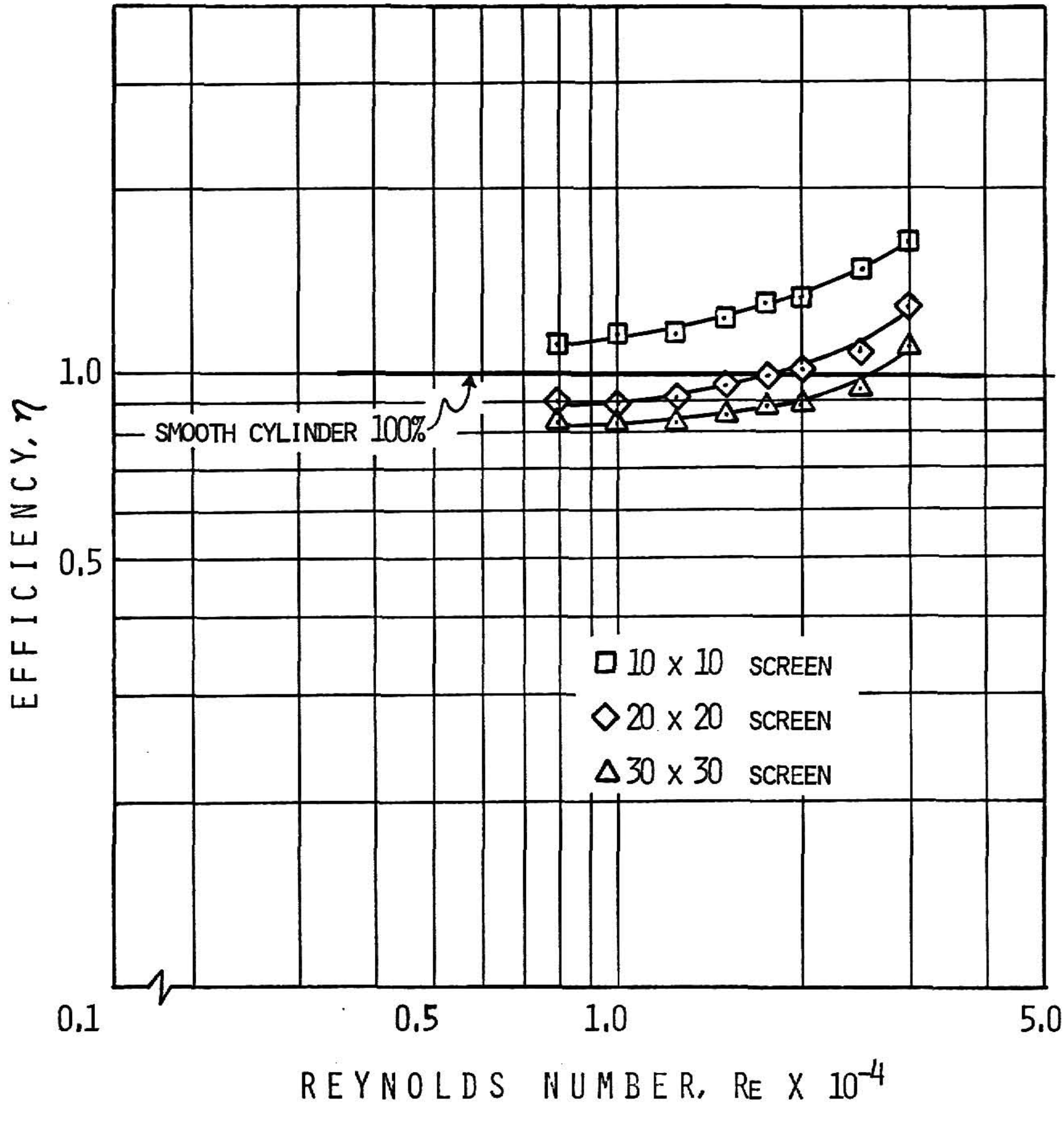


FIGURE 8. COMPARISON OF EFFICIENCY RESULTS: SMOOTH CYLINDER AND CYLINDERS WITH SCREEN

other cylinders are evaluated by comparing their efficiencies to the 100% smooth cylinder value. Figure 8 shows the 10 x 10 mesh screen to have the highest efficiency of all the screens tested. The 20 x 20 and 30 x 30 cross the 100% line at a Re of 17,500 and 27,500 respectively. At a Re of 30,000 the 10 x 10 has a 60% higher efficiency than the smooth cylinder. This indicates that a heat exchanger using 10 x 10 screen wire covered tubes could be 27.5% smaller than a heat exchanger using smooth tubes of the same diameter for the same pressure drop.

Probable Error

Great care was taken to minimize experimental error. Instruments were read carefully and data was taken only when it was evident that the test section had reached equilibrium. An uncertainty study of the equations used to calculate Nu and Re estimates a maximum error in Nu and Re to be 6% and 2% respectively.

End losses were determined to be negligible. Conduction loss from the 1/4 inch copper band at each end of the outside cylinder was calculated to be 3%. Radiation losses were calculated to be 5% at the low Re ranges and less than 1% at high Re. Since the experiment was run at constant temperature, radiation losses remain constant for all runs.

It was desirable to have only the points of contact between the cylinder and the wire screen soldered. It was estimated that 80% of the screen was attached properly. The 20% of the incorrect attachment included contact points that were not soldered and meshes that filled with solder. It is not known what the effect would be if 100% correct attachment were achieved. However, it is speculated that Nu values would increase for screened cylinders.

A velocity profile was taken across the duct for the first four runs. The velocity variation across the test section was 1% from the measured value. This was in accordance with Mayhill (7), who did a study on this particular oval duct.

Drag measurements were not taken for lower Re range because the uncertainty of the measurement would be equal to or greater than the magnitude of the measurement.

CHAPTER IV

CONCLUSIONS AND RECOMMENDATIONS

From the results of the experiment it is seen that application of wire screen roughness is an effective method of enhancing the heat transfer rate of tubes in crossflow. The three screen roughened cylinders showed dramatic increases in heat transfer with a minimal increase in drag, increases in Nusselt number up to 80% over smooth cylinder values having been obtained.

The 22% difference between the experimental smooth cylinder results and the accepted literature curve is well within the range of differences which have been reported elsewhere. Kreith (4) states that a wire grid (such as the one used as a flow straightener in the duct in this experiment) placed upstream of cylinders can increase Nu for the cylinders by as much as 50%. Comings, Clapp and Taylor (3) reported similar findings, Nu increases of 40% being measured for 7 - 13% turbulence levels. The experimental smooth cylinder Nu are within this range. Since the purpose of the experiment was to test and compare different screened cylinders to a smooth cylinder, the level of Nu is less important than is the increase in Nu due to application of the screen.

A separate study of drag coefficients over screened cylinders is suggested. The fact that the 10 x 10 meshed cylinder showed the greatest increase in Nu and the smallest increase in drag is inconsistent with reasoning based on momentum--heat transfer analogies. Further investigation of this phenomena seems of sufficient interest to merit additional work.

It would be of interest to extend the Re range past 10^5. The Nu curves of Figure 6 indicate that further increases in Nu can be expected in the Re range beyond 10^5. This study would require a different flow system as it would exceed the capability of the present equipment. It is probable that a Nu limit will be reached or that drag will increase at a rate faster than Nu, such that the efficiency values will begin to decrease.

In future studies different mesh sizes, 5 x 5 or 5 x 2 meshes to the inch could be compared, since this experiment indicates that the coarser mesh is the most efficient. There are endless numbers of different weave patterns that would be desirable for study. Comparison of screen wire roughness elements to other types (single wires, discrete elements, etc.) would be of additional interest and would be necessary for selection of a "best" type.

REFERENCES

1. Grigull, Hahne, Heat Transfer 1970. Volume Three, Amsterdam, Elsevier Publishing Co., 1970.

2. Bergles, Webb, "Augmentation of Convective Heat and Mass Transfer", New York, ASME United Engineering Center, 1970.

3. Knudsen and Katz, Fluid Dynamics and Heat Transfer, New York, McGraw Hill, 1958.

4. Kreith, Principles of Heat Transfer, Scranton Pennsylvania, International Textbook Co., 1969.

5. Parker, Boggs, and Blick, Introduction to Fluid Mechanics and Heat Transfer, Readings, Massachusetts, Addison Wesley, 1969.

6. Schlichting, Hag, Boundary Layer Theory, New York, McGraw Hill, 1955.

7. Mayhill, Terrence D., "A Simplified Method for Measurement of Air Flow Rates in Flat Oval Ducts." (Unpublished Master's Report, Kansas State University, 1971).

APPENDIX

APPENDIX A

REDUCED DATA for HEAT TRANSFER RUNS

Current (amperes)	Voltage (volts)	Velocity Head in. H_2O	Air Temp (°F)	Test Cylinder Temp °F 1	2	3	4	5	Nu	Re (X10-3)
			Smooth Cylinder #1							
1.04	100.0	1.022	80.0	194	201	196	209	207	141.8	31.10
.960	96.0	.752	80.0	194	201	196	206	204	127.0	27.00
.665	66.0	.35	82.0	147	151	148	153	153	105.5	18.39
.840	82.0	.238	79.5	195	201	197	204	203	95.2	15.12
.680	69.0	.075	80.0	194	199	195	201	201	66.2	8.54
.578	58.0	.011	83.0	225	228	227	230	229	39.1	3.24
.483	48.0	.009	80.0	197	199	197	201	200	31.4	2.96
			Smooth Cylinder #2							
1.00	100.0	1.052	85.5	197	206	202	210	210	139.5	31.80
.730	73.0	.142	84.5	196	201	200	203	203	75.6	11.72
.568	57.0	.022	86.0	196	200	198	202	202	46.8	4.61
.549	55.0	.018	86.0	194	198	196	200	200	44.6	4.15
.548	55.0	.012	86.0	199	203	202	205	205	43.6	3.40
.510	51.0	.010	86.0	196	199	197	200	200	38.3	3.11
.484	48.0	.007	86.0	198	201	200	202	202	33.5	3.63
			Smooth Cylinder #3							
.929	93.0	.603	85.0	195	203	197	206	206	122.2	24.20
.640	64.0	.044	86.0	199	205	200	205	205	58.5	6.82
.530	53.0	.014	87.0	195	198	196	200	200	41.4	3.67
			10 x 10 Mesh Screen; Cylinder Rotated 180°							
1.38	138.0	1.045	79.0	191	202	203	194	196	266.0	31.80
.878	88.0	.139	78.0	192	199	199	197	197	107.2	11.58
.530	53.0	.008	78.5	198	200	200	199	200	38.0	2.92

APPENDIX A

REDUCED DATA for HEAT TRANSFER RUNS (Continued)

Current (amperes)	Voltage (volts)	Velocity Head in. H_2O	Air Temp (°F)	Test Cylinder Temp °F 1	2	3	4	5	Nu	Re (X10-3)
			10 x 10 Mesh Screen							
1.37	139.0	1.060	83.0	193	201	203	209	211	258.2	31.95
1.27	128.0	.751	82.0	190	201	201	207	210	224.0	26.90
1.19	120.0	.562	84.0	196	206	207	212	214	191.5	23.25
1.08	110.0	.428	83.5	195	204	204	209	211	162.6	20.30
.950	96.0	.238	82.0	194	199	199	205	218	124.6	15.10
.850	85.1	.142	84.0	192	198	198	201	202	106.8	11.70
.762	78.2	.073	82.0	199	204	204	207	208	80.2	8.37
.600	60.0	.022	84.5	197	200	200	205	207	50.2	4.61
.530	53.0	.012	84.0	194	196	195	197	198	40.5	3.40
.520	52.0	.008	82.0	198	201	200	202	202	36.4	2.89
			20 x 20 Mesh Screen							
1.10	110.0	1.035	86.0	156	166	166	177	175	236.0	31.60
1.29	129.5	1.035	86.5	188	204	204	209	210	234.0	31.60
1.28	128.1	.980	86.5	192	205	205	211	214	228.5	30.70
1.20	120.0	.760	85.5	191	204	208	210	210	202.0	27.0
.970	97.0	.343	85.0	191	197	201	203	205	135.9	19.20
.830	83.0	.147	84.5	190	204	204	206	208	96.5	11.90
.725	72.5	.074	85.5	198	204	204	206	207	73.4	8.45
.610	61.0	.024	86.0	199	203	203	205	205	52.6	4.83
.540	54.0	.013	86.5	196	199	199	201	201	42.7	3.54
.500	50.5	.007	86.0	200	203	203	205	205	35.2	2.60

APPENDIX A

REDUCED DATA for HEAT TRANSFER RUNS (Continued)

Current (amperes)	Voltage (volts)	Velocity Head in. H_2O	Air Temp (°F)	Test Cylinder Temp °F 1	2	3	4	5	Nu	Re (X10-3)
30 x 30 Mesh Screen										
1.20	120.5	1.052	85.5	192	204	199	207	210	202.0	31.90
1.03	103.0	.59	84.0	196	208	203	210	211	144.4	23.80
.868	87.0	.245	83.5	197	206	202	208	206	103.8	15.35
.720	72.0	.094	84.5	195	201	197	202	203	74.3	9.58
.670	67.0	.042	85.5	205	211	208	210	212	58.2	6.39
.550	55.0	.016	85.5	195	199	197	200	199	44.3	4.02
.480	48.0	.007	86.0	194	197	196	197	198	34.4	2.63
.480	48.0	.007	85.5	195	198	196	199	199	34.7	2.63

APPENDIX B

REDUCED DATA FOR DRAG RUNS

Velocity Head in. H_2O	Measured Drag lbf x 10^{-2}	C_D	Re $X10^{-3}$
	Smooth Cylinder		
1.075	3.20	1.76	39.0
.574	3.02	1.66	28.60
.428	3.09	1.70	23.75
.241	3.02	1.66	18.46
.139	3.09	1.70	14.03
.073	2.87	1.58	10.15
.041	3.22	1.78	7.58
	10 x 10 Mesh Screen		
1.075	3.40	1.87	39.00
.574	3.20	1.76	28.60
.428	3.42	1.88	23.75
.241	3.31	1.82	18.96
.139	3.31	1.82	14.03
.073	3.11	1.71	10.15
.041	3.58	1.97	7.58
	20 x 20 Mesh Screen		
1.075	3.20	1.76	39.00
.574	3.71	2.02	28.60
.428	3.89	2.14	23.75
.241	3.98	2.19	18.46
.139	3.89	2.14	14.03
.073	3.84	2.11	10.15
.041	3.37	1.85	7.58
	30 x 30 Increasing Re		
1.075	2.84	1.56	39.00
.574	3.60	1.98	28.60
.428	4.13	2.27	23.75
.241	3.68	2.20	18.46
.139	3.96	2.18	14.03
.073	3.86	2.12	10.15
.041	3.35	1.84	7.58

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VITA

John Thomas Ratcliffe

Master of Science

Thesis: EXPERIMENTAL STUDY OF HEAT TRANSFER AUGMENTATION BY SCREEN WIRE ROUGHNESS ON CYLINDERS IN CROSS FLOW

Major Field: Mechanical Engineering

Biographical:

Personal Data: Born in El Paso, Texas, December 10, 1947, the son of Lamar C. and Ruth W. Ratcliffe.

Education: Graduated from Manhattan High School, Manhattan, Kansas in May 1965; received a Bachelor of Science degree in Mechanical Engineering from Kansas State University, May 1970; completed requirements for the Masters of Science degree in Mechanical Engineering in March 1972.

Professional Experience: Worked as a part time teachers assistant in the Department of Mathematics, KSU, from February 1966 to May 1966 and from September 1966 to January 1967; was employed by NASA's Flight Research Center, Edwards Air Force Base, California as an Engineer Trainee, COOP student, from February 1967 to August 1969; since September 1970, except for the summer period in 1971, was employed by Mechanical Engineering Department at Kansas State University as a Graduate Research Assistant.

EXPERIMENTAL STUDY OF HEAT TRANSFER AUGMENTATION BY SCREEN WIRE ROUGHNESS ON CYLINDERS IN CROSS FLOW

by

JOHN THOMAS RATCLIFFE

B. S., Kansas State University, 1970

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

1972

Name: John Thomas Ratcliffe Date of Degree: 1972

Institution: Kansas State University Location: Manhattan, Kansas

Title of Study: EXPERIMENTAL STUDY OF HEAT TRANSFER AUGMENTATION BY SCREEN WIRE ROUGHNESS ON CYLINDERS IN CROSS FLOW

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

Major Field: Mechanical Engineering

Scope and Method of Study: The most widely accepted method of increasing heat transfer from tubes in cross flow is by annular finning. However, in some instances finning is impractical and uneconomical.

Because of the increasing number of applications which require improved heat transfer rates from unfinned tubes in crossflow a study was instituted to investigate one class of surface roughness and to determine its potential for heat transfer enhancement. Wire screen was selected for this experimental study because it has potential for application in practical heat exchangers. Three screen sizes were tested, 10 x 10, 20 x 20 and 30 x 30 meshes to the inch. The Reynolds number range tested was from 2,700 to 33,000. The 10 x 10 mesh showed the greatest increase in heat transfer rates, and, surprisingly, the least increase in drag. The Nusselt number for the 10 x 10 mesh was increased by as much as 80% over smooth tube values at the same Reynold's number. All three screen sizes tested showed significant increases in heat transfer, with a minimal increase in drag.

MAJOR PROFESSOR'S APPROVAL Robert L. Gorton