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TUBING FOR AUGMENTED HEAT TRANSFER

Final Report

By
J. S. Yampolsky
P. Pavlics

August 1, 1983

Work Performed Under Contract No. AC22-79ET15210

GA Technologies, Inc.
San Diego, California

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TUBING FOR AUGMENTED HEAT TRANSFER

by
J. S. YAMPOLSKY and P. PAVLICS

Prepared under
Contract DE-AC22-79ET15210
for the **Pittsburgh Energy Technology Center**
Department of Energy

GA PROJECT 3317
AUGUST 1, 1983

GA Technologies

1. INTRODUCTION AND BACKGROUND

Heat exchangers represent a significant part of the investment in a wide variety of chemical and mechanical processes and power plants. Their importance increases when the recovery of waste heat from these processes is required. If economics is the criteria for plant or systems selections, then improving the performance and reducing the cost of heat exchange equipment can lead to increased energy conservation.

The objectives of this program were twofold; first to determine the heat transfer and friction characteristics on the outside of spiral fluted tubing in single phase flow of water, and second to assess the relative cost of a heat exchanger constructed with spiral fluted tubing with one using conventional smooth tubing. An energy conservation application was used for this study.

Spiral fluted tubing have the potential for materially reducing the cost of heat exchange equipment because they enhance the heat transfer coefficient, extend the heat transfer area, and are fabricated by low cost, high-volume manufacturing processes.

The fabrication technique of rolling flutes on a flat strip, subsequently spiralling and simultaneously welding the corrugated strip to form tubing, results in low fabrication costs approximately equal to those of commercially welded tubing.

The heat transfer coefficient is increased without a concomitant increase in the friction coefficient. The physical process is that in single phase flow of liquid, the spiral-flutes continuously induce rotation of the flow both inside and outside the tube as a result of the curvature of the flutes. This rotation enhances turbulent exchange both on the inside and outside of the tube in the immediate vicinity of the wall. When heat is transferred from a high temperature fluid external to the tube through the tube wall to a colder fluid in the tube, the density gradients established close to the walls are destabilizing in the presence of rotation. Since it

is widely recognized that the resistance to the transmission of heat (or any other scalar) from the wall to the internal flow in a tube is in the laminar sublayer adjacent to the wall where the process is dependent on the molecular heat conductivity, instabilities in the vicinity of the tube wall should increase the diffusivity of heat. Normally, improvements in the rate of radial heat transport in a tube resulting from increases in the turbulence level in the flow results in increases in the momentum losses relative to the increase in heat transfer. However, there are examples of instabilities in the atmospheric boundary layer that result in substantial increase of the thermal diffusivity relative to the momentum diffusivity.

In addition to the enhancement of the heat transfer coefficient relative to the friction coefficient, the fluting increases the perimeter of the tube, thereby increasing the heat exchange area.

In a previously completed DOE supported program, (1) measurements of the heat transfer and friction characteristics on the inside of a spiral fluted tube were made by W. J. Marner of Heat Transfer Research, Inc. These results are summarized below together with those of D. J. Reilly. (2)

The heat transfer performance on the inside of helically fluted tubes as indicated by the Nusselt modulus, $Nu/Pr^{0.4}$, as a function of the Reynolds number is shown in Fig. 1. The Nusselt number is defined as $Nu = h d_{hyd}/k$, where h is the conductance coefficient (Btu/hr-ft²-°F), d_{hyd} is the hydraulic diameter (ft) and is equal to $4 \times \text{cross-sectional area (ft}^2\text{)}/\text{perimeter (ft)}$, and k is the thermal conductivity of the water (Btu/hr-°F-ft.). The Reynolds number is defined as $Re = v d_{hyd}/\nu$, where v is the mean axial velocity (ft/sec) and ν is the kinematic viscosity (ft²/sec). The Prandtl number is $Pr = c_p \mu/k$, where c_p is the specific heat at constant pressure and μ is the absolute viscosity. The Dittus Boelter correlation, $Nu/Pr^{0.4} = 0.023 Re^{0.8}$, for a smooth tube is shown for comparison.

The ratio of the ordinates for the four curves shown in Fig. 1 to that of the smooth tube curve indicates the amount of heat transfer enhancement achieved on the inside of the helically fluted tubes relative to a smooth tube. These curves do not include the effect of the extended area of the

Figure 1

**ENHANCEMENT OF 30° HELIX FOR VARIED HEAT FLUX
(DOES NOT INCLUDE AREA EXTENSION)**

helically fluted tubes relative to that of a smooth tube of the same cross-sectional area. This is an additional factor of 1.33 for the tube reported here; in some of the tubes under development the extended area is 1.6 greater than the area of a smooth tube of equal cross-sectional area. Curves 1, 3 and 4 are the results from tests conducted by Heat Transfer Research, Inc. (HTRI), on a 1 in. o.d. tube that had flutes with a 30° helix angle and was approximately 7 ft. long. (The helix angle is defined as the angle the helix makes with the tube axis.) Curve 1 is for cooling; that is, the direction of heat flow is radially outward. Hot water flowing on the inside of the tube was cooled by colder water flowing on the outside. Although enhancement is achieved in this mode of operation, it is less than in the other curves since the density gradient is stabilizing.

Curves 3 and 4, however, show significant enhancement. These data were obtained with the direction of the heat flow radially inward, which leads to density gradients that are destabilizing and hence increases turbulent exchange. Steam was condensed on the outside of the tube by colder water flowing on the inside. The condensing pressure was approximately 9 atm for curve 4 and 5 atm for curve 3 in these tests. The steam flow was parallel to the tube axis, which was horizontal. Data from two tests conducted by D. J. Reilly at the Naval Postgraduate School(2) on a 5/8-in. o.d., 30° helix angle spirally fluted tube are shown by curve 2. In these tests steam at 0.25 atm flowed transverse to the tube axis, which was horizontal. Curves 2, 3 and 4 indicate that as the condensing pressure and hence temperature were increased relative to the tube-side coolant, heat transfer increased. The higher condensing temperatures indicate higher heat fluxes and result in larger temperature gradients on the inside of the tube.

The frictional performance of the 30° helix angle tube can be seen in Fig. 2, where the isothermal friction coefficient (Moody definition) is plotted against Reynolds number for both the HTRI tests (indicated by the line) and the Reilly tests (indicated by the triangles). The hydraulic diameter is used in the calculation for the frictional coefficient. These measurements were made on heat transfer test rigs and questions on their validity could be raised.(1) They were repeated in this program and are reported herein.

Figure 2

MOODY FRICTION COEFFICIENT VS. REYNOLDS NUMBER FOR HELICALLY FLUTED TUBE

2. DESIGN AND MANUFACTURE OF 7-TUBE BUNDLE

Performance test data on the shell side of heat exchangers require that the measurements be made in a field of similar tubes. This required a tube bundle of at least 7 tubes for a triangular pitch array or nine tubes for a square pitch array. A 60° triangular pitch array was selected so that instrumentation could be located around the center tube of the array.

The design of the 7-tube heat exchanger is shown in Fig. 3. A 5" schedule 80 aluminum pipe was used as the body of the shell. A 5" schedule 90 tee was welded to each end of the shell. The tube side nozzles were weld neck flanges, welded to extensions of the axial ends of the tees, and the shell side nozzles were made by welding slip on flanges to pipe extensions welded to the stem of the tees. The tube bundle consists of a fixed and floating tube sheet to which the tubes are welded. The top tube sheet is the fixed one and is trapped between the flanges of the shell and the piping and sealed by 2 concentric rubber O-rings on the flange faces. The bottom tube sheet is floating and is sealed by rubber O-rings on the machined ID of the weld end flange of the shell. The measuring stations are shown into and out of both the tube side and shell side of the exchanger and at 3 axial stations for the shell side flow along the tube length.

The details of the tubing is shown in Fig. 4. The flutes are of the same shape and spacing as was used previously(1) but the depth has been increased to compensate for the slightly larger 1.25" diameter tubing used. Short spacers were welded to the tubes in the flutes to prevent vibration. Since the helix of each tube is opposite to each of its neighbors, there was only a point contact of the adjacent spacers as each was set in the flutes before welding. Extruded 6061 aluminum fluted strip as shown in detail 003 of Fig. 4 was used as the starting material for tube forming.

The tubes were formed from this strip on the GA forming mill shown in Fig. 5. The strip is driven by a set of driving rolls into a closed cylindrical die at the desired angle to form a spiral fluted tube. As the strip emerges from the cylindrical die, a set of 3 hydraulic motors exerts

NOTES (CONTINUED)

14. ALL WELDS SHALL BE LIQUID PENETRANT EXAMINED PER APPENDIX B OF SECTION III, DIV. 1. GA SPEC. Q01-PT-53801.

NOTES (CONTINUED)

15. RECORD MAXIMUM AND MINIMUM PIPE I.D. PRIOR TO HONING. HONE TO .005 OVER MAXIMUM I.D. RECORDED PREVIOUSLY. RECORD FINISHED DIMENSIONS. HONE ENTIRE LENGTH OF UNIT TO POINT (2).
16. INSULATION: SIZE AND TYPE TO BE DETERMINED BY EXPERIMENTAL ENGINEERING.
17. VIBRATION SUPPORT SYSTEM TO BE DETERMINED LATER.
18. PROVIDE 32 SURFACE FINISH BETWEEN 7.50 DIA AND 4.80 DIA ON BOTH SURFACES OF TUBESHEET.
19. WELDERS AND WELD PROCEDURES SHALL BE QUALIFIED IN ACCORDANCE WITH SECTION IX OF THE ASME BOILER AND PRESSURE VESSEL CODE.

NOTES

1. FABRICATED IN ACCORDANCE WITH ASME B.P.V. CODE SECTION VIII DIVISION 1. ALL WELDS FULL PENETRATION. NO CODE STAMP REQUIRED.
2. O-RINGS: PARKER SEAL CO. OR EQUAL
FLANGES: FLOWLINE CORP. OR EQUAL
TEE: FLOWLINE CORP. OR EQUAL
3. GROOVE DIMENSIONS AND TOLERANCES TO BE PER PARKER SEAL CO. SPECIFICATIONS OF EQUAL
4. DESIGN TEMPERATURE: 300 °F
DESIGN PRESSURE: 300 PSI
5. PRESSURE RELIEF VALVE TO BE SUPPLIED ON EXTERNAL PRESSURE SYSTEM
6. TUBE TO TUBESHEET WELD TO BE DETERMINED BY TEST
7. TIGHTEN TO 1250 LB/IN
8. HYDROSTATIC TEST AT 813 PSI AT 70 °F ± 2 °F

Figure 3

ITEM	QTY	PART NO.	DESCRIPTION	MAT/MATL SPEC
1	1	026382-001	TUBE, SPIRAL ASSEMBLY	
2	1	026382-002	INSULATION	
3	1	026382-003	1/8" NPT 150" X 1/2" DIA. TUBE	AL 6061-T6
4	1	026382-004	PIPE 5" SCH 80 X 21' LG	AL 6061-T6 SB-241
5	1	026382-005	PIPE 5" SCH 80 X 17 1/2' LG	AL 6061-T6 SB-241
6	1	026382-006	FLANGE 150 LB WELD NECK SCH 80	AL 6061-T6 SB-247
7	1	026382-007	WASHER 1/2" DIA X 1 1/2" THK	CARBON ST
8	1	026382-008	NUT 1/2" DIA X 1 1/2" THK	CARBON ST
9	1	026382-009	BOLT 3/4" DIA X 1 1/2" THK	CARBON ST
10	1	026382-010	FLANGE 150 LB WELD NECK SCH 80	AL 6061-T6 SB-247
11	1	026382-011	TEE 5" SCH 80 WELDED	AL 6061-T6 SB-241
12	1	026382-012	PIPE 5" SCH 80 X 122.75' LG	AL 6061-T6 SB-241
13	1	026382-013	FLANGE, SPECIAL 5" 150 LB	AL 6061-T6 SB-247
14	1	026382-014	TUBE, SPIRAL ASSEMBLY	
15	1	026382-015	TUBESHEET, LARGE - 2 1/4" THK	AL 6061-T6 SB-209
16	1	026382-016	TUBESHEET, SMALL - 2 1/4" THK	AL 6061-T6 SB-209
17	1	026382-017	ASSEMBLY	

REQUISITION		PARTS LIST	
REQD/ASSEMBLY	ITEM	DESCRIPTION	MAT/MATL SPEC
	1	026382-001	TUBE, SPIRAL ASSEMBLY
	2	026382-002	INSULATION
	3	026382-003	1/8" NPT 150" X 1/2" DIA. TUBE
	4	026382-004	PIPE 5" SCH 80 X 21' LG
	5	026382-005	PIPE 5" SCH 80 X 17 1/2' LG
	6	026382-006	FLANGE 150 LB WELD NECK SCH 80
	7	026382-007	WASHER 1/2" DIA X 1 1/2" THK
	8	026382-008	NUT 1/2" DIA X 1 1/2" THK
	9	026382-009	BOLT 3/4" DIA X 1 1/2" THK
	10	026382-010	FLANGE 150 LB WELD NECK SCH 80
	11	026382-011	TEE 5" SCH 80 WELDED
	12	026382-012	PIPE 5" SCH 80 X 122.75' LG
	13	026382-013	FLANGE, SPECIAL 5" 150 LB
	14	026382-014	TUBE, SPIRAL ASSEMBLY
	15	026382-015	TUBESHEET, LARGE - 2 1/4" THK
	16	026382-016	TUBESHEET, SMALL - 2 1/4" THK
	17	026382-017	ASSEMBLY

TITLE		7 TUBE HEAT EXCHANGER - ALUMINUM	
SIZE	CODE	IDENT NO	DWG. NO
E	32334	026335	
SCALE		1" = 1'-0"	
DATE		1-9-

026382

torque on the periphery of the formed tube to keep it tightly wound for welding. The welding is accomplished by a Tungsten Inert Gas (TIG) welding head with automatic voltage control. The welded tube is cut to the desired length after forming and welding. It is subsequently leak checked.

The tube bundle and shell were fabricated and assembled into the 7-tube heat exchanger by the Component Manufacturing Division of GA Technologies. Figure 6 is a photograph showing the assembly of the bundle into the shell. The O-ring seals on the flange face and the tube to tube sheet welds are seen in this view. The shell side tube field as seen through one of the nozzles on the shell is depicted by the photograph in Fig. 7. After assembly, and hydrastatic test, the heat exchanger was installed in the test facility for testing.

Figure 6 - Tube Bundle Being Assembled into Shell

Figure 7 - Tubing As Seen Through the Shell Side Nozzle

3. TESTS OF SEVEN TUBE HEAT EXCHANGER

3.1 Test Facility

The GA Circulator Test Facility was used for the heat exchanger test. In this facility there is a closed loop consisting of a motor driven 7 stage compressor and a heat exchanger for removing the heat of compression in the loop to a cooling tower. The test heat exchanger was installed in this cooling loop between the heat exchanger and the cooling tower. A pump and flow meter circulated and measured the water flow rate in the high temperature circuit between the heat exchanger in the loop, and the test heat exchanger and another pump and flow meter circulated the water in the low temperature circuit between the test heat exchanger and the cooling tower. Bypass valves around the pumps allowed the flow to be varied independently in each circuit. The arrangement is shown in Fig. 8. The test water pressures were of the order of 100 psig.

Both tube-side and shell-side water heating was achieved by using heat generated in the nitrogen gas compressor loop. Heat transfer from the nitrogen gas to the shell-side water took place in the heat exchanger denoted E-2 (Fig. 8). Tube-side and shell-side water cooling was accomplished by the dry cooling tower E-4.

3.2 Test Methodology

A series of tests were performed with the 7-tube heat exchanger in which the power level of the compressor was varied so as to vary the heat flux in the test heat exchanger. In each of these tests, first the tube-side water flow rate was held constant and then the shell-side water flow rate was varied from about 10,000 to 100,000 in Reynolds number. Alternately, the shell-side water flow rate was held constant and the tube-side water flow rate was varied from about 10,000 to 110,000 in Reynolds number. In all tests, except one, the heat transfer was from the shell-side fluid to the tube-side fluid. In this additional test the direction of heat flow was reversed, the heat transfer was from the tube-side to the shell-side. The

Fig. 8 - Test Facility

purpose of varying the heat flux and the direction of heat flow was to attempt to correlate the heat transfer enhancement with heat flux found previously.

The instrument measuring stations are shown in Fig. 9. The overall heat transfer performance on the tube-side was obtained from temperature measurements at stations T_1 and T_2 and on the shell-side from stations S_1 and S_5 . Temperature measurements were also made around the center tube in the tube field at three axial locations at stations S_2 , S_3 and S_4 . At each of these stations there were three chromel-alumel thermocouples which were recorded on a Doric Digitemd 210 recorder. All the thermocouples belonged to a matched set and it was determined by calibration that the temperature measuring system was shown to be consistent within 0.1°F .

The procedure followed in the testing was to set the water flow on both the shell-side and tube-side and then record a set of temperature data at minute intervals until the temperatures stabilized. It took approximately 7 seconds for each temperature data set. A heat balance between the tube-side and shell-side was then used as the criterium for the acceptability of the data. In most of the data used in the analysis, the heat balances were within 5% but some were accepted up to approximately 10%. One of the flows was then varied through approximately a 10:1 range of flow while the other was held constant. This flow range was deemed desirable as the Wilson Plot technique(4) was to be used in the data reduction.

3.3 Data Reduction Procedures

Thermodynamic and transport properties of water were evaluated using the ASME property routines.(3) Unless otherwise noted, properties were evaluated at the film water temperatures and pressures. The shell-side conditions are denoted by the subscript s, tube-side conditions by the subscript t, inlet conditions by the subscript 1, and outlet conditions by the subscript 2.

For a particular test run, the heat duty Q was calculated from both shell- and tube-side data:

Fig. 9 - Instrumentation Stations

$$Q_S = W_S C_{pS} (T_{S2} - T_{S1})$$

$$Q_T = W_T C_{pT} (T_{t2} - T_{t1})$$

where W = mass flow rate
 C_p = specific heat

In most cases the heat duties calculated by these two methods did not vary by more than a few percent, and the two values were averaged in the data reduction.

The overall heat transfer coefficient U is defined from the equation

$$Q = UA (LMTD), \text{ where}$$

A = heat transfer area

$LMTD$ = log-mean temperature difference,

$$LMTD = \frac{(T_{S2} - T_{t1}) - (T_{S1} - T_{t2})}{\ln \left(\frac{T_{S2} - T_{t1}}{T_{S1} - T_{t2}} \right)}$$

Constant Tube-Side Flow

For the set of data in which the tube-side water flow rate was held constant and the shell-side water flow rate varied, the standard Wilson plot technique⁽⁴⁾ was used to find the tube-side heat transfer coefficient h_t at the constant flow rate and the shell-side heat transfer coefficient h_s as a function of shell-side water velocity.

The quantity $(1/U)$ is plotted versus $(1/V_s^{0.8})$, and a least-squares line is fit through the data to yield a slope m and intercept b . The tube-side heat transfer coefficient at its constant flow rate is

$$h_t = \left[b - \left(\frac{A_i}{A_o} \right) \left(\frac{t}{K} \right) \right]^{-1}, \quad \text{where}$$

t = tube wall thickness

K = tube wall thermal conductivity.

The shell-side heat transfer coefficient as a function of shell-side water velocity is

$$h_s = \frac{V_s^{0.8}}{m}$$

Since the data were taken over a range of shell side water flow rates, shell-side Nusselt number Nu_s can be evaluated as a function of shell-side Reynolds number Re_s . Heat transfer results are presented in terms of

$$Nu_s / Pr_s^{0.4} \quad \text{vs.} \quad Re_s, \quad \text{where}$$

$Nu_s = h_s D_{hs} / K_s$, shell-side Nusselt number

$Pr_s = C_p / \mu_s K_s$, shell-side Prandtl number

$Re_s = D_{hs} V_s \rho_s / \mu_s$, shell-side Reynolds number

$D_{hs} = 4A_s / P_s$, shell-side hydraulic diameter

K_s = shell-side water thermal conductivity

ρ_s = shell-side water density

μ_s = shell-side water viscosity

A_s = shell-side flow area

P_s = shell-side wetted perimeter

Constant Shell-Side Flow

For the set of data in which the shell-side water flow rate was held constant and the tube-side water flow rate varied, the Wilson plot technique (4) was also used to find the shell-side heat transfer coefficient at the constant flow rate and the tube-side heat transfer coefficient as a function of tube-side water velocity.

Again by plotting $(1/U)$ against $(1/V_t^{0.8})$ and fitting a least-squares

line through the data, a slope m and intercept b are found. The shell side heat transfer coefficient at its constant flow rate is

$$h_s = \left[b - \left(\frac{A_o}{A_i} \right) \left(\frac{t}{K} \right) \right]^{-1}$$

and the tube-side heat transfer coefficient as a function of tube side water velocity is

$$h_t = \frac{V_t^{0.8}}{m} \quad .$$

Tube-side Nusselt number Nu_t is evaluated as a function of tube-side Reynolds number Re_t and the heat transfer results are presented as

$$Nu_t / Pr_t^{0.4} \quad \text{vs.} \quad Re_t ,$$

where the definitions of the tube-side Nusselt, Prandtl, and Reynolds numbers are similar to those for shell-side flow. The geometric parameters used in the data reduction are listed in Table 1.

3.4 Seven-Tube Heat Exchanger Results

A total of 9 sets of data were taken; 5 with constant tube-side flow and 4 with constant shell-side flow. Tables 2-10 show the reduced data for these 9 sets. The Wilson Plots for each of these runs are shown in Figs. 10-18. In each of these figures the ordinate is $(1/U)$ and the abscissa is $(1/V^{0.8})$, where U is the overall heat transfer coefficient and V is the shell side water velocity in Figs. 10-14 and the tube-side velocity in Figs. 15-18. The least-square regression which fits the data is shown as the dashed line in each of the figures. The equation for this line is

$$1/U = a_1 (1/V^{0.8}) + a_0, \text{ where the constants } a_0 \text{ and } a_1 \text{ are given in Tables 2-10 as } A_1 \text{ and } A_0.$$

The correlation coefficient for the least-squares regression was quite high,

Table 1

Seven-Tube Heat Exchanger Parameters

<u>Symbol</u>	<u>Parameter</u>	<u>Value</u>
-	Tube material	Aluminum
K_t	Tube thermal conductivity (BTU/hrft ² °F)	120
t	Tube wall thickness (in.)	.065
A_t	Tube flow area (ft. ²)	.00614
P_i	Tube inside perimeter (in.)	5.36
P_o	Tube outside perimeter (in.)	5.33
D_{ht}	Tube hydraulic diameter (ft.)	.0549
L	Total tube length (ft.)	12.5
L_{HT}	Heat transfer length (ft.)	10.6
-	Number of tubes	7
D	Nominal tube outside diameter (in.)	1.25
-	Tube pitch (in.)	1.60
A_i	Inside heat transfer area (ft. ²)	33.143
A_o	Outside heat transfer area (ft. ²)	32.938
D_i	Shell inside diameter (in.)	4.75
A_s	Shell flow area (ft. ²)	.0718
$D_{H,SF}$	Shell hydraulic diameter (flow) (ft.)	.0657
$D_{H,SH}$	Shell hydraulic diameter (heat transfer) (ft.)	.0924

Table 2

DATA SET1 HEAT RATE BASED ON AVERAGE Q

RUN	FS	FT	TSI	TSO	TTI	TT0	TAUS	TAUT	FLOS	FLOT	OS	QT	ERR	DTLM	US	US
1	.0180	.1720	848.4	160.6	182.0	130.9	204.6	128.4	2.61	28.16	230.58	232.62	.2	70.0	357.	.61
2	.0370	.1720	834.9	170.5	187.4	141.3	202.7	134.3	5.40	28.12	349.77	362.59	3.5	65.1	598.	1.25
3	.0550	.1720	825.0	174.3	189.0	144.0	199.6	136.9	8.07	28.11	411.00	414.68	.9	61.1	739.	1.87
4	.0724	.1720	816.7	175.9	130.1	147.4	196.3	138.7	10.66	28.10	436.85	450.94	3.1	56.7	855.	2.47
5	.0920	.1720	811.6	175.9	127.7	146.1	193.7	138.9	13.52	28.12	485.12	479.24	-1.1	56.4	938.	3.14
6	.1100	.1720	805.5	176.2	129.9	148.5	190.9	139.2	16.27	28.10	472.39	484.85	1.3	51.6	1023.	3.76
7	.1290	.1720	800.1	174.8	128.4	147.2	187.4	137.8	19.12	28.11	489.08	490.28	.2	49.5	1022.	4.41
8	.1470	.1720	187.1	174.8	127.5	147.2	185.5	137.4	21.22	28.12	505.35	513.22	1.7	48.2	1156.	5.03
9	.1660	.1720	193.2	172.5	125.7	145.9	182.9	135.8	24.68	28.13	511.92	527.19	2.9	47.0	1207.	5.68
10	.1840	.1720	188.0	169.5	124.4	144.4	178.7	134.4	27.40	28.14	507.89	522.17	2.7	44.3	1269.	6.30

CONSTANTS FOR WILSON PLOT REGRESSION A0=.000426 A1=.00157 COR=9.990-01

RUN	HS	HT	TAUM	TFILM	NU	ST	RE	PR	NU/PR ^{1/4}	U	K	NU	CP
1	487.	2294.	139.	172.	103.	.00452	9249.	2.31	73.5	.01645	.3845	.8861	1.0012
2	764.	2956.	142.	176.	183.	.00391	20901.	2.24	132.7	.01648	.3853	.8619	1.0015
3	1052.	3651.	155.	177.	252.	.00361	31513.	2.21	183.4	.01649	.3857	.8521	1.0017
4	1313.	4613.	158.	177.	315.	.00342	41570.	2.21	229.0	.01649	.3856	.8524	1.0017
5	1592.	5498.	159.	177.	382.	.00328	52627.	2.22	277.2	.01649	.3855	.8556	1.0016
6	1839.	6448.	161.	176.	441.	.00314	62870.	2.23	319.7	.01648	.3854	.8586	1.0016
7	2090.	7377.	161.	174.	502.	.00304	72852.	2.26	361.7	.01647	.3850	.8703	1.0014
8	2321.	8446.	161.	173.	557.	.00298	82510.	2.28	400.8	.01646	.3848	.8763	1.0013
9	2559.	9424.	160.	171.	615.	.00289	92128.	2.31	439.9	.01645	.3845	.8874	1.0012
10	2720.	10483.	158.	168.	669.	.00283	100016.	2.37	474.2	.01643	.3838	.9075	1.0009

Table 3

DATA SET4 HEAT RATE BASED ON AVERAGE Q

RUN	FS	FT	TSI	TSO	TTI	TTO	TAUS	TAUT	FLOS	FLOT	QS	QT	ERR	DTLM	US	US
1	.1800	.1796	185.2	163.7	112.9	135.1	174.5	124.0	26.84	27.36	577.80	606.67	4.8	50.4	1283.	6.16
2	.1803	.1794	185.4	163.2	111.6	134.2	174.3	122.9	26.88	27.34	597.55	617.01	3.2	51.4	1291.	6.17
3	.1169	.1813	194.7	163.8	113.6	134.1	179.2	123.9	17.37	27.62	537.64	565.46	4.9	55.2	1091.	3.99
4	.0768	.1789	205.5	161.5	111.1	130.0	183.5	120.6	11.36	27.26	501.11	514.59	2.6	62.1	894.	2.62
5	.0541	.1790	221.3	163.0	111.0	129.1	192.1	120.1	7.95	27.28	465.22	493.09	5.7	70.2	746.	1.84
6	.0416	.1794	236.5	163.7	112.2	129.3	200.1	120.7	6.07	27.34	444.07	466.82	4.9	76.0	655.	1.41
7	.0319	.1797	249.4	164.7	111.6	127.1	207.0	119.4	4.63	27.38	394.17	423.88	7.0	82.9	539.	1.07
8	.0208	.1797	271.7	162.3	111.7	124.9	217.0	118.3	2.98	27.38	328.95	360.97	8.9	90.3	417.	.70
9	.0180	.1796	278.9	162.4	112.8	124.7	220.6	118.7	2.57	27.36	302.20	325.20	7.1	92.2	372.	.60

CONSTANTS FOR WILSON PLOT REGRESSION A0=.000423 A1=.00149 COR=9.996-01

RUN	HS	HT	TAUM	TFILM	NU	ST	RE	PR	NU/PR ^{1/4}	U	K	MU	CP
1	2865.	2469.	151.	163.	692.	.00298	94099.	2.47	481.8	.01640	.3826	.9449	1.0005
2	2868.	2497.	150.	162.	693.	.00297	93954.	2.48	481.9	.01640	.3824	.9480	1.0004
3	2025.	2518.	148.	164.	489.	.00325	61488.	2.45	341.6	.01641	.3828	.9372	1.0006
4	1444.	2493.	144.	164.	349.	.00353	40195.	2.46	243.4	.01641	.3828	.9395	1.0005
5	1088.	2525.	142.	167.	262.	.00379	28897.	2.39	185.0	.01643	.3835	.9164	1.0008
6	879.	2751.	140.	170.	211.	.00401	22607.	2.34	150.6	.01645	.3842	.8962	1.0011
7	709.	2381.	139.	173.	170.	.00423	17657.	2.28	122.5	.01646	.3848	.8759	1.0013
8	501.	2678.	134.	175.	120.	.00462	11598.	2.24	86.9	.01648	.3853	.8627	1.0015
9	445.	2385.	135.	178.	107.	.00476	10167.	2.20	77.8	.01649	.3858	.8491	1.0017

Table 4

DATA SET6 HEAT RATE BASED ON AVERAGE Q

RUN	FS	FT	TSI	TSO	TTI	TTO	TAUS	TAUT	FLOS	FLOT	QS	QT	ERR	DTLM	US	VS
1	.1800	.1788	260.5	233.6	178.1	204.2	247.0	191.1	25.97	26.73	708.12	699.95	-1.2	55.9	1376.	6.14
2	.1178	.1779	279.8	238.2	177.3	204.4	259.0	190.8	16.83	26.60	712.02	723.30	1.6	67.9	1155.	4.00
3	.0772	.1777	302.2	240.4	178.0	203.6	271.3	190.8	10.00	26.56	686.80	682.32	-.7	79.1	946.	2.61
4	.0543	.1757	287.3	206.7	141.3	165.0	247.0	153.1	7.73	26.58	631.35	629.65	-.3	90.9	758.	1.83

CONSTANTS FOR WILSON PLOT REGRESSION A0=.000359 A1=.00154 COR=9.989-01

RUN	HS	HT	TAUM	TFILM	NU	ST	RE	PR	NU/PR ^{1/4}	U	K	MU	CP
1	2776.	2937.	218.	233.	651.	.00294	141788.	1.56	544.9	.01686	.3940	.6085	1.0103
2	1971.	3008.	218.	238.	462.	.00321	95014.	1.51	391.2	.01691	.3944	.5903	1.0116
3	1399.	3157.	215.	243.	327.	.00350	63362.	1.48	280.2	.01695	.3947	.5752	1.0126
4	1053.	2912.	178.	212.	248.	.00373	38014.	1.75	198.4	.01671	.3919	.6814	1.0065

Table 5

DATA SETS HEAT RATE BASED ON AVERAGE Q

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RUN	FS	FT	TSI	TSO	TTI	TTO	TAUS	TAUT	FLOS	FLOT	QS	QT	ERR	DTLM	US	VS
1	.1790	.1770	158.3	145.6	116.1	129.4	151.9	122.7	26.94	26.98	341.23	358.04	4.7	29.2	1309.	6.14
2	.1170	.1800	165.0	146.9	117.3	129.0	155.9	123.1	17.57	27.41	317.55	319.44	.6	32.7	1065.	4.01
3	.0770	.1800	175.1	148.3	116.4	128.1	161.7	122.2	11.52	27.41	309.28	319.48	3.2	39.0	882.	2.63
4	.0540	.1800	185.2	148.6	115.8	126.8	168.9	121.3	8.05	27.42	294.90	302.80	2.6	44.4	736.	1.84
5	.0410	.1800	194.2	148.0	113.7	124.4	171.1	119.0	6.09	27.42	281.60	293.02	3.9	50.0	629.	1.40
6	.0320	.1800	200.7	146.3	112.5	122.3	173.5	117.4	4.74	27.43	258.36	267.59	3.5	53.0	542.	1.09
7	.0260	.1810	205.3	145.6	113.0	122.1	175.5	117.5	3.85	27.58	229.99	249.78	7.9	54.0	485.	.88
8	.0210	.1810	214.2	144.4	111.6	120.3	179.3	115.9	3.10	27.58	216.70	239.64	9.6	58.1	429.	.71
9	.0180	.1810	226.2	147.6	113.3	121.4	186.9	117.3	2.64	27.58	208.08	223.89	7.1	63.1	374.	.61

CONSTANTS FOR WILSON PLOT REGRESSION A0=.000445 A1=.00148 COR=9.992-01

RUN	HS	HT	TAUM	TFILM	NU	ST	RE	PR	NU/PRX.4	U	K	MU	CP
1	2893.	2547.	138.	145.	708.	.00300	82253.	2.87	464.7	.01632	.3774	1.0828	.9990
2	2057.	2337.	138.	147.	503.	.00327	54603.	2.81	332.4	.01633	.3781	1.0646	.9992
3	1469.	2335.	137.	150.	358.	.00356	36515.	2.76	238.9	.01634	.3788	1.0447	.9994
4	1105.	2340.	136.	151.	269.	.00383	25911.	2.71	180.5	.01635	.3793	1.0300	.9995
5	885.	2296.	133.	152.	215.	.00404	19774.	2.69	144.9	.01635	.3796	1.0226	.9996
6	725.	2279.	131.	152.	176.	.00425	15398.	2.69	118.7	.01635	.3796	1.0233	.9996
7	613.	2464.	129.	152.	149.	.00443	12503.	2.69	100.4	.01635	.3796	1.0228	.9996
8	516.	2731.	126.	153.	126.	.00463	10107.	2.68	84.6	.01635	.3797	1.0198	.9996
9	455.	2210.	129.	158.	110.	.00478	8991.	2.57	75.7	.01638	.3812	.9787	1.0001

Table 6

DATA SET12 HEAT RATE BASED ON AVERAGE Q

RUN	FS	FT	TSI	TSO	TTI	TTO	TAUS	TAUT	FLOS	FLOT	QS	QT	ERR	DTLM	US	VS
1	.1800	.1790	109.7	113.7	123.7	119.8	111.7	121.7	27.44	27.21	-108.77	-104.36	-4.2	-10.0	1161.	6.18
2	.1170	.1800	109.9	115.5	125.7	122.1	112.7	123.9	17.83	27.35	-100.80	-96.14	-4.8	-11.2	963.	4.02
3	.0771	.1820	109.1	117.3	127.4	124.1	113.2	125.7	11.75	27.64	-95.77	-91.09	-5.1	-12.4	825.	2.65
4	.0543	.1820	109.1	119.0	128.3	125.5	114.1	126.9	8.28	27.63	-82.08	-77.26	-6.2	-12.5	695.	1.87
5	.0409	.1810	109.6	120.8	129.4	127.0	115.2	128.2	6.23	27.47	-69.73	-65.84	-5.9	-12.5	590.	1.41
6	.0318	.1810	110.2	122.6	130.5	128.4	116.4	129.5	4.85	27.46	-60.02	-59.24	-1.3	-12.4	526.	1.09
7	.0262	.1810	110.4	123.2	130.6	129.0	116.8	129.8	3.99	27.46	-51.28	-45.53	-12.6	-12.2	435.	.90
8	.0210	.1810	110.0	123.6	130.9	129.5	116.8	130.2	3.20	27.46	-43.38	-39.21	-10.6	-12.4	364.	.72

CONSTANTS FOR WILSON PLOT REGRESSION A0=.000429 A1=.00171 COR=9.948-01

RUN	HS	HT	TAUM	TFILM	NU	ST	RE	PR	NU/PR ^{1/4}	U	K	MU	CP
1	2511.	2282.	117.	114.	632.	.00257	63433.	3.88	367.5	.01619	.3670	1.4266	.9986
2	1779.	2215.	119.	116.	447.	.00280	41912.	3.81	261.8	.01619	.3678	1.4032	.9986
3	1275.	2483.	121.	117.	320.	.00304	28020.	3.75	188.4	.01619	.3684	1.3831	.9986
4	963.	2667.	124.	119.	241.	.00326	20008.	3.69	143.0	.01620	.3691	1.3640	.9986
5	768.	2733.	125.	120.	192.	.00346	15277.	3.63	114.5	.01620	.3696	1.3451	.9986
6	628.	3560.	127.	122.	157.	.00364	12059.	3.57	94.2	.01621	.3701	1.3243	.9986
7	538.	2428.	127.	122.	134.	.00378	9953.	3.57	80.7	.01621	.3702	1.3220	.9986
8	450.	1994.	128.	122.	112.	.00395	7989.	3.56	67.6	.01621	.3702	1.3202	.9986

Table 7

DATA SETS HEAT RATE BASED ON AVERAGE Q

RUN	FS	FT	TSI	TSO	TTI	TTO	TAUS	TAUT	FLOS	FLOT	QS	QT	ERR	DTLM	US	US
1	.1794	.1809	194.0	170.5	117.9	140.6	182.2	129.2	26.66	27.54	627.91	624.43	-.6	53.0	1283.	10.43
2	.1791	.1134	197.0	176.8	118.7	149.7	186.9	134.2	26.58	17.26	538.51	534.50	-.8	52.5	1110.	6.55
3	.1792	.0770	199.9	181.3	113.9	155.7	190.6	134.8	26.57	11.73	495.81	489.68	-1.3	55.0	973.	4.45
4	.1785	.0519	196.1	182.5	118.0	163.5	189.3	140.7	26.50	7.99	361.59	359.99	-.7	46.8	837.	3.00
5	.1787	.0408	197.3	184.5	114.6	167.0	190.9	140.8	26.52	6.22	340.61	325.24	-4.7	47.4	763.	2.36
6	.1791	.0316	197.1	186.5	114.5	171.3	191.8	142.9	26.58	4.81	282.75	273.10	-3.5	45.0	671.	1.83
7	.1789	.0260	197.9	188.4	114.0	174.4	193.1	144.2	26.54	3.96	253.09	238.99	-5.9	44.2	605.	1.51
8	.1792	.0208	198.9	190.5	113.5	178.3	194.7	145.9	26.58	3.17	224.12	205.16	-9.2	42.8	545.	1.21
9	.1788	.0183	197.9	190.4	111.8	178.2	194.1	145.0	26.53	2.79	199.72	184.98	-8.0	42.6	491.	1.06

CONSTANTS FOR WILSON PLOT REGRESSION A0=.000555 A1=.00152 COR=9.992-01

RUN	HS	HT	TAUM	TFILM	NU	ST	RE	PR	NU/PR ^{0.4}	U	K	MU	CP
1	1942.	4285.	146.	137.	627.	.00186	109767.	3.07	400.4	.01628	.3751	1.1533	.9987
2	1882.	2952.	155.	144.	430.	.00205	72847.	2.88	281.3	.01632	.3772	1.0887	.9990
3	1870.	2168.	161.	148.	315.	.00221	50775.	2.80	208.4	.01633	.3782	1.0604	.9992
4	1882.	1582.	167.	154.	229.	.00240	35916.	2.66	154.6	.01636	.3800	1.0098	.9997
5	1949.	1306.	171.	156.	188.	.00251	28638.	2.62	128.2	.01637	.3806	.9955	.9999
6	1919.	1065.	174.	159.	153.	.00265	22656.	2.56	105.3	.01638	.3814	.9744	1.0001
7	1907.	911.	177.	161.	131.	.00275	18938.	2.51	90.6	.01639	.3820	.9590	1.0003
8	2030.	763.	181.	164.	109.	.00288	15470.	2.45	76.4	.01641	.3828	.9389	1.0005
9	1807.	688.	181.	163.	99.	.00296	13530.	2.47	68.8	.01640	.3826	.9447	1.0005

Table 8

DATA SET7 HEAT RATE BASED ON AVERAGE Q

RUN	FS	FT	TSI	TSO	TTI	TTO	TAUS	TAUT	FIOS	FLOT	QS	QT	ERR	DTLM	US	VS
1	.1812	.1805	226.5	203.0	150.6	173.6	214.7	162.1	26.57	27.23	628.72	626.53	-.4	52.6	1295.	10.41
2	.1800	.1172	226.5	205.6	148.4	179.4	216.0	163.9	26.39	17.69	552.24	548.74	-1.7	51.9	1158.	6.77
3	.1793	.0769	228.0	211.6	150.1	189.0	220.2	169.5	26.26	11.60	457.95	451.79	-1.4	49.9	990.	4.45
4	.1800	.0538	231.9	216.0	143.3	195.0	223.9	169.1	26.33	8.13	428.23	420.89	-.3	52.8	867.	3.12
5	.1801	.0410	230.9	217.3	141.0	198.8	224.1	169.9	26.36	6.20	361.53	358.86	-.7	51.1	766.	2.38
6	.1797	.0315	231.3	219.8	140.2	203.8	225.5	172.0	26.29	4.77	305.05	303.50	-.5	49.0	674.	1.83
7	.1796	.0264	230.5	220.6	141.3	206.6	225.5	173.9	26.29	3.99	262.56	261.18	-.6	46.2	616.	1.53
8	.1799	.0208	230.0	221.6	139.7	209.3	225.8	174.5	26.34	3.15	222.81	219.29	-1.7	44.6	540.	1.21
9	.1799	.0177	230.4	223.2	139.6	210.6	226.8	175.1	26.33	2.68	194.22	190.38	-.5	44.2	468.	1.03

CONSTANTS FOR WILSON PLOT REGRESSION A0=.000513 A1=.001000 COR=0.981-08

RUN	HS	HT	TAUM	TFILM	NU	ST	RE	PR	NU/PR ^{1/3}	U	K	NU	CP
1	2016.	4073.	179.	171.	582.	.00179	140388.	2.32	415.5	.01645	.3843	.8912	1.0011
2	2055.	2886.	185.	175.	411.	.00195	93648.	2.25	297.2	.01647	.3852	.8668	1.0015
3	2021.	2063.	195.	182.	293.	.00212	64556.	2.14	216.2	.01652	.3866	.8242	1.0022
4	2092.	1552.	201.	185.	220.	.00228	46047.	2.09	163.8	.01654	.3872	.8090	1.0026
5	2109.	1250.	204.	187.	177.	.00241	35574.	2.07	132.5	.01655	.3876	.7982	1.0029
6	2148.	1013.	208.	190.	143.	.00254	27915.	2.02	108.2	.01657	.3882	.7813	1.0033
7	2188.	880.	211.	192.	124.	.00263	23713.	1.99	94.4	.01658	.3886	.7708	1.0036
8	2242.	728.	213.	194.	103.	.00276	18866.	1.97	78.3	.01659	.3889	.7631	1.0037
9	1847.	640.	213.	194.	90.	.00285	16100.	1.96	68.9	.01659	.3890	.7610	1.0038

Table 9

DATA SET9 HEAT RATE BASED ON AVERAGE 0

RUN	FS	FT	TSI	TSO	TTI	TTO	TAUS	TAUT	FLOS	FLOT	QS	QT	ERR	DTLM	US	VS
1	.1800	.1800	168.9	156.2	127.3	139.7	162.5	133.5	27.00	27.34	343.85	338.52	-1.6	29.0	1277.	10.36
2	.1800	.1170	170.8	157.5	119.4	138.9	164.1	129.2	26.98	17.81	360.65	346.81	-4.0	34.9	1102.	6.74
3	.1800	.0770	176.6	162.4	111.8	142.7	169.5	127.2	26.92	11.73	381.07	361.38	-5.4	41.7	967.	4.44
4	.1800	.0540	185.3	170.8	105.0	150.1	178.0	127.5	26.83	8.24	390.59	370.94	-5.3	48.9	845.	3.12
5	.1800	.0400	195.4	181.2	102.1	161.1	188.3	131.6	26.73	6.10	379.97	359.59	-5.7	53.6	749.	2.31
6	.1800	.0315	201.8	188.1	97.8	168.6	194.9	133.2	26.67	4.81	367.56	339.86	-8.1	57.0	674.	1.82
7	.1790	.0258	207.5	194.5	94.1	175.3	201.0	134.7	26.46	3.94	345.58	319.66	-8.1	59.9	603.	1.49
8	.1794	.0207	217.1	204.8	92.9	185.9	210.9	139.4	26.42	3.16	327.78	293.68	-11.6	63.1	535.	1.20
9	.1793	.0179	229.8	218.1	95.6	198.2	224.0	146.9	26.25	2.73	309.79	280.29	-10.5	67.1	478.	1.04

CONSTANTS FOR WILSON PLOT REGRESSION A0=.000553 A1=.00155 COR=9.988-01

RUN	HS	HT	TAUM	TFILM	NU	ST	RE	PR	NU/PR**1.4	U	K	NU	CP
1	1945.	4195.	143.	138.	614.	.00183	109607.	3.05	392.6	.01629	.3753	1.1475	.9987
2	1850.	2974.	143.	136.	436.	.00200	70028.	3.12	276.6	.01628	.3746	1.1694	.9987
3	1875.	2129.	147.	137.	312.	.00217	46590.	3.08	198.7	.01628	.3750	1.1570	.9987
4	1891.	1604.	155.	141.	234.	.00233	33795.	2.97	151.5	.01630	.3762	1.1184	.9987
5	1954.	1263.	166.	149.	183.	.00248	26621.	2.77	121.8	.01634	.3786	1.0510	.9993
6	2014.	1044.	174.	153.	151.	.00260	21742.	2.67	102.0	.01636	.3799	1.0130	.9997
7	1977.	891.	180.	158.	128.	.00271	18359.	2.58	87.9	.01638	.3811	.9824	1.0000
8	1989.	748.	191.	165.	107.	.00284	15599.	2.42	75.2	.01642	.3832	.9268	1.0007
9	1777.	667.	203.	175.	95.	.00293	14416.	2.25	68.7	.01648	.3852	.8657	1.0015

Table 10

DATA SET11 HEAT RATE BASED ON AVERAGE Q

RUN	FS	FT	TSI	TSO	TTI	TTO	TAUS	TAUT	FLOS	FLOT	QS	QT	ERR	DTLM	US	VS
1	.1800	.1790	104.3	109.4	121.4	116.4	106.8	118.9	27.46	27.23	-138.18	-134.34	-2.9	-12.0	1228.	10.28
2	.1790	.1160	100.3	104.0	115.1	109.6	102.2	112.3	27.32	17.67	-100.93	-96.34	-4.8	-10.1	1058.	6.66
3	.1800	.0779	100.1	103.0	114.0	107.6	101.5	110.8	27.47	11.87	-81.20	-76.20	-6.6	-9.2	934.	4.48
4	.1800	.0549	101.1	103.4	114.6	107.4	102.2	111.0	27.47	8.36	-61.44	-60.13	-2.2	-8.5	776.	3.15
5	.1800	.0412	102.6	104.5	115.9	108.0	103.5	111.9	27.46	6.27	-51.28	-49.50	-3.6	-8.0	683.	2.37
6	.1800	.0263	104.9	106.5	118.9	109.4	105.7	114.1	27.45	4.00	-42.22	-38.22	-10.5	-7.8	562.	1.51
7	.1810	.0179	109.0	110.0	121.9	112.2	109.5	117.1	27.59	2.72	-27.55	-26.29	-4.8	-6.7	438.	1.03

CONSTANTS FOR WILSON PLOT REGRESSION A0=.000557 A1=.00176 COR=9.988-01

RUN	HS	HT	TAUM	TFILM	MU	ST	RE	PR	MU/PR ² .4	U	K	MU	CP
1	1957.	3673.	115.	117.	548.	.00161	90289.	3.77	322.1	.01619	.3682	1.3899	.9986
2	1891.	2597.	108.	110.	390.	.00175	54910.	4.05	223.0	.01618	.3652	1.4826	.9986
3	1959.	1888.	106.	108.	284.	.00190	36249.	4.13	161.2	.01617	.3644	1.5087	.9986
4	1795.	1427.	106.	109.	215.	.00204	25572.	4.13	121.9	.01617	.3645	1.5071	.9986
5	1816.	1134.	107.	109.	171.	.00216	19339.	4.09	97.1	.01618	.3648	1.4954	.9986
6	2060.	792.	108.	111.	119.	.00236	12559.	4.01	68.2	.01618	.3656	1.4693	.9986
7	1867.	582.	111.	114.	87.	.00255	8801.	3.88	50.6	.01619	.3670	1.4257	.9986

Figure 10

DATA SET 1 AVERAGE 0

Figure 11

Figure 12

DATA SET 6 AVERAGE 0

Figure 13

Figure 14

DATA SET 12 AVERAGE 0

Figure 15

Figure 16

DATA SET 7 AVERAGE 0

Figure 17

Figure 18

DATA SET 11 AVERAGE 0

0.999 in most of the sets. It is also shown in Tables 2-10.

The heat transfer results for the 9 sets of data are shown in Figs. 19-27 in terms of

$$\text{Nu}/\text{Pr}^{0.4} \text{ vs. } \text{Re} .$$

The Dittus-Boelter correlation for a plain tube is shown for comparison as the solid line while the dashed line represents the least-squares fit of the data. The Dittus-Boelter correlation is

$$\text{Nu}/\text{Pr}^{0.4} = 0.022 \text{ Re}^{0.8}$$

The correlations for the spirally fluted tube are given in each of the figures for an exponent of 0.8 for ease of comparison with the Dittus Boelter correlation. The ratio of the coefficient is the enhancement.

Examination of these results indicate that there is greater enhancement on the shell-side than the tube-side. However no conclusions could be drawn for the effect of heat flux. The data for the reversal of heat flux (data sets 11 and 12) indicated essentially the same enhancement. Data sets 6 and 7, shown in Tables 4 and 8, resulted from a test on the same day appeared to deviate from the bulk of the data. Figures 28 and 29 show the entire data set for the shell-side and tube-side performance. The scatter is a result of the 2 data sets mentioned above. The correlation coefficients for the Wilson plots used in the data reduction were 0.992. While this scatter could be considered acceptable, we examined the data with 4 sets deleted, (sets 6 and 7 and the 2 sets for the flux reversed tests), in Figs. 30 and 31 for the Wilson Plot, and in Figs. 32 and 33 for the Nusselt Modulus vs. Reynolds Number correlation. These indicate very little scatter with correlation coefficients for the Wilson Plots is 0.999.

It appears from this data there was no dependency of enhancement on heat flux in the range of this variable. These heat fluxes were considerably less than previously tested. Tables 11 and 12 is a compilation of all of the data with these 4 data sets removed. The enhancement factor for the spirally fluted tube as given by these data for single phase liquid flow is 1.59 for

Figure 19

NU/PR^{0.4} = .04661 RE^{.800}
 DATA SET 1 AVERAGE 0

Figure 20

Figure 21

Figure 22

NU/PR^{0.4} = .053213 RE^{0.800}
 DATA SET 8 AVERAGE 0

Figure 23

NU/PR**0.4 = .05177 RE** .800
 DATA SET 12 AVERAGE 0

Figure 24

NU/PR^{0.4} = 0.035113 RE^{0.800}
 DATA SET 5 AVERAGE 0

Figure 25

Figure 26

Figure 27

NU/PR^{0.4} = 0.03585 RE^{0.8}
 DATA SET 11 AVERAGE 0

Figure 28

Figure 29

Figure 30

Figure 31

Figure 32

Figure 33

NU/PR^{0.4} = 0.035053 RE^{0.8}
 DATA SET 99 AVERAGE 0

Table 11

DATA SET99 HEAT RATE BASED ON AVERAGE Q

RUN	FS	FT	TSI	TSO	TTI	TTO	TAUS	TAUT	FLOS	FLOT	QS	QT	ERR	DTLM	US	VS
1	.1794	.1809	194.0	170.5	117.9	140.6	182.2	129.2	26.66	27.54	627.91	624.43	-.6	53.0	1283.	10.43
2	.1791	.1134	197.0	176.8	118.7	149.7	186.9	134.2	26.58	17.26	538.51	534.50	-.8	52.5	1110.	6.55
3	.1792	.0770	199.9	181.3	113.9	155.7	190.6	134.8	26.57	11.73	495.81	489.68	-1.3	55.0	973.	4.45
4	.1785	.0519	196.1	182.5	118.0	163.5	189.3	140.7	26.50	7.90	361.59	359.09	-.7	46.8	837.	3.00
5	.1787	.0408	197.3	184.5	114.6	167.0	190.9	140.8	26.52	6.22	340.61	325.24	-4.7	47.4	783.	2.36
6	.1791	.0316	197.1	186.5	114.5	171.3	191.8	142.9	26.58	4.81	282.75	273.10	-3.5	45.0	671.	1.83
7	.1789	.0260	197.9	188.4	114.0	174.4	193.1	144.2	26.54	3.96	253.00	238.89	-5.0	44.2	605.	1.51
8	.1792	.0208	198.9	190.5	113.5	178.3	194.7	145.9	26.58	3.17	224.12	205.16	-8.2	42.8	545.	1.21
9	.1788	.0183	197.9	190.4	111.8	178.2	194.1	145.0	26.53	2.79	199.72	184.98	-8.0	42.6	491.	1.06
10	.1800	.1800	168.9	156.2	127.3	139.7	162.5	133.5	27.00	27.34	343.85	338.52	-1.6	29.0	1277.	10.36
11	.1800	.1170	170.8	157.5	119.4	138.9	164.1	129.2	26.98	17.81	360.65	346.81	-4.0	34.9	1102.	6.74
12	.1800	.0770	176.6	162.4	111.8	142.7	169.5	127.2	26.92	11.73	381.07	361.38	-5.4	41.7	967.	4.44
13	.1800	.0540	185.3	170.8	105.0	150.1	178.0	127.5	26.83	8.24	390.59	370.94	-5.3	48.9	845.	3.12
14	.1800	.0400	195.4	181.2	102.1	161.1	188.3	131.6	26.73	6.10	379.97	359.59	-5.7	53.6	749.	2.31
15	.1800	.0315	201.8	188.1	97.8	168.6	194.9	133.2	26.67	4.81	367.56	339.86	-8.1	57.0	674.	1.82
16	.1790	.0258	207.5	194.5	94.1	175.3	201.0	134.7	26.46	3.94	345.58	319.66	-8.1	59.9	603.	1.49
17	.1794	.0207	217.1	204.8	92.9	185.9	210.9	139.4	26.42	3.16	327.78	293.68	-11.6	63.1	535.	1.20
18	.1793	.0179	229.8	218.1	95.6	198.2	224.0	146.9	26.25	2.73	309.79	280.29	-10.5	67.1	478.	1.04

CONSTANTS FOR WILSON PLOT REGRESSION A0=.000554 A1=.1536-02 COR=9.989-01

RUN	HS	HT	TAUM	TFILM	NU	ST	RE	PR	NU/PR ^{1/4}	U	K	NU	CP
1	1949.	4250.	146.	138.	622.	.00185	109829.	3.07	397.2	.01628	.3751	1.1527	.9987
2	1892.	2928.	155.	145.	426.	.00203	72896.	2.88	279.0	.01632	.3773	1.0879	.9990
3	1883.	2149.	161.	148.	312.	.00219	50818.	2.80	206.7	.01633	.3783	1.0595	.9992
4	1901.	1569.	167.	154.	227.	.00238	35945.	2.65	153.4	.01636	.3801	1.0090	.9998
5	1973.	1295.	171.	156.	187.	.00249	28665.	2.61	127.2	.01637	.3806	.9946	.9999
6	1949.	1056.	175.	159.	152.	.00263	22678.	2.55	104.5	.01638	.3814	.9735	1.0001
7	1941.	904.	178.	161.	130.	.00273	18956.	2.51	89.9	.01639	.3820	.9580	1.0003
8	2076.	756.	182.	164.	108.	.00286	15486.	2.45	75.8	.01641	.3828	.9379	1.0006
9	1848.	683.	181.	163.	98.	.00293	13544.	2.47	68.3	.01640	.3826	.9437	1.0005
10	1938.	4228.	143.	138.	619.	.00185	109575.	3.05	395.7	.01629	.3753	1.1478	.9987
11	1841.	2998.	142.	136.	439.	.00201	69997.	3.12	278.8	.01628	.3746	1.1699	.9987
12	1862.	2146.	147.	137.	314.	.00219	46560.	3.08	200.3	.01628	.3750	1.1577	.9987
13	1874.	1617.	155.	141.	236.	.00235	33765.	2.97	152.6	.01630	.3762	1.1194	.9987
14	1930.	1273.	166.	149.	185.	.00250	26593.	2.78	122.7	.01634	.3785	1.0521	.9993
15	1984.	1053.	173.	153.	152.	.00262	21716.	2.67	102.7	.01636	.3799	1.0143	.9997
16	1943.	898.	180.	157.	129.	.00273	18335.	2.58	88.5	.01638	.3810	.9837	1.0000
17	1949.	754.	191.	155.	108.	.00286	15577.	2.42	75.8	.01642	.3831	.9682	1.0007
18	1741.	672.	202.	175.	96.	.00295	14395.	2.25	69.2	.01647	.3851	.8670	1.0015

Table 12

DATA SET98 HEAT RATE BASED ON AVERAGE Q

RUN	FS	FT	TSI	TSO	TTI	TTO	TAUS	TAUT	FLOS	FLOT	QS	QT	ERR	DTLM	US	US
1	.0180	.1720	248.4	160.6	122.0	130.9	204.5	126.4	2.61	26.16	230.56	232.52	.8	70.9	357.	.61
2	.0370	.1720	234.9	170.5	127.4	141.3	202.7	134.3	5.40	26.12	340.77	348.59	3.5	65.1	500.	1.25
3	.0550	.1720	225.0	174.3	129.0	144.9	199.6	136.9	8.07	26.11	411.00	414.58	.9	61.1	730.	1.87
4	.0724	.1720	216.7	175.9	130.1	147.4	195.3	138.7	10.66	26.10	436.85	450.94	3.1	56.7	855.	2.47
5	.0920	.1720	211.5	175.9	127.7	146.1	193.7	136.9	13.58	26.12	485.18	479.94	-1.1	56.4	936.	3.14
6	.1100	.1720	206.5	176.2	129.9	148.5	190.9	139.2	16.27	26.10	478.39	484.85	1.3	51.5	1023.	3.76
7	.1290	.1720	200.1	174.6	128.4	147.2	187.4	137.8	19.12	26.11	489.68	490.28	.2	49.5	1082.	4.41
8	.1470	.1720	197.1	174.0	127.5	147.2	185.5	137.4	21.82	26.12	505.35	513.82	1.7	48.2	1156.	5.03
9	.1660	.1720	193.2	172.5	125.7	145.9	182.9	135.8	24.68	26.13	511.88	527.19	2.9	47.0	1207.	5.68
10	.1840	.1720	188.0	169.5	124.4	144.4	178.7	134.4	27.40	26.14	507.89	522.17	2.7	44.3	1269.	6.30
11	.1800	.1796	185.2	163.7	112.9	135.1	174.5	124.0	26.84	27.36	577.80	606.67	4.8	50.4	1283.	6.16
12	.1803	.1794	185.4	163.2	111.6	134.2	174.3	122.9	26.88	27.34	597.55	617.01	3.2	51.4	1291.	6.17
13	.1169	.1813	194.7	163.8	113.6	134.1	179.2	123.9	17.37	27.62	537.64	565.46	4.9	55.2	1091.	3.00
14	.0768	.1789	205.5	161.5	111.1	130.0	183.5	120.6	11.36	27.25	501.11	514.59	2.6	62.1	894.	2.62
15	.0541	.1780	221.3	163.0	111.0	129.1	180.1	120.1	7.95	27.28	465.22	493.09	5.7	70.2	746.	1.84
16	.0416	.1794	236.5	163.7	112.2	129.3	200.1	120.7	6.07	27.34	444.07	466.82	4.9	76.0	655.	1.41
17	.0319	.1797	248.4	164.7	111.6	127.1	207.0	119.4	4.63	27.38	394.17	423.88	7.0	82.9	530.	1.07
18	.0208	.1797	271.7	162.3	111.7	124.9	217.0	118.3	2.98	27.38	328.95	360.97	8.9	90.3	417.	.70
19	.0180	.1796	278.9	162.4	112.8	124.7	220.6	118.7	2.67	27.36	302.20	325.20	7.1	92.2	372.	.60
20	.1790	.1770	158.3	145.6	116.1	129.4	151.9	122.7	26.94	26.96	341.23	358.04	4.7	29.2	1399.	6.14
21	.1170	.1800	165.0	146.9	117.3	129.0	155.9	123.1	17.57	27.41	317.55	319.44	.6	32.7	1065.	4.01
22	.0770	.1800	176.1	148.3	116.4	128.1	161.7	122.2	11.52	27.41	309.28	319.48	3.2	39.0	822.	2.63
23	.0540	.1800	185.2	148.6	115.8	126.8	166.9	121.3	8.85	27.42	294.00	302.80	2.6	44.4	736.	1.84
24	.0410	.1800	194.2	148.0	113.7	124.4	171.1	119.0	6.09	27.42	281.60	293.02	3.9	50.0	689.	1.40
25	.0320	.1800	200.7	146.3	112.5	122.3	173.5	117.4	4.74	27.43	258.36	267.59	3.5	53.0	548.	1.09
26	.0260	.1810	205.3	145.6	113.0	122.1	175.5	117.5	3.85	27.58	229.99	249.78	7.9	54.0	485.	.88
27	.0210	.1810	214.2	144.4	111.6	120.3	179.3	115.9	3.10	27.58	216.70	239.64	9.6	58.1	429.	.71
28	.0180	.1810	226.2	147.6	113.3	121.4	186.9	117.3	2.64	27.58	200.08	223.89	7.1	63.1	374.	.61

CONSTANTS FOR WILSON PLOT REGRESSION A0=.000437 A1=.1498-02 COR=9.986-01

RUN	HS	HT	TAHM	TFILM	MU	ST	RE	PR	MU/PR ^{0.4}	U	K	MU	CP
1	447.	1847.	142.	173.	107.	.00473	9947.	2.29	77.2	.01646	.3848	.8769	1.0013
2	800.	2517.	151.	177.	102.	.00410	21069.	2.22	139.3	.01649	.3856	.8547	1.0016
3	1101.	2380.	157.	178.	264.	.00378	31721.	2.20	192.5	.01650	.3859	.8462	1.0017
4	1375.	2398.	160.	178.	329.	.00358	41803.	2.20	240.2	.01649	.3858	.8474	1.0017
5	1667.	2254.	161.	177.	399.	.00341	52953.	2.21	290.7	.01649	.3857	.8511	1.0017
6	1925.	2309.	163.	177.	461.	.00329	63144.	2.22	335.3	.01649	.3856	.8547	1.0016
7	2188.	2261.	162.	175.	525.	.00318	73139.	2.25	379.2	.01647	.3852	.8667	1.0015
8	2430.	2135.	162.	174.	583.	.00310	82826.	2.27	420.1	.01647	.3849	.8729	1.0014
9	2679.	2125.	161.	172.	644.	.00302	92448.	2.30	461.2	.01646	.3846	.8842	1.0012
10	2910.	2187.	159.	169.	700.	.00295	100339.	2.36	497.0	.01644	.3839	.8944	1.0010
11	2858.	2474.	151.	163.	690.	.00297	94082.	2.47	480.7	.01649	.3825	.9451	1.0006
12	2862.	2502.	150.	162.	691.	.00297	93937.	2.48	480.7	.01649	.3824	.9482	1.0004
13	2021.	2525.	148.	164.	489.	.00324	61473.	2.45	340.8	.01641	.3828	.9374	1.0005
14	1441.	2503.	143.	163.	348.	.00352	40182.	2.46	242.8	.01641	.3829	.9398	1.0005
15	1086.	2538.	142.	167.	262.	.00370	28886.	2.39	184.6	.01643	.3835	.9167	1.0008
16	877.	2771.	140.	170.	211.	.00400	22597.	2.34	150.3	.01644	.3842	.8986	1.0011
17	707.	2400.	139.	173.	170.	.00422	17648.	2.28	122.2	.01646	.3848	.8764	1.0013
18	500.	2711.	134.	175.	120.	.00451	11591.	2.24	86.7	.01648	.3853	.8632	1.0015
19	444.	2415.	135.	178.	106.	.00475	10161.	2.21	77.6	.01649	.3857	.8497	1.0017
20	2050.	2581.	138.	145.	698.	.00296	82188.	2.87	457.8	.01632	.3774	1.0037	.9990
21	2027.	2377.	138.	147.	495.	.00322	54548.	2.82	327.4	.01633	.3780	1.0057	.9992
22	1448.	2392.	137.	149.	353.	.00351	36464.	2.76	235.3	.01635	.3792	1.0318	.9995
23	1083.	2416.	135.	151.	265.	.00377	25866.	2.72	177.7	.01635	.3795	1.0240	.9996
24	872.	2388.	133.	152.	212.	.00399	19732.	2.70	142.7	.01635	.3795	1.0258	.9996
25	714.	2391.	130.	152.	174.	.00419	15362.	2.70	116.8	.01635	.3795	1.0256	.9996
26	604.	2621.	128.	152.	147.	.00437	12471.	2.69	98.9	.01635	.3796	1.0229	.9996
27	509.	2966.	125.	152.	124.	.00456	10077.	2.69	83.3	.01638	.3811	.9819	1.0000
28	449.	2383.	128.	158.	109.	.00471	8963.	2.58	74.5	.01638	.3811	.9819	1.0000

tube-side and 2.28 for shell-side. This is in addition to the area extension factor of 1.6. Therefore one would expect in comparing for equal tube diameters and lengths the spiral fluted tube with a plain tube a factor for increased heat transfer of 2.54 and 3.65 on the tube-side and shell-side respectively.

4. DETERMINATION OF THE FRICTIONAL COEFFICIENT OF SPIRAL FLUTED TUBING

Previous measurements of the pressure losses in spiral fluted tubing have indicated that the friction coefficient is less than that of smooth tubing. This result has been very surprising as an increase would have been anticipated on the basis of Reynolds analogy between heat and momentum transfer. In fact, an equivalent friction coefficient would indicate a substantial performance improvement over existing heat transfer enhancement methods and predictions based on this analogy. These previous measurements were made in conjunction with the heat transfer tests. This required corrections for the transition into and out of the tube under test as the difficulties of obtaining reliable static pressure measurements on an undulating surface, such as the spiral fluted tube, prevented direct measurement of the pressure drop within the tube. Therefore the scope of this program was increased to determine the friction coefficient on the inside of a spiral fluted tube in single phase flow without introducing further uncertainty in the measurement. The following method was developed to accomplish this objective.

4.1 Experimental Systems

4.1.1 Water Test

A diagram of the frictional pressure drop water test experimental system is shown in Fig. 34. The system operated as an open loop, using water obtained directly from a high-pressure city water main. Flow rates were controlled by a standard gate valve and flow rate measurement was accomplished using a calibrated turbine flow meter connected to a digital frequency counter.

Water gauge pressure was measured just upstream of entrance to the spirally fluted tube test section by a water/gas pressure sensing system. The static pressure tap upstream of the test section entrance led into the bottom of a vertical cylinder. A high pressure nitrogen gas supply was connected into the top of the cylinder. At each water flow rate, a

Figure 34 - Water Presssure Drop Test System

sufficient amount of gas was let into the cylinder so that the water level in the cylinder and the water level in the test tube coincided; at such a condition, the gas pressure and water pressure are equal. Gas pressures were then measured using a set of two calibrated dial pressure gauges capable of measurements from about 0.05 to 100 psig.

After flowing through the spirally fluted test tube, the water discharged freely to atmospheric pressure. Water temperature was measured by a dial-type thermometer downstream of the flow meter.

Two spirally fluted tube sections were tested, a 15-foot long tube was cut into a 5-foot long tube and 10-foot long tube. Relevant parameters are provided in Table 13. Data were taken from each test section with water flow rates from about 10 GPM to 90 GPM. The test Reynolds numbers ranged from about 20,000 to 160,000.

4.1.2 Air Test

The experimental system used in the frictional pressure drop air test is shown in Fig. 35. The system operated as an open loop, with ambient air being induced into the test apparatus and subsequently exhausted to atmosphere by two parallel compressors. Air flow entered the system through a calibrated Merriam laminar flow element, which consists of pressure taps on either side of a honeycomb grid. The volumetric flow rate through the element is a linear function of the pressure drop across the grid; this pressure drop was measured by an inclined water manometer.

Each spirally fluted test tube was installed between two 2-inch diameter plain tube sections, each of which contained a static pressure tap approximately two inches from the test tube connections. The differential pressure across the test tube was measured between these two pressure taps. The differential pressure between the downstream tap and ambient was also measured in order to fix the absolute air pressure at the downstream tap. Each differential pressure measurement was made with one of three parallel manometers, depending on the pressure range. Two-inch inclined water manometers, 60-inch vertical water manometers, and 14-inch vertical mercury

Table 13

Spirally Fluted Tube Test Section Parameters

<u>Parameter</u>	<u>Test Section 1</u>	<u>Test Section 2</u>
Material	Aluminum Alloy 6061	Aluminum Alloy 6061
Spiral Angle	30°	30°
Length (in.)	60.000	119.375
Nominal O.D. (in.)	1.25	1.25
Wall thickness (in.)	0.065	0.065
Flow area (ft ²)	0.00614	0.00614
Perimeter (in.)	5.28	5.28
Hydraulic Diameter (in.)	0.6705	0.6705

Figure 35 - Air Pressure Drop Test System

manometers were used for low, medium, and high range differential pressure measurements respectively.

Downstream of the test section, the system line was split into two lines, one to each compressor. Flow rates were controlled by the two line flow control valves and by the bypass control valve on compressor 2.

Barometric pressure was taken before and after each set of test runs and ambient temperature was measured during each test run. The test was assumed to be isothermal from system entrance to compressor inlet.

The same two spirally fluted tube sections were tested, the 5-foot long tube and the 10-foot long tube. Relevant parameters are listed in Table 13. System flow rates were limited by the maximum pressure rise of the parallel compressors. The 5-foot long tube tests reached a maximum flow rate of 114 ACFM, while the 10-foot long tube tests reached 93 ACFM.

4.2 Data Reduction

The Moody friction factor is conventionally defined from the equation

$$\Delta P = f \frac{L \rho V^2}{D_h 2g} \quad , \quad \text{where}$$

P = pressure drop across test section

f = Moody friction factor

L = test section length

D_h = tube hydraulic diameter

ρ = fluid density

V = fluid velocity

g = gravitational constant

The hydraulic diameter

$$D_h = \frac{4A}{P} \quad , \quad \text{where}$$

A = tube cross-section flow area

P = tube wetted perimeter

Friction factor is normally taken to be a function of Reynolds number, which is defined as

$$Re = \frac{D_h V \rho}{\mu}, \text{ where}$$

Re = Reynolds number

μ = fluid absolute viscosity

The friction factor for a particular tube is a function only of Reynolds number, $f=f(Re)$, the friction factor can be found using a novel technique. Denoting the data obtained from the 5-foot long test section by subscript 1 and the data obtained from the 10-foot long test section by subscript 2, at constant Reynolds number

$$f \frac{L_1}{D_h} = \left[\frac{\frac{\Delta P}{\rho V^2}}{\frac{2g}{2g}} \right]_1 \quad \text{and} \quad f \frac{L_2}{D_h} = \left[\frac{\frac{\Delta P}{\rho V^2}}{\frac{2g}{2g}} \right]_2$$

Subtracting the second equation from the first yields

$$f \frac{(L_2 - L_1)}{D_h} = \left[\frac{\frac{\Delta P}{\rho V^2}}{\frac{2g}{2g}} \right]_2 - \left[\frac{\frac{\Delta P}{\rho V^2}}{\frac{2g}{2g}} \right]_1,$$

$$f = \left[\frac{D_h}{(L_2 - L_1)} \right] \left[\left(\frac{\frac{\Delta P}{\rho V^2}}{\frac{2g}{2g}} \right)_2 - \left(\frac{\frac{\Delta P}{\rho V^2}}{\frac{2g}{2g}} \right)_1 \right].$$

The advantage of using this subtraction method of data reduction is that all test section entrance and exit pressure losses at constant Reynolds number cancel, leaving only frictional pressure drop under consideration. This also assumes that the shorter tube was sufficiently long for the flow to be fully developed. Its length to diameter ratio was greater than 90 which is sufficient.

4.2.1 Water Test

Water was assumed incompressible in the analysis, so the data reduction was straightforward with all water properties evaluated at the measured temperature. The reduced data from the water tests is given in Tables 14 and 15.

4.2.2 Air Test

The compressibility of air caused the reduction of the air test data to be somewhat more involved. Specifically, the average air pressure in the test section required evaluation in order to find the average air density, velocity, and viscosity in the test section. The static pressures at the two pressure taps in the 2-inch diameter plain tube sections could not simply be averaged; the static pressure loss at test tube entrance and the static pressure rise at test tube exit (from Ref. 1) were first applied to the static pressure readings, and these modified pressures averaged to find the mean test tube air pressure. Air properties were then evaluated at this mean pressure and the measured temperature. The reduced data from the air tests is given in Tables 16 and 17.

4.3 Results

The Moody friction factor for a completely turbulent smooth tube results for both water and air flow in a spirally fluted tube are plotted as a function of Reynolds number in Fig. 36. Shown for comparison is the friction factor curve for a completely smooth plain tube.

The pressure measurements made during the 7-tube heat exchanger tests

Table 14

START=012771, PROG SIZE(I/D)=5239/2263
 SYSRLIBS. LEVEL 74R1GA

END MAP. ERRORS: 0 TIME: 5.917 STORAGE: 24408/0/035777/0202777/027377/0163174

5 FT TUBE, WATER

CASE	TEMP	GPM	RHO	VEL	DP	RATIO	RE
1	62.0	11.63	62.333	4.222	.475	3.9615	19911.0
2	62.0	18.18	62.333	6.599	.995	3.3965	31122.1
3	62.0	25.00	62.333	9.071	1.695	3.0629	42781.7
4	62.0	30.76	62.333	11.163	2.400	2.8629	52647.6
5	62.0	37.05	62.333	13.445	3.365	2.7671	63410.2
6	62.5	42.29	62.329	15.346	4.170	2.6320	72897.3
7	62.5	49.67	62.329	17.395	5.000	2.7258	88389.7
8	62.5	54.43	62.329	19.484	5.995	2.8521	99347.6
9	62.5	57.36	62.329	21.627	6.990	2.9869	11152.9
10	62.5	59.59	62.329	23.835	7.995	3.1602	12416.4
11	62.5	61.04	62.329	26.100	8.990	3.4433	13825.0
12	62.5	62.13	62.329	28.425	9.990	3.8092	15468.7
13	62.5	63.48	62.329	30.800	10.995	4.2732	17123.8
14	62.5	64.87	62.329	33.225	12.000	4.8468	18876.8
15	62.5	66.73	62.329	35.700	13.005	5.4408	20812.4
16	62.5	68.64	62.329	38.225	14.010	6.0653	22948.7
17	62.5	70.45	62.329	40.800	15.015	6.7308	25285.8
18	62.5	72.88	62.329	43.425	16.020	7.4375	27825.4
19	63.5	82.64	62.323	49.989	18.254	8.3562	31449.9
20	63.5	74.78	62.323	27.136	11.885	2.3993	130750.0
21	63.5	67.71	62.323	24.569	9.762	2.4042	118380.5
22	63.5	59.85	62.323	21.717	7.739	2.4394	104636.7
23	63.5	50.94	62.323	18.484	6.616	2.4438	89060.3
24	63.5	43.86	62.323	15.917	4.333	2.5426	76690.8

EXIT LOC=013166
 DATA IGNORED - IN CONTROL MODE

WAIT-LAST INPUT IGNORED*
 START=012771, PROG SIZE(I/D)=5239/2268
 SYS\$RLIBS. LEVEL 74R1GA
 END MAP. ERRORS: 0 TIME: 5.302 STORAGE: 24408/0/035777/0202777/027377/0163174

10 FT TUBE, WATER

CASE	TEMP	GPM	RHO	VEL	DP	RATIO	RE
1	64.0	30.24	62.320	10.972	3.800	4.6924	53244.5
2	64.0	19.23	62.320	6.979	1.800	5.4942	33866.1
3	64.0	10.06	62.320	3.651	.575	6.4125	17717.4
4	64.0	14.25	62.320	5.172	1.080	6.0014	25099.7
5	64.0	23.95	62.320	8.690	2.560	5.0393	42171.1
6	64.0	27.88	62.320	10.117	3.320	4.8226	40092.0
7	64.0	3.77	62.320	1.369	.147	11.6578	6644.0
8	64.0	21.33	62.320	7.740	2.125	5.2739	37557.2
9	64.5	32.33	62.317	11.733	4.250	4.5899	57340.3
10	64.5	39.15	62.317	14.205	5.814	4.2835	69421.7
11	64.5	45.44	62.317	16.487	7.443	4.0707	80573.8
12	64.5	51.46	62.317	18.674	9.466	4.0359	91261.2
13	65.0	59.32	62.313	21.527	12.181	3.9084	105947.4
14	65.0	65.61	62.313	23.809	14.748	3.8683	117178.6
15	64.0	70.59	62.320	25.615	16.574	3.7554	124298.8
16	64.0	76.88	62.320	27.897	19.684	3.7602	135372.2
17	64.0	81.33	62.320	29.513	21.954	3.7471	143215.8
18	64.0	85.00	62.320	30.845	23.731	3.7083	149675.3
19	64.0	87.36	62.320	31.700	24.619	3.6422	153827.9
20	64.0	83.43	62.320	30.274	22.744	3.6893	146907.0
21	64.0	79.76	62.320	28.943	20.967	3.7211	140447.5
22	64.0	73.74	62.320	26.756	18.104	3.7597	129835.5
23	64.0	71.64	62.320	25.995	17.216	3.7875	126144.4
24	64.0	68.23	62.320	24.759	15.686	3.8040	120146.3
25	64.0	62.99	62.320	22.858	13.514	3.8453	110918.5
26	64.0	60.11	62.320	21.812	12.527	3.9144	105843.2
27	64.0	56.87	62.320	20.671	11.293	3.9291	100306.5
28	64.0	54.35	62.320	19.720	10.453	3.9964	95692.5
29	64.0	48.58	62.320	17.620	8.528	4.0801	85541.9
30	64.0	42.03	62.320	15.251	6.653	4.2523	74007.1
31	63.5	35.22	62.323	12.779	4.827	4.3938	61572.5

EXIT LOC11013166

Table 16

WAIT-LAST INPUT IGNORED
 START-013306, PROG SIZE(I/D)-5651/2513
 SYS\$RLIBS. LEVEL 74R1GA
 END MAP. ERRORS: 0 TIME: 5.418 STORAGE: 24408/0/035777/0202777/027377/0163300

10 FOOT TUBE, AIR

CASE	TEMP	P0	ACFM	MDOT	P1	P5	P3	RHO3	RE3	RATIO
1	523.1	14.690	6.54	.0083	14.661	14.641	14.651	.0756	6208.6	7.532
2	523.1	14.690	8.24	.0104	14.652	14.621	14.637	.0755	7822.9	7.504
3	521.8	14.690	7.83	.0099	14.656	14.634	14.646	.0757	7464.3	5.823
4	521.8	14.690	13.58	.0172	14.620	14.539	14.581	.0754	12938.2	7.236
5	521.8	14.690	21.48	.0272	14.580	14.403	14.495	.0748	20464.7	6.249
6	521.8	14.690	25.59	.0324	14.560	14.321	14.445	.0745	24383.5	5.903
7	522.0	14.690	31.34	.0397	14.530	14.202	14.373	.0740	29849.3	5.391
8	522.0	14.690	37.48	.0475	14.505	14.087	14.306	.0735	35694.8	4.773
9	522.2	14.690	45.98	.0582	14.451	13.813	14.147	.0724	43767.3	4.769
10	522.2	14.690	52.39	.0663	14.415	13.619	14.037	.0717	49859.9	4.528
11	522.4	14.690	59.27	.0750	14.379	13.405	13.918	.0708	56372.6	4.279
12	522.4	14.690	65.15	.0824	14.344	13.196	13.802	.0699	61966.4	4.126
13	522.6	14.690	70.33	.0890	14.308	12.953	13.668	.0689	66858.6	4.123
14	522.6	14.690	75.17	.0951	14.279	12.748	13.556	.0681	71456.6	4.029
15	522.6	14.690	75.82	.0959	14.285	12.747	13.560	.0681	72078.0	3.978
16	522.7	14.690	80.43	.1017	14.255	12.531	13.443	.0672	76407.1	3.914
17	522.7	14.690	84.74	.1071	14.215	12.295	13.311	.0662	80507.0	3.864
18	522.7	14.690	89.06	.1126	14.199	12.090	13.207	.0653	84606.9	3.796
19	522.7	14.690	92.91	.1175	14.183	11.873	13.097	.0645	88272.0	3.768
20	526.0	14.690	11.91	.0150	14.638	14.588	14.614	.0750	11190.0	5.912
21	526.2	14.690	18.49	.0232	14.596	14.495	14.548	.0745	17367.7	4.875
22	526.2	14.690	23.76	.0298	14.572	14.370	14.475	.0741	22312.2	5.872
23	526.2	14.690	28.56	.0359	14.546	14.268	14.413	.0737	26824.1	5.556
24	526.3	14.690	37.00	.0465	14.511	14.108	14.319	.0730	34726.1	4.752
25	526.3	14.690	45.36	.0570	14.463	13.883	14.188	.0721	42573.5	4.508
26	526.3	14.690	50.83	.0638	14.434	13.727	14.098	.0715	47702.1	4.333
27	526.5	14.690	57.36	.0720	14.388	13.472	13.954	.0705	53805.1	4.348
28	526.5	14.690	65.27	.0819	14.350	13.232	13.822	.0695	61217.9	4.039
29	526.5	14.690	71.33	.0895	14.303	12.932	13.656	.0683	66901.1	4.081
30	526.7	14.690	75.11	.0943	14.284	12.789	13.579	.0677	70403.6	3.980
31	526.7	14.690	78.99	.0991	14.230	12.588	13.456	.0668	74047.3	3.901
32	526.7	14.690	83.74	.1051	14.228	12.397	13.367	.0661	78493.8	3.827
33	526.7	14.690	89.01	.1117	14.164	12.123	13.206	.0649	83434.4	3.707
34	526.7	14.690	94.08	.1181	14.130	11.825	13.048	.0637	88189.8	3.677

EXIT LOC014022

Table 17

WAIT-LAST INPUT IGNORED
 START-013306, PROG SIZE(I/D)-5651/2513
 SYS\$RLIBS. LEVEL 74R1GA
 END MAP. ERRORS: 0 TIME: 5.415 STORAGE: 24408/0/035777/02027110011012022

5 FOOT TUBE, AIR

CASE	TEMP	P0	ACFM	MDOT	P1	P5	P3	RH03	RE3	RATIO
1	524.2	14.690	1.84	.0023	14.681	14.680	14.680	.0756	1735.6	5.337
2	524.2	14.690	3.74	.0047	14.669	14.663	14.666	.0755	3533.3	7.505
3	524.2	14.690	8.13	.0103	14.652	14.634	14.643	.0754	7686.4	4.524
4	524.4	14.690	10.49	.0132	14.644	14.620	14.633	.0753	9915.3	3.528
5	524.4	14.690	15.81	.0199	14.610	14.552	14.583	.0750	14935.0	3.814
6	524.5	14.690	23.55	.0297	14.575	14.463	14.523	.0746	22241.6	3.308
7	524.5	14.690	29.00	.0365	14.542	14.380	14.467	.0742	27383.8	3.152
8	524.5	14.690	34.12	.0430	14.517	14.304	14.419	.0738	32216.3	2.971
9	524.7	14.690	39.12	.0493	14.495	14.241	14.379	.0735	36915.0	2.689
10	524.7	14.690	43.98	.0554	14.468	14.154	14.325	.0731	41498.4	2.611
11	524.9	14.690	51.61	.0650	14.418	13.987	14.221	.0723	48670.3	2.577
12	524.9	14.690	60.67	.0764	14.365	13.788	14.103	.0713	57215.4	2.463
13	525.1	14.690	69.49	.0875	14.318	13.607	13.997	.0704	65495.5	2.287
14	525.1	14.690	75.20	.0947	14.277	13.432	13.896	.0696	70881.2	2.291
15	525.1	14.690	80.26	.1010	14.247	13.305	13.824	.0689	75647.9	2.222
16	525.1	14.690	86.17	.1085	14.205	13.114	13.715	.0680	81219.4	2.201
17	525.1	14.690	91.82	.1156	14.183	12.992	13.651	.0673	86543.2	2.095
18	525.1	14.690	97.01	.1221	14.139	12.776	13.530	.0663	91433.7	2.116
19	524.9	14.690	102.69	.1293	14.089	12.573	13.414	.0652	96845.1	2.066
20	524.9	14.690	109.78	.1382	14.053	12.313	13.279	.0639	103532.6	2.032
21	524.9	14.690	113.98	.1435	14.023	12.148	13.191	.0630	107495.6	2.005
22	526.5	14.690	10.67	.0134	14.644	14.620	14.633	.0750	10007.4	3.427
23	526.5	14.690	16.99	.0213	14.614	14.560	14.589	.0747	15937.7	3.105
24	526.5	14.690	24.70	.0310	14.567	14.446	14.510	.0742	23165.2	3.260
25	526.5	14.690	30.16	.0379	14.539	14.370	14.461	.0739	28292.4	3.052
26	526.5	14.690	36.09	.0453	14.507	14.276	14.400	.0734	33852.1	2.885
27	526.5	14.690	41.36	.0519	14.487	14.215	14.363	.0731	38794.0	2.580
28	526.5	14.690	50.58	.0635	14.433	14.040	14.255	.0723	47442.3	2.462
29	526.5	14.690	55.52	.0697	14.405	13.940	14.194	.0718	52075.4	2.402
30	526.5	14.690	63.23	.0794	14.352	13.739	14.074	.0709	59302.9	2.408
31	526.5	14.690	68.36	.0858	14.330	13.649	14.023	.0704	64121.3	2.274
32	526.7	14.690	76.23	.0957	14.283	13.453	13.911	.0694	71453.5	2.200
33	526.7	14.690	82.55	.1036	14.236	13.248	13.792	.0684	77382.2	2.202
34	526.7	14.690	86.96	.1091	14.205	13.109	13.714	.0677	81520.0	2.177
35	526.7	14.690	90.92	.1141	14.175	12.984	13.642	.0671	85225.4	2.144
36	526.7	14.690	95.92	.1204	14.145	12.830	13.558	.0663	89919.0	2.101
37	526.7	14.690	100.40	.1260	14.082	12.639	13.439	.0653	94118.5	2.074
38	526.7	14.690	106.00	.1330	14.078	12.460	13.357	.0644	99367.9	2.058
39	526.7	14.690	110.16	.1382	14.048	12.289	13.265	.0636	103258.6	2.045
40	526.7	14.690	113.32	.1422	14.008	12.142	13.178	.0628	106223.0	2.025

Figure 36

had too many inconsistencies when reduced to be included in this report. They will have to be repeated under more controlled conditions.

An uncertainty analyses on the spirally fluted tube friction factor test data for both water and air flow yielded the following uncertainties:

<u>Water Test</u>	<u>Air Test</u>
$\pm 10\%$ at Re = 20,000	$\pm 5\%$ at Re = 30,000
$\pm 5\%$ at Re = 90,000	$\pm 4\%$ at Re = 55,000
$\pm 2\%$ at Re = 140,000	$\pm 4\%$ at Re = 80,000

Error bars reflecting these uncertainties are shown in Fig. 36. The error analysis is shown in Appendix II.

The data shown in Fig. 36 indicates that the friction coefficient for the spiral fluted tube is the same as that for a smooth tube within the uncertainty of the measurements below a Reynolds number of 80,000. At higher values of Reynolds number it appears to be less than that of a smooth tube by approximately 5-10%. The air and water data are essentially the same.

5. APPLICATION STUDY

The selection of a heat exchanger application to serve as the basis for a comparative study of spiral fluted tubes and plain tubes was guided by the criteria of single phase liquid flow on both the tube-side and shell-side and that the application would lead to the conservation of low temperature energy. In the Residual Energy Applications Program (REAP) there was an isolation water/water heat exchanger used to transfer the heat from the gaseous diffusion plant to an external system for energy recovery.

The performance requirement of this heat exchanger(6) are shown in Table 18.

The heat exchanger selected in Reference 6 was an Alpha Laval plate type of which 48 units were required for the specified flow. Titanium was the material of construction and the cost/unit was \$302,000. The cost of the 48 heat exchangers is then \$14,500,000. Each unit was 456 ft³ in volume with a plan area of 96 ft². The installation of these could be accommodated in 115 ft² resulting in an installation plot of at least 5530 ft² required for the entire field.

The initial purpose of this application study was to compare spiral fluted tubes with conventional plain tubes. The comparison was enlarged to include plate type exchangers as they were used in the reference (6) study. Therefore the three compared heat exchanger types are SFT, plain tubes and plate type exchangers.

As the high cost of these exchangers results from the choice of material it was also decided to base the comparison on AISI type 304 stainless steel. The reason for this is that 304 is acceptable for this application. Since the base material cost is the major cost in the heat exchangers, high cost materials unduly favors enhanced tubing such as spiral fluted tubing.

A design study was made with spiral fluted tubes as well as with

TABLE 18

	HOT FLUID	COLD FLUID
FLUID	WATER	WATER
LOCATION	SHELLSIDE	TUBESIDE
FLOW RATE (GPM)	128,000	128,000
INLET TEMPERATURE (°F)	155	125
OUTLET TEMPERATURE (°F)	130	150
MAXIMUM PRESSURE DROP (PSI)	5	5
FOULING (HR FT ² °F/BTU)	0.0003	0.0003

conventional plain tubes using the Heat Transfer Research, Inc. (HTRI) CST-4 computer program. Two tubing sizes were used in the design study and the results are shown in Table 19. The justification for the lower fouling on the shell-side of the spiral fluted tube is from a test performed at Oregon State University by Prof. J. Knudson in comparing fouling between a spiral fluted tube and a plain tube. He found a reduction of approximately 30% for the spiral fluted tube; the fouling was the same except that the SFT had 33% more area. This justifies the reduction of the fouling factor by one third on the outer diameter of the tube as this was the only case tested. Cases 2 and 3 were selected for the design to be used in the cost study. While an argument could be made for the 5/8 inch plain tube it was eliminated; the length to diameter of this tube was such that its flexibility would have made the assembly a more difficult task, and the 35% increase in the amount of tubing would have more than mitigated the advantages in the reduction of the number of shells for the smaller tube. The differences between case 2 - spiral fluted, and case 3 - plain, are in the number of shells, 6 and 16, the amount of tubing footage required 1,001,100 vs. 3,272,000, and in surface area, 453,076 ft² vs. 1,070,000 ft². This is a dramatic illustration of the advantages that result from the enhancement and area extension of the spiral fluted tube.

Alpha Laval was requested to update their design with AISI Type 304 stainless steel. They found that by using a different configuration and a once-through design, considerable improvement was possible over the previous design in Reference 6. Their new design is given in Table 20. However, in this design there was no fouling factor. Their position is that the high shear in the plate exchanger justifies this assumption. This argument can also be made for the spiral fluted tube as the basic reasons for the observed enhancement is the creation of an instability in the laminar sublayer. Therefore we recalculated the cases previously studied without any provision for a fouling factor. The results are shown in Table 21 for the previous cases 2 and 3. Sixteen shells are still required for the plain tube but they are somewhat smaller and require less tubes. The required surface is reduced by approximately 20%.

For case 2, the spiral fluted tube, the number of shells is reduced to 4

TABLE 19
IN-SERVICE HEAT EXCHANGERS

CASES	1	2	3	4	5	6
	SFT	SFT	PLAIN	SFT	SFT	PLAIN
TUBE O.D. (W)	1-1/4	1-1/4	1-1/4	5/8	5/8	5/8
SHELLS IN SERIES	2	2	4	1	1	2
SHELLS IN PARALLEL	3	3	4	3	3	3
TOTAL SHELLS	6	6	16	3	3	6
SHELL I.D. (IN.)	102	102	107	102	102	117
TUBE LENGTH (FT)	42	47	50	41	43	47
TUBES/SHELL	3,645	3,550	4,090	12,850	13,500	17,500
TOTAL SURFACE (FT ²)	415,102	453,076	1,070,759	359,540	397,607	807,487
TOTAL FT OF TUBING	918,540	1,001,100	3,272,000	1,580,550	1,741,500	4,935,000
HEAT DUTY (MMBTU/HR)	1,572	1,572	1,572	1,572	1,572	1,572
U (BTU/HR FT ² F)	767	707	301	884	803	398
TUBESIDE FOULING RES	0.0002	0.0003	0.0003	0.0002	0.0003	0.0003
SHELLSIDE FOULING RES	0.0002	0.0002	0.0003	0.0002	0.0002	0.0003
CONTROLLING RES	TUBESIDE	FOULING	TUBESIDE	FOULING	FOULING	TUBESIDE
ΔP, SHELL (PSI)	4.32	4.07	4.75	2.96	3.50	4.44
ΔP, TUBE (PSI)	3.09	3.53	2.37	4.95	4.77	4.25
V, SHELL (FT/S)	3.9	3.8	2.7	3.5	3.7	2.7
V, TUBE (FT/S)	3.4	3.5	2.3	4.3	4.1	3.1

Table 20

Alpha-Laval Plate Heat Exchanger

PHE PERFORMANCE FOR COOLING OF WATER ONE OF 36 UNITS

AX30 HA 1 PHE IN PARALLEL. PER PHE 625 PLATES. TOTALLY 625.

TOT. AREA PER UNIT 10629.37 SQFT TOTALLY 10629.37 SQFT

		DUTY REQUIREMENTS	
FLOWTYPE		COUNTER CURRENT	
MASS FLOW RATE	LB/H	1777777.0/	1777777.0
INLET TEMP.	F	155.00/	125.00
OUTLET TEMP.	F	129.99/	150.01

		PHYSICAL PROPERTIES	
NAME OF FLUID		WATER/	WATER
LIQUID NUMBER		1/	1
REF. TEMPERATURE	F	142.497/	137.509
VISCOSITY	CST	0.464/	0.484
VISCOSITY - WALL	CST	0.473/	0.475
SPECIFIC GRAV		0.982/	0.983
SP. HEAT	BTU/LB, F	0.997/	0.997
TH. COND.	BTU/FT, H, F	0.378/	0.376

		PERFORMANCE DATA	
HEAT LOAD	BTU/H	44332054.5	
TRANSF. AREA EFF.	SQFT	10595.352	
LOG. MEAN TEMP. DIFF	F	4.99	
K-VALUE	BTU/SQFT, H, F	838.7/	838.7/ 838.7
FOULING	SQFT, H, F/BTU	-0.00/	0. / 0.
ADD. MARGIN	PCT	0. /	-0.0/ -0.0
THETA REACHED	HTU	5.013/	5.014
PRESS. DROP TOTAL	PSI	4.184/	4.194
PRESS. DROP CONN.	PSI	1.219/	1.217
CONN. VELOCITY	FT/S	10.59/	10.58
NET PUMP EFFECT	KW	6.58/	6.59

		CONSTRUCTION DATA	
NO OF CONNECTIONS		1/	1
CONN. LOCATION INLET		S1/	S3
CONN. LOCATION OUTL.		S2/	S4
CONN. DIAMETER	INCH	12.00/	12.00
CONN. MATERIAL		RUBB. SPOOL/RUBB. SPOOL	
PLATE THICKN./	MTRL.	0.6 MM	AISI 304
PLATE GASKET	MTRL.	NITRILE/	NITRILE
CHAN. ARRANG. SIDE 1		F 1*312 H	
CHAN. ARRANG. SIDE 2		F 1*312 H	
NO OF PLATES TOTAL		625	
NO OF PLATES EFF		623	
WORKING TEMP.	F	200.0	
WORKING PRESSURE	PSI	150.0	
TEST PRESSURE	PSI	225.0	

Table 21

Clean Heat Exchangers (O Fouling)

<u>Case #</u>	<u>2</u> <u>SFT</u>	<u>3</u> <u>Plain</u>
Tube O.D. (in.)	1-1/4	1-1/4
Shells in Series	2	4
Shells in Parallel	2	4
Total Shells	4	16
Shell I.D. (in.)	103	98
Tube Length (ft.)	37	47
Tubes/Shell	3400	3300
Total Surface (ft. ²)	231,430	812,101
Total Feet of Tubing	503,200	2,481,600
Heat Duty	1572	1572
U(BTU/Hr.Ft. ²)	1374	401
Controlling	Tubeside	Tubeside
ΔP , Shell (PSI)	4.98	4.93
ΔP , Tube (PSI)	4.82	2.95
V, Shell (ft./sec.)	5.03	3.0
V. Tube (ft./sec.)	5.26	2.7

and the required surface by almost a factor of 2, as is the total amount of tubing. The pressure drop has increased but is still within the specified limit. This illustrates the importance of fouling but it cannot be seriously considered without experimental substantiation. Cases 2 and 3 from Table 19 will be used for the cost comparison of the tubular designs, and a fouling factor of .0002 for each side of the plate side exchanger will be used to penalize the overall coefficient of the plate exchanger. This assumes that the fouling can be compensated by an increased heat transfer area within the proposed design. This results in a penalty factor of 1.336 which will be applied to their estimated price.

6. COST ESTIMATE OF HEAT EXCHANGERS

6.1 Spiral Fluted Tubing Cost Estimate

The prototype tube forming mill shown in Fig. 5 produces spiral fluted tubing from fluted strip. The fluted strip is made from coils of strip which have been slit to the proper width from mill sheet width. The cost of the strip ready for the roll-forming of the flutes was supplied by vendor quotations in response to inquiries.

6.1.1 Assumptions

- o Tube forming and welding is accomplished in a separate section of an existing manufacturing facility.
- o Two tube forming mills and one roll forming machine.
- o Two 8-hour shift operation.
- o 30° helix and 0.020 AISI 304SS.
- o Labor to operate 3.2 man average over 4160 hrs/year (2 shifts) 13,312 manhours/year.
- o Tungsten inert gas arc welding (TIG).
- o Efficiency 0.85 (labor).
- o Yield 5% scrap roll forming
5% scrap tube forming

6.1.2 Production Capacity

- o Tube welding 5.0 ft/machine minute.

- o Tube forming $(5)(2)(.866)(249,600) = 2.16 \times 10^6$ ft/yr.
($\cos 30^\circ = 0.866$)
- o Roll forming $(37.5) \times (249,600) = 9.4 \times 10^6$ ft/yr.

6.1.3 Production Realization

Efficiency x tube forming capacity.

- o Tube forming $.85 \times 2.16 \times 10^6 = 1.833 \times 10^6$ ft/yr.
- o Roll forming $.85 \times 9.4 \times 10^6 = 8,000,000$ /year.

6.1.4 Capital Cost Estimate

- o Twenty-four stand roll forming machine including tooling, payout and takeup rolls and miscellaneous hardware - \$127,000.
- o Two production version of prototype tube forming machines @ \$75,000/each - \$150,000.
- o Two flying cutoffs for tube mills @ \$12,500/each - \$25,000.
- o Two welding machine and automatic voltage control heads - \$8,000.
- o Test and checking equipment - \$26,000.
- o Spare rolls, bearing and spacers for roll forming - \$17,000.
- o Die repair and maintenance for tube forming - \$13,000.
- o Eddy current weld inspection, 2 @ \$15,000 - \$30,000.

Total capital required: \$396,000

Assume 5 year depreciation

Capital cost/ft. tube = $\$396,000 / 5 \text{ yr.} \times 1.83 \times 10^6 \text{ ft./yr} =$
\$0.043/ft.

6.1.5 Labor Cost

$$\begin{aligned} 3.2 \text{ manhours @ } \$9.88/\text{hr.}^* &= \$31.62 \\ \text{Fringe benefits and overhead @ } \$120\% &= \$37.94 \\ \text{Labor Cost} &= \$69.56/\text{hr.} \end{aligned}$$

$$\text{Labor cost/ft.} = \frac{(69.50)(4160)}{1,830,000} = \$0.1581/\text{ft.}$$

6.1.6 Operating Costs

The operating costs are primarily electricity and argon gas for the tube mill and electricity for the roll forming machine.

$$\text{Roll forming machine } 10\text{KW} \times 925 \text{ hrs.} = 9.25 \times 10^3 \text{ KW/hrs.}$$

$$\begin{aligned} \text{Tube forming including welding } 2 \times 2.2\text{KW} \times 4160 \text{ hrs.} &= \\ 18.3 \times 10^3 \text{ KW/hrs.} \end{aligned}$$

$$\text{Electricity } 27.55 \times 10^3 \text{ KW/hrs. @ } 0.12 = \$3.306/\text{yr.}$$

$$\begin{aligned} \text{Argon for welders } 20 \text{ SCFH} \times 2 \times 4160 &= 166,640 \text{ SCF/yr. @ } 0.041/\text{ft}^3 \\ &= \$6,832. \end{aligned}$$

$$\text{Miscellaneous water, lubricating oil, etc.} = \$1,000.$$

$$\text{Total operating cost} = \$11,138$$

$$\begin{aligned} \text{Operating cost/ft.} &= \$11,138 \\ 1.183 \times 10^6 \text{ ft.} &= \$0.0094/\text{ft.} \end{aligned}$$

6.1.7 Subtotal Base Cost

$$\begin{aligned} \text{Capital} + \text{Material} + \text{Labor} + \text{Operating} \\ 0.043 + 0.4334^{**} + 0.1581 + 0.0094 &= \$0.6439/\text{ft.} \end{aligned}$$

*Labor cost assumes 1 @ \$12.00/hr, 1 @ \$10.00/hr and 1.2 @ \$8.00/hr.

**Material cost based on telephone quote for slit coils of 1500#.

6.1.8 Yield

5% scrap in roll forming.

Cost of scrap in roll forming.

Loss of material - 0.4334/ft.

Additional roll forming 1/4 labor = $\frac{\$0.1583}{4 \text{ ft.}} = \$0.0396/\text{ft.}$

Cost 0.4334/ft. + .0396 = 0.4730/ft.

$(.05) (0.473) (1.83 \times 10^6) = \$43,277$

+ operating cost $(.05) (9250) (0.12) = 55$

Cost of scrap from roll forming = \$43,332

Roll forming scrap/ft. = $43,332 / 1,830,000 = \$0.0237/\text{ft.}$

5% scrap in tube forming

Cost of scrap in tube forming

5% base cost $(.05) (.6439) = \$0.0322/\text{ft.}$

Cost of yield loss = roll forming and tube forming

$0.0257 + 0.0322 = \$0.0559/\text{ft.}$

6.1.9 Contingency

Assume 10% of base cost + yield loss

$(0.10) (0.6439 + .0559) = \$0.0700/\text{ft.}$

6.1.10 Factory Cost

Base cost + yield + contingency

$0.6439 + .0559 + .0700 = \$0.7698/\text{ft.}$

6.1.11 G&A

Assume 20% = .1540.

6.1.12 Total Cost

Factory cost + G&A = \$.9237/ft.

6.1.13 Profit

Assume 20% = 0.1847.

6.1.14 Price

Factory cost + GA + Profit = \$1.1084.

6.1.15 Alternate Estimate

Results of inquiries from a number of welded tubing manufacturers established the current market price for 1-1/4" O.D. by .028 wall tubing at \$.67/ft. The forming and welding process for smooth tubing is similar to that of spiral fluted tubing except for the additional fluting process. The welding speed, tube testing and handling are identical and the forming machines are of comparable complexity except as noted above for the fluting process. Therefore the price of spiral fluted tubing can be estimated from the quoted price of plain tubing by accounting for the incremental material and fabrication costs.

The material costs of the .028 strip is estimated to be \$.3825/ft. based on the quoted price of the .020 strip has a \$.055 add-on for the smaller gauge material. Subtracting the material price from the quoted tubing price of \$.67/ft. results in a price of \$.2874/ft. for the fabrication of plain tubing from strip. When the helix angle of the spiral fluted tubing is considered, the tube forming cost from spiral strip is increased to \$.3319. An estimate of the fluting cost was obtained from Technology, Inc. a roll forming manufacturer. Their estimate which includes their G&A and a profit of 20% when converted to a cost per linear foot of tube is less than \$.02/ft. Using the material cost of 0.4334 marked up by 20% the resulting price of the spiral fluted tubing would be \$.8720/ft. This is only 0.79% of the above estimate. The present tubing market is very competitive and the assumed margins in our estimate may not be achieved with the \$.67 price that was quoted to us. We will use the price as derived in our estimate.

6.2 Cost Estimate of Heat Exchangers

A cost estimate made for the case 2 and 3 heat exchangers of Table 19 is summarized in Table 22. The basis for the rates and markups used in the cost estimate are representative for heat exchanger fabrication in the Southern California area and are as follows:

Labor rate	\$12.00/hr.
Fringe and overhead @ 120%	\$14.40/hr.
Hourly labor cost	\$26.40/hr.
Tubing was taken as a pass through cost	
General and administrative cost was applied at 20% rate	
Profit was 20%.	

The cost ratio of the SFT exchangers to plain tube exchangers is 0.4129.

Alpha Laval estimated the cost for the frame heat exchangers of Table 20 as \$77,500 each so that the cost for the 36 units required is \$2,790,000 for a cost ratio of 0.765. These units do not have any allowance for fouling. If a minimum fouling factor of 0.0001 were used as a combined factor the cost would be \$3,027,150 resulting in a cost ratio of 0.7048 in favor of the spiral fluted tube exchangers. However, if the plate exchangers are penalized to a fouling factor of 0.0002 on each side, the resulting cost would be \$3,727,440 for the plate heat exchangers and the cost factor becomes 0.5724.

Table 22

Heat Exchanger Cost Estimate

Case # from Table 19	2(1-1/4" SFT)	3(1-1/4" plain tube)
Labor cost	2969 hrs. @ 26.40=78,250	3189 hrs. @ 26.40=84,190
Material	40,245	44,910
Factory cost	118,495	129,100
G&A	23,699	25,820
Profit	28,439	30,984
Tubing	@ \$1.084/ft. 184,937	@ \$0.67/ft. 137,000
Unit heat exchanger	355,570	322,920
Total heat exchanger	6 required \$2,133,420	16 required 5,166,714

7. DISCUSSION OF RESULTS

The results of the heat transfer tests on the 7-tube spirally fluted heat exchanger indicate an enhancement factor relative to a plain tube of 1.59 for the tube-side and 2.28 for the shell-side. This is in addition to the area extension of 1.6. No variation of enhancement with heat flux was observed, nor did the enhancement decrease when the heat flux direction was reversed. Previous measurements on spiral-fluted tubing did show a dependency of the enhancement factor for large values of heat flux. This is shown in Fig. 1. The present results are in agreement with curve 2, the Reilly tests where the enhancement ratio was 1.59-1.64. Curves 3 and 4, the HTRI low flux and high flux tests indicated enhancement factors of 1.86 and 2.34 respectively. The minimum enhancement factor for curve 1, the HTRI tests with the heat flux direction reversed, was 1.22. The data from the present tests indicates a minimum enhancement factor of 1.6 which is greater than the factor of 1.22 previously observed. (The minimum enhancement is the value observed when the direction of the heat flux is reversed.) Geometric differences of the tube geometry offer an explanation for the improved minimum enhancement. The ratio of flute height to the equivalent diameter of the cross sectional area was 0.0465 for the tube used in the HTRI test and 0.0566 for the tubes used in the seven tube heat exchanger. This increase of the flute height ratio in excess of 20%, could have improved the performance by increasing the secondary flow within the flute resulting in transport of the laminar sublayer away from the wall.

A purely empirical approach to a heat flux parameter is proposed to compare the data. As there were variations in the tube dimensions, heat flux and Reynolds number for each of the test data, the heat flux parameter suggested is

$$\psi^* = \frac{\psi J}{R_e^{1/5} \mu} \frac{\epsilon}{D_h}$$

ψ^* is dimensionless and has a mean value of 4×10^8 for the high flux HTRI test data (curve 4 of Fig. 1), 1.91×10^8 for the low flux HTRI test (curve 3,

Fig. 1), 0.416×10^8 for the Reilly test (curve 1 Fig. 1), 0.403×10^8 for data set #5 and 0.393×10^8 for data set 9 of the 7-tube bundle tests. The heat flux parameter for the Reilly data and the 7-tube data are essentially the same as is the enhancement factor. Both the heat flux parameter and enhancement factor increase for the low flux HTRI test and further increase for the high flux HTRI test.

Where

ψ	=	heat flux ($h_i \Delta T = Q/A$)	BTU/hr.ft ²
J	=	mechanical equivalent of heat	ft. #/BTU
Re	=	Reynolds number	
μ	=	absolute viscosity	#/hr-ft.
ϵ	=	flute height	ft.
D_h	=	hydraulic diameter	ft.
Q	=	heat transferred	BTU/hr.
A	=	heat transfer area	ft ²
h_i	=	heat transfer coefficient on inside of tube	BTU/ft ² hr.
ΔT	=	temperature difference between the tube wall and bulk temperature of the fluid.	OF

ψ^* seems to indicate a threshold for the influence of buoyancy forces which results from the increase of heat flux directed radially inward. Moreover, this threshold is greater than the values of heat flux achieved in these tests and less than that of the low flux HTRI tests.

It would appear that at least two mechanisms for enhancement exist in the spiral fluted tube, one below this threshold and another above it. One can speculate that the secondary flow induced by the flutes causes both a displacement of the laminar sublayer and a rotation of the core flow. At heat flux levels above the threshold value the rotation leads to instability at the wall while at heat flux values less than this threshold, the secondary flow within the flutes is the mechanism for transporting the heat from the wall to the through flow. The fact that enhancement is achieved with the heat flux radially outward supports this speculation.

A rational explanation for the influence of heat flux on heat transfer

enhancement is not possible at this time because of the absence of a theoretical model for the fluid mechanics in spiral fluted tubes. Theoretical and experimental work underway at present with support from the Office of Naval Research might provide this model at some future time.

The measurements of the frictional coefficient accomplished in this program while essentially equivalent to that of a smooth plain tube, disagree with the data of Fig. 2. This was not surprising in that it had been thought that confirming tests were required before the data of Fig. 2 could be accepted. However this does not alter the basic fact that enhancement of heat transfer is achieved without a frictional penalty.

The cost comparison accomplished in this study shows the advantage of enhanced and surface extended tubing that is competitive in price with plain tubing. While the focus of this comparison was with a plain tube heat exchanger, it was found that a plate type heat exchanger was much more cost effective than the plain tube units, but the spiral fluted tube heat exchanger was still the economic choice. It should be noted that the original plate exchanger in the reference design could be considerably improved.

8. CONCLUSIONS

The work reported herein supports the premise that spiral-fluted tubing can materially reduce the cost of heat exchanger equipment in single phase liquid service. The reasons for this are: the heat transfer enhancement and area extension on the tube-side by a factor of 2.54 and on the shell-side by a factor of 3.65 leads to a substantial decrease in the tubing quantity required when compared to a plain tube. The heat transfer enhancement is achieved without any increase in tube-side friction coefficient.

In the example studied, the total surface area required was reduced from 1.071×10^6 to 0.453 ft^2 , the total length of tubing required was reduced from 3.27×10^6 to 1.001×10^6 feet, and the number of shells required from 16 to 6. The six spiral fluted tubing heat exchangers would require a 3240 ft^2 area while the sixteen plain tube exchangers would need a 8736 ft^2 area. The new plate exchangers would require an installation area of 4147 ft^2 (36 units instead of the 48 specified in Ref. 6).

9. REFERENCES

1. Yampolsky, J. S., Spirally Fluted Tubing for Augmented Heat Transfer, GA-A15442, 1979.
2. Reilly, D. J., "An Experimental Investigation of Enhanced Heat Transfer on Horizontal Condenser Tubes," M.S. Thesis, Naval Postgraduate School, 1978.
3. Thermodynamic and Transport Properties of Steam 1967 ASME Steam Tables, New York (1967).
4. Wilson, E. E., "A Basis for Rational Design of Heat Transfer Apparatus," ASME Transactions, p. 47-83, 1915.
5. Kays, W. M., "Loss of Coefficients for Abrupt Changes in Flow Cross Section with Low Reynolds Number Flow in Single and Multiple-Tube Systems," Transaction of the ASME, November, 1950.
6. Lockheed Missiles and Space Company Report LMSC D-566926.

APPENDIX 1

COMPUTER PROGRAM LISTING

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1: REAL MFLOW(50),MFLOWT(50),LMTD,NU,NUP
2: DIMENSION TSIN(50),TSOUT(50),TTIN(50),TTOUT(50),LMTD(50),HS(50)
3: DIMENSION QSIN(50),QTIN(50),QS(50),QT(50),HEATB(50),HT(50),NU(50)
4: DIMENSION U(50),TAUGS(50),TAUGT(50),Q(50),RE(50),ST(50),DPSO(50)
5: DIMENSION U(50),X(50),Y(50),NUP(50),XLN(50),VLN(50),UHSI(50)
6: DIMENSION XP(2),YP(2),UAUGS(50),UAUGT(50),IALPHA(2,3),UHSO(50)
7: DIMENSION DPTR(50),DPST(50),DPOT(50),FT(50),F(50),XJ(50),FAC(50)
8: DIMENSION DPTI(50),DPTO(50),UHTI(50),UHTO(50),DPSR(50),DPSI(50)
9: DIMENSION DPSS(50),DPOS(50),FS(50),VPNU(2),FACPLT(50),REPLT(50)
10: DIMENSION XFREQS(50),XFREQT(50),PRESS(50),XKR(50),S(50),SN(50)
11:
12:
13: C PLOTTING ARRAYS WITH NEW ON-LINE PLOT SYSTEM.
14:
15: DIMENSION IPTS1(2),IPTS2(3),LINE1(2),LINE2(3),ISYMB1(2),ISYMB2(3)
16: DIMENSION ILEN1(2,2),ILEN2(3,2),ITITL1(6),ITITL2(7),ITITL3(4)
17: DIMENSION ITYPE(3,2),LR2A(6),LR1(6)
18:
19: DATA IALPHA /6H SHELL ,6H TUBE S,6H SIDE U,6H IDE U ,4H SIDE,3H IDE /
20: DATA ILEN1 /6H DATA ,3H FIT,2H 1H /
21: DATA ILEN2 /6H DATA ,6H FIT ,3H D-B,3H 1H /
22: DATA IPTS1 /0,2 /
23: DATA IPTS2 /0,2,2 /
24: DATA LINE1 /-4,2 /
25: DATA LINE2 /-4,2,0 /
26: DATA ISYMB1 /4,0 /
27: DATA ISYMB2 /4,0,0 /
28: DATA ITITL1 /6H WILS,6H ON PLO,6H T - ,6H XXXXXX,6H XXXXXX,0 /
29: DATA ITITL2 /2H XXXXXX,6H NU-R,6H EY COR,6H RELATI,2H ON,0 /
30: DATA ITITL3 /6H XXXXXX,6H XXXXXX,6H 2H J/F ,0 /
31: DATA LR1 /6H DAT,6H A SET ,6H XXXXXX,6H XXXXXX,6H XXXXXX,0 /
32: DATA LR2A /6H NU/P,6H RXX.4-,6H XXXXXX,6H REXX ,6H XXXXXX,0 /
33: DATA ITYPE /6H AVERA,6H SHELL,6H TUBES, 4H GE Q,6H SIDE Q,5H IDE Q /
34:
35: CALL LOGIN
36: FACTK=((1000.0*0.1337)/(0.90375*60.0)
37: ICURU=2
38: T=0.040/12.0
39: XKM=120.0
40: DHSF=0.0658
41: DHSN=0.0924
42: DHT =0.0549
43: AREAS=0.0718
44: AREAT=0.0429
45: AREAS=32.94
46: AREAT=33.14
47: ART=.34
48: ARS=.5685
49: XKTO=0.42
50: XKSO=0.9
51: XKTI=0.31
52: XKSI=0.45
53: XLT=150.19/12.
54: XLS=10.6
55:
56: C READ NO. OF DATA RUNS AND OTHER FLAGS
57: C
58: READ(5,1) IDATA,IMAX,IFLAG,IQ
59: JDATA=IWORD(IDATA,0)
60: IF(IQ.EQ.0) WRITE(6,5) JDATA

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60:)

APPENDIX 1

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61:      IF(IQ.EQ.1) WRITE(6,6) JDATA
62:      IF(IQ.EQ.2) WRITE(6,7) JDATA
63:      5  FORMAT(2X,'DATA SET',A2,5X,'HEAT RATE BASED ON AVERAGE Q',///)
64:      6  FORMAT(2X,'DATA SET',A2,5X,'HEAT RATE BASED ON SHELLSIDE Q',///)
65:      7  FORMAT(2X,'DATA SET',A2,5X,'HEAT RATE BASED ON TUBESIDE Q',///)
66:
67:C     READ DATA INPUT AND REDUCE DATA TO CONSTRUCT WILSON PLOT
68:C
69:C     IPASS=0
70:C     19 IPASS=IPASS+1
71:      WRITE(6,14)
72:      WRITE(6,130)
73:      130 FORMAT(' RUN FS FT TSI TSO TTI TTO TAUS TAUT
74:      + FLOS FLOT QS QT ERR DTLN US US'//)
75:      DO 10 I=1,IMAX
76:C     IF(IPASS.EQ.2) GO TO 17
77:      READ(5,1)XFREQS(I),XFREQT(I),PRESS(I)
78:      READ(5,1)TSIN(I),TSOUT(I),TTIN(I),TTOUT(I),DPTR(I),DPSR(I)
79:
80: 1      FORMAT( )
81:
82:      17 CONTINUE
83:
84:      CALL STEAM(PRESS(I),TSIN(I),XS,HSAT,CPS,USIN,VISC,COND)
85:      CALL STEAM(PRESS(I),TTIN(I),XS,HSAT,CPS,UTIN,VISC,COND)
86:      CALL STEAM(PRESS(I),TTOUT(I),XS,HSAT,CPS,UTOUT,VISC,COND)
87:      CALL STEAM(PRESS(I),TSOUT(I),XS,HSAT,CPS,USOUT,VISC,COND)
88:      DPTR(I)=DPTR(I)*12.55*62.2/1728.
89:      DPSR(I)=DPSR(I)*12.55*62.2/1728.
90:
91:      QSIN(I)=FACTK*XFREQS(I)
92:      QTIN(I)=FACTK*XFREQT(I)
93:      MFLOUS(I)=QSIN(I)/USIN
94:      MFLOUT(I)=QTIN(I)/UTIN
95:      VELTI=MFLOUT(I)/(AREAT/UTIN)
96:      VELTO=MFLOUT(I)/(AREAT/UTOUT)
97:      UHTI(I)=VELTI*32/(2.*UTIN*32.17*144.)
98:      UHTO(I)=VELTO*32/(2.*UTOUT*32.17*144.)
99:      DPTI(I)=(1.-ART*32+KTI)*UHTI(I)
100:      DPTO(I)=(-1.+ART*32+KTO)*UHTO(I)
101:      DPST(I)=DPTR(I)-DPTI(I)-DPTO(I)
102:      DPOT(I)=DPST(I)+UHTI(I)-UHTO(I)
103:      VELSI=MFLOUS(I)/(AREAS/USIN)
104:      VELSO=MFLOUS(I)/(AREAS/USOUT)
105:      UHSI(I)=VELSI*32/(2.*USIN*32.17*144.)
106:      UHSO(I)=VELSO*32/(2.*USOUT*32.17*144.)
107:      DPSI(I)=(1.-ARS*32+KSI)*UHSI(I)
108:      DPSO(I)=(-1.+ARS*32+KSO)*UHSO(I)
109:      DPSS(I)=DPSR(I)-DPSI(I)-DPSO(I)
110:      DPOS(I)=DPSS(I)+UHSI(I)-UHSO(I)
111:
112:      TAUGS(I)=(TSIN(I)+TSOUT(I))/2.0
113:      TAUGT(I)=(TTIN(I)+TTOUT(I))/2.0
114:
115:      CALL STEAM(PRESS(I),TAUGS(I),XS,HSAT,CPAUGS,UAUGS(I),VISC,COND)
116:      CALL STEAM(PRESS(I),TAUGT(I),XS,HSAT,CPAUGT,UAUGT(I),VISC,COND)
117:
118:      QS(I)=MFLOUS(I)*CPAUGS*(TSIN(I)-TSOUT(I))
119:      QT(I)=MFLOUT(I)*CPAUGT*(TTOUT(I)-TTIN(I))
120:
120:)

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APPENDIX 1

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121: HEATB(I)=(QT(I)-QS(I))/QT(I)*100.0
122:
123: TEMPU=((TSIN(I)-TTOUT(I))-(TSOUT(I)-TTIN(I)))
124: LMTD(I)=TEMPU/(ALOG((TSIN(I)-TTOUT(I))/(TSOUT(I)-TTIN(I))))
125:
126: Q(I)=0.0
127: IF(IQ.EQ.0) Q(I)=(QS(I)+QT(I))/2.0
128: IF(IQ.EQ.1) Q(I)=QS(I)
129: IF(IQ.EQ.2) Q(I)=QT(I)
130:
131: IF(IFLAG.EQ.10) GOTO 25
132: IF(IFLAG.NE.0) GOTO 777
133:
134: U(I)=Q(I)/(LMTD(I)*AREAMS)*3600.0
135: U(I)=MFLWS(I)/(AREAS/VAUGS(I))
136: GOTO 83
137:
138: 25 U(I)=Q(I)/(LMTD(I)*AREAMT)*3600.0
139: U(I)=MFLOWT(I)/(AREAT/VAUGT(I))
140:
141: C PRINT OUT VALUES AND CONSTRUCT WILSON PLOT
142:
143: 83 V(I)=1.0/U(I)
144: X(I)=1.0/(U(I)*X0.8)
145: C IF(IPASS.EQ.1) X(I)=1.0/(U(I)*X0.8)
146: C IF(IPASS.EQ.2) X(I)=1.0/(XKR(I)*X0.8)
147:
148: 14 FORMAT(
149: WRITE(6,987)I,XFREQS(I),XFREQT(I),TSIN(I),TSOUT(I),TTIN(I)
150: 1 ,TTOUT(I),TAUGS(I),TAUGT(I),MFLWS(I),MFLOWT(I),QS(I),QT(I),
151: 2 HEATB(I),LMTD(I),U(I),V(I)
152: 987 FORMAT(I3,2F6.4,6(F7.1),2F7.2,2F8.2,F6.1,F6.1,F7.0,F6.2)
153: 10 CONTINUE
154: 11 FORMAT(2X,I2,7(2X,F7.2))
155: 12 FORMAT(2X,I0(2X,F7.2))
156:
157:
158: CALL LINREG(X,Y,IMAX,A0,A1,COR)
159:
160: XMAX=0.0
161: DO 128 I=1,IMAX
162: 128 IF(X(I).GT.XMAX) XMAX=X(I)
163:
164: XP(1)=0.0
165: XP(2)=XMAX
166: YP(1)=A0
167: YP(2)=A0+XMAX*A1
168: C
169: WRITE(6,133) A0,A1,COR
170: 133 FORMAT(///,2X,'CONSTANTS FOR WILSON PLOT REGRESSION',3X,
171: +'A0=',F7.6,2X,'A1=',E12.4,2X,'COR=',1PE8.3,///)
172: C
173: SUMD=0.0
174: DO 333 I=1,IMAX
175: S(I)=(Y(I)-(A1*X(I)+A0))*X(I)**2.0
176: SUMD=SUMD+S(I)
177: 333 CONTINUE
178: SDS=SUMD/FLOAT(IMAX)
179: SD=SDS**0.5
180: SD2=SD**2.0
180: )

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```

181:      WRITE(6,334) SD,SD2
182: 334 FORMAT(IX,'SIGMA-',E12.5,1X,'2 SIGMA-',E12.5)
183:      CALL TWAIT(20000)
184:C      IF(IPASS.EQ.1) GO TO 18
185:
186:C      PLOT ROUTINES
187:
188:      CALL OPENPT(4,9600,4014,4,50)
189:      IPTS1(1)=IMAX
190:      CALL SETNPT(ICURU,IPTS1)
191:      CALL DATAPT(X(1),0,1,20,1,IMAX,1)
192:      CALL DATAPT(XP(1),0,1,2,1,2,1)
193:
194:      CALL DATAPT(Y(1),0,1,20,1,IMAX,1)
195:      CALL DATAPT(YP(1),0,1,2,1,2,1)
196:      CALL LINEPT(ICURU,LINE1)
197:      CALL SYMBPT(ICURU,ISYMB1)
198:      CALL LEGNPT(2,ICURU,ILEN1)
199:      CALL AXISPT(' 1/U*8 ', ' 1/U ')
200:      ITITL1(4)=IALPHA(IFLAG+1,1)
201:      ITITL1(5)=IALPHA(IFLAG+1,2)
202:      LR1(3)=JDATA
203:      LR1(4)=ITYPE(IQ+1,1)
204:      LR1(5)=ITYPE(IQ+1,2)
205:      CALL TITLPT(ITITL1,1,0,0,LR1,0)
206:      CALL PLOTPT
207:      CALL WAITPT
208:C      CALL HARDPT
209:
210:      18 CONTINUE
211:
212:C      CONSTRUCT PLOT OF NU VS. REV
213:
214:      WRITE(6,131)
215: 131 FORMAT(' RUN   HS      HT      TAUM      TFILM      NU      ST      RE
+ PR   NU/PR*4      U      K      MU      CP',/)
216:
217:      DO 77 I = 1,IMAX
218:      IF(IFLAG.EQ.1) GOTO 48
219:      IF(IFLAG.NE.0) GOTO 777
220:      HS(I)=U(I)*X0.8/A1
221:C      IF(IPASS.EQ.1) HS(I)=U(I)*X0.8/A1
222:C      IF(IPASS.EQ.2) HS(I)=XKR(I)*X0.8/A1
223:      HTINU=1.0/U(I)-1.0/HS(I)-T/XKR
224:      HT(I)=(1.0/HTINU)*X(AREAHS/AREAHT)
225:      HAT=HT(I)*AREAHT
226:      HAS=HS(I)*AREAHS
227:      TH=(HAT*TAUGT(I)+HAS*TAUGS(I))/(HAT+HAS)
228:      TFS=(TH+TAUGS(I))/8.0
229:      CALL STEAM(PRESS(I),TFS,XS,HSAT,CPSF,USF,VISCSF,CONDSF)
230:      PR=VISCFS/CPSF/CONDSF
231:      MU(I)=HS(I)*DHSH/CONDSF
232:      RE(I)=U(I)*DHGF/(USF*VISCFS)*13600.0
233:      XKR(I)=CONDSF*RE(I)*X0.8
234:      SPVOL=USF
235:      FS(I)=(2.*XDPOS(I)*DHGF)/(4.*XLSZU(I)*X2/UAUGS(I))*144.0*32.17
236:      F(I)=FS(I)
237:      CONDF=CONDSF
238:      VISCF=VISCFS
239:      CPF=CPSF
240:      TFILM=TFS
240: )

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APPENDIX 1

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241:      GOTO 49
242:      48 HT(I)=U(I)**0.8/A1
243:      IF(IPASS.EQ.1) HT(I)=U(I)**0.8/A1
244:      IF(IPASS.EQ.2) HT(I)=XKR(I)**0.8/A1
245:      HSINU=1.0/U(I)-1.0/HT(I)-T/XKM
246:      HS(I)=(1/HSINU)**(AREAHT/AREAH)
247:      HAT=HT(I)*AREAH
248:      HAS=HS(I)*AREAH
249:      TM=(HAT*TAUGT(I)+HAS*TAUGS(I))/(HAT+HAS)
250:      TFT=(TM+TAUGT(I))/2.0
251:      CALL STEAM(PRESS(I),TFT,XS,HSAT,CPTF,UTF,VISCTF,CONDTF)
252:      PR=VISCTF*CPTF/CONDTF
253:      NU(I)=HT(I)*DHT/CONDTF
254:      RE(I)=U(I)*DHT/(UTF*VISCTF)*3600.0
255:      XKR(I)=CONDSF*RE(I)**0.8
256:      FT(I)=(2.*DPOT(I)*DHT)/(4.*XLT*U(I)**2/UAUGT(I))*144.0*32.17
257:      F(I)=FT(I)
258:      SPUOL=UTF
259:      CONDF=CONDTF
260:      VISCF=VISCTF
261:      CPF=CPTF
262:      TFILM=TFT
263:
264:      49 ST(I)=NU(I)/(RE(I)*PR)
265:      NUP(I)=NU(I)/PR**0.4
266:      XJ(I)=ST(I)*PR**1.666667
267:      FAC(I)=2.*XJ(I)/F(I)
268:
269:      X(I)=RE(I)
270:      Y(I)=NUP(I)
271:      XLN(I)=ALOG(X(I))
272:      YLN(I)=ALOG(Y(I))
273:
274:      WRITE(6,543) I,HS(I),HT(I),TM,TFILM,NU(I),ST(I),RE(I),PR,NUP(I)
275:      1 ,SPUOL,CONDF,VISCF,CPF
276:      543 FORMAT(I3,2F8.0,3F7.0,F9.5,F10.0,F7.2,F8.1,F10.5,F7.4,F8.4,F8.4)
277:      77 CONTINUE
278:      CALL TWAIT(20000)
279:      WRITE(6,81)
280:      IF(IFLAG.EQ.1) GOTO 79
281:
282:      WRITE(6,81)
283:
284:      WRITE(6,571)
285:      571 FORMAT('          INPUT      VEL      INLET      VEL      OUTLET
286:      1 SHELL      SHELL      FRIC//
287:      1' RUN      DP      HEAD IN      DP      HEAD OUT      DP      STADIC
288:      1DP TOTAL DP      FACT      J FACT      2XJ/F//)
289:      DO 432 I=1,IMAX
290:      432 WRITE(6,560)I,DPSR(I),UHSI(I),DPSI(I),UHSO(I),DPSO(I),DPSS(I)
291:      1 ,DPOS(I),F(I),XJ(I),FAC(I)
292:      560 FORMAT(I4,7(3X,F7.3),2X,F9.5,1X,F9.5,F9.3)
293:      GOTO 76
294:      79 CONTINUE
295:      WRITE(6,561)
296:      561 FORMAT('          INPUT      VEL      INLET      VEL      OUTLET
297:      1 TUBE      TUBE      FRIC//
298:      1' RUN      DP      HEAD IN      DP      HEAD OUT      DP      STADIC
299:      1DP TOTAL DP      FACT      J FACT      2XJ/F//)
300:      DO 772 I=1,IMAX
300:      >

```

```

301: 772 WRITE(6,560)I,DPTR(I),UNTI(I),DPTI(I),UNTO(I),DPTO(I),DPST(I)
302: 1 ,DPOT(I),F(I),XJ(I),FAC(I)
303: 76 CONTINUE
304: C CALL LINREQ(XLN,VLN,IMAX,A0,A1,COR)
305:
306: SUM=0.0
307: A1=0.8
308: DO 176 I=1,IMAX
309: 176 SUM=SUM+A1*XLN(I)-VLN(I)
310: A0=-SUM/IMAX
311: XMAX=0.0
312: XMIN=100000.0
313:
314: DO 78 I=1,IMAX
315: IF(XLN(I).GT.XMAX) XMAX=XLN(I)
316: 78 IF(XLN(I).LT.XMIN) XMIN=XLN(I)
317:
318: XP(1)=EXP(XMIN)
319: XP(2)=EXP(XMAX)
320:
321: VPNU(1)=0.022*XP(1)**0.8
322: VPNU(2)=0.022*XP(2)**0.8
323:
324: YMIN=A0+A1*XMIN
325: YMAX=A0+A1*XMAX
326: VP(1)=EXP(YMIN)
327: VP(2)=EXP(YMAX)
328:
329: C1=EXP(A0)
330:
331: WRITE(6,81)
332: 81 FORMAT(//)
333: WRITE(6,183) C1,A1
334: 183 FORMAT(' NU/PR1X0.4-C1XRESXN C1=',F5.4,' N=',F4.3)
335: SUMN=0.0
336: SUMY=0.0
337: DO 1122 I=1,IMAX
338: SN(I)=(VLN(I)-(A1*XLN(I)+A0))**2.0
339: SUMN=SUMN+SN(I)
340: SUMY=SUMY+VLN(I)
341: 1122 CONTINUE
342: SDSN=SUMN/FLOAT(IMAX)
343: SDN=(SDSN)**0.5
344: SDN2=SDN**2.0
345: VLNBAR=SUMY/FLOAT(IMAX)
346: SDSY=SDSN*EXP(VLNBAR)
347: SDY=SDSY**0.5
348: SDY2=SDY**2.0
349: WRITE(6,1123) SDY,SDY2
350: 1123 FORMAT(1X,'SIGMA=',E12.5,3X,'2 SIGMA=',E12.5)
351: CALL TUAIT(20000)
352: WRITE(6,81)
353: C IF(IPASS.EQ.1) GO TO 10
354: ICURU=ICURU+1
355: IPTS2(1)=IMAX
356: CALL SETNPT(ICURU,IPTS2)
357: CALL DATAPT(X(1),0,1,20,1,IMAX,1)
358: CALL DATAPT(XP(1),0,1,2,1,2,1)
359: CALL DATAPT(XP(1),0,1,2,1,2,1)
360: CALL DATAPT(Y(1),0,1,20,1,IMAX,1)
360: )

```

APPENDIX 1

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361:      CALL DATAPT(YP(1),0,1,2,1,2,1)
362:      CALL DATAPT(YPMU(1),0,1,2,1,2,1)
363:      CALL LINEPT(ICURU,LINE2)
364:      CALL SYMBPT(ICURU,ISYMB2)
365:      CALL LEGNPT(3,ICURU,ILEN2)
366:      CALL RANGPT(1.0E03,1.0E05,0.0,0.0)
367:      CALL AXISPT(' REYNOLDS NUMBER ',' NU/PR1x.4 ')
368:      CALL TRANPT(2,2)
369:      ITITL2(1)=IALPHA(IFLAG+1,1)
370:      ITITL2(2)=IALPHA(IFLAG+1,3)
371: 876  FORMAT('30'/A6,A4,' 2xJ/F '//YES')
372:C    LR2A(3)=C1
373:C    LR2A(5)=A1
374:
375:      ENCODE(654,LR2A(3))C1
376: 654  FORMAT(F6.5)
377:      ENCODE(655,LR2A(5))A1
378: 655  FORMAT(F4.3)
379:      CALL TITLPT(ITITL2,1,LR2A,0,LR1,0)
380:      CALL PLOTPT
381:      CALL WAITPT
382:C    CALL HARDPT
383:
384:C    NEXT PLOT (3)
385:
386:      ICOUNT=0
387:      DO 427 I=1,IMAX
388:      IF(FAC(I).GT.2.5.OR.FAC(I).LT.1.0) GOTO 427
389:      ICOUNT=ICOUNT+1
390:      FACPLT(ICOUNT)=FAC(I)
391:      REPLT(ICOUNT)=RE(I)
392: 427  CONTINUE
393:
394:      ICURU=1
395:      CALL SETNPT(ICURU,ICOUNT)
396:      CALL DATAPT(REPLT(1),0,1,20,1,ICOUNT,1)
397:      CALL DATAPT(FACPLT(1),0,1,20,1,ICOUNT,1)
398:      CALL LINEPT(ICURU,-4)
399:      CALL SYMBPT(ICURU,4)
400:      CALL LEGNPT(1,ICURU,'2xJ/F')
401:      CALL AXISPT(' REYNOLDS NUMBER ',' 2xJ/F ')
402:      CALL TRANPT(2,2)
403:      ITITL3(1)=IALPHA(IFLAG+1,1)
404:      ITITL3(2)=IALPHA(IFLAG+1,3)
405:      CALL TITLPT(ITITL3,1,0,0,LR1,0)
406:      CALL PLOTPT
407:      CALL WAITPT
408:C    CALL HARDPT
409:      CALL TERMP
410:      CALL ENDRUN
411:
412:
413:
414: 777  WRITE(6,779)
415: 779  FORMAT(' INPUT FLAG NOT SET PROPERLY. ')
416:      CALL ENDRUN
417:      END
EOF:417 SCAN:56
0:

```

APPENDIX 1

```
1: SUBROUTINE LINREG(X,V,IMAX,A0,A1,COR)
2:
3: DIMENSION X(IMAX),Y(IMAX),XY(100,2)
4: DIMENSION ABLAB(3),DES(5),ANOVA(14),STAT(8),PRED(20,7)
5:
6: DO 1 I=1,IMAX
7:   XY(I,1)=X(I)
8:   1 XY(I,2)=Y(I)
9:
10:  ABLAB(1)=0.05
11:  ABLAB(2)=0.10
12:  ABLAB(3)=0.01
13:  CALL RLONE(XY,100,IMAX,0,-1,ABLAB,DES,ANOVA,STAT,PRED,
14:    1 20,20,IER)
15:
16:  A0=STAT(5)
17:  A1=STAT(1)
18:
19:  COR=DES(5)
20:
21:  RETURN
22:  END
EOF:22
01>
```

APPENDIX 2

UNCERTAINTY ANALYSES FOR WATER, AIR ΔP TESTS

Define friction factor f from

$$\Delta P = f \frac{L}{D_h} \frac{\rho V^2}{2g} \quad (1)$$

Using the data reduction procedure outlined in Section 4,

$$f = \frac{D_h}{(L_2 - L_1)} \left[\left(\frac{\Delta P}{\frac{\rho V^2}{2g}} \right)_2 - \left(\frac{\Delta P}{\frac{\rho V^2}{2g}} \right)_1 \right] \quad (2)$$

where

D_h = hydraulic diameter = $4A/P$

A = tube cross-sectional flow area

P = tube perimeter

L_2 = length of long tube

L_1 = length of short tube

ΔP = pressure drop across test section

ρ = water density

V = average axial water velocity

g = gravitational constant

The measured values and uncertainties of tube parameters are

$$L_1 = 5.00 \pm 0.0052 \text{ ft}$$

$$L_2 = 9.948 \pm 0.0052 \text{ ft}$$

$$A = 0.00614 \pm 1.74 \cdot 10^{-5} \text{ ft}^2$$

$$D_h = 0.0559 \pm 0.0002 \text{ ft}$$

$$g = 32.17 \text{ ft/s}^2$$

Measurements made were $P(\text{gage})$, T , frequency (F) because the exit pressure was atmospheric pressure,

$$\Delta P = P(\text{gage})$$

Two pressure gages were used; their uncertainties were

$$\text{Low end gage} - W_p = 0.01 \text{ psi}$$

$$\text{High end gage} - W_p = 0.05 \text{ psi}$$

Water temperature uncertainty was estimated to be

$$W_T = 0.5^\circ\text{F}$$

Density was taken to be a function only of temperature; $\rho_W = \rho_W(T)$; over the tested range, the temperature uncertainty translates to a density uncertainty of

$$W_\rho = 0.01 \text{ lb/ft}^3$$

The turbine flow meter was calibrated in terms of gallons per minute vs. frequency. The given calibration uncertainty was

$$\begin{aligned} W_Q (Q < 30 \text{ GPM}) &= 0.5 \text{ GPM} = 0.00111 \text{ ft}^3/\text{s} \\ W_Q (Q > 30 \text{ GPM}) &= 0.2 \text{ GPM} = 0.00046 \text{ ft}^3/\text{s} \end{aligned}$$

The velocity V is $V = Q/A$

Then

$$W_V = \left[\left(\frac{\partial V}{\partial Q} \right)^2 W_Q^2 + \left(\frac{\partial V}{\partial A} \right)^2 W_A^2 \right]^{1/2} \quad (3)$$

$$\frac{\partial V}{\partial Q} = \frac{1}{A} \quad , \quad \frac{\partial V}{\partial A} = - \frac{Q}{A^2}$$

so

$$W_V = \left[\left(\frac{1}{A} \right)^2 W_Q^2 + \left(- \frac{Q}{A^2} \right)^2 W_A^2 \right]^{1/2} \quad (4)$$

define

$$R \triangleq \frac{\Delta P}{\frac{\rho V^2}{2g}} \quad ; \quad \text{then}$$

$$W_R = \left[\left(\frac{\partial R}{\partial P} \right)^2 W_P^2 + \left(\frac{\partial R}{\partial \rho} \right)^2 W_\rho^2 + \left(\frac{\partial R}{\partial V} \right)^2 W_V^2 \right]^{1/2} \quad (5)$$

$$W_R = \left[\left(\frac{2g}{\rho V^2} \right)^2 W_P^2 + \left(\frac{-2Pg}{\rho^2 V^2} \right)^2 W_\rho^2 + \left(\frac{-4Pg}{\rho V^3} \right)^2 W_V^2 \right]^{1/2} \quad (6)$$

define

$$\Delta L = L_2 - L_1 \quad ; \quad \text{then}$$

$$W_{\Delta L} = \left[\left(\frac{\partial \Delta L}{\partial L_1} \right)^2 W_{L_1}^2 + \left(\frac{\partial \Delta L}{\partial L_2} \right)^2 W_{L_2}^2 \right]^{1/2} \quad (7)$$

$$= \left[W_{L_1}^2 + W_{L_2}^2 \right]^{1/2} = 0.0074 \text{ ft} \quad (8)$$

define

$$\Delta R = R_2 - R_1 \quad ; \quad \text{then}$$

$$W_{\Delta R} = \left[\left(\frac{\partial \Delta R}{\partial R_2} \right)^2 W_{R_2}^2 + \left(\frac{\partial \Delta R}{\partial R_1} \right)^2 W_{R_1}^2 \right]^{1/2} \quad (9)$$

$$= \left(W_{R_2}^2 + W_{R_1}^2 \right)^{1/2} \quad (10)$$

Then

$$f = \frac{D_h}{\Delta L} \Delta R \quad (11)$$

So

$$W_f = \left[\left(\frac{\partial f}{\partial D_h} \right)^2 W_{D_h}^2 + \left(\frac{\partial f}{\partial \Delta L} \right)^2 W_{\Delta L}^2 + \left(\frac{\partial f}{\partial \Delta R} \right)^2 W_{\Delta R}^2 \right]^{1/2} \quad (12)$$

$$W_f = \left[\left(\frac{\Delta R}{\Delta L} \right)^2 W_{D_h}^2 + \left(\frac{D_h}{\Delta L} \right)^2 W_{\Delta R}^2 + \left(\frac{-D_h \Delta R}{\Delta L^2} \right)^2 W_{\Delta L}^2 \right]^{1/2} \quad (13)$$

Perform the analysis at 15, 50, and 80 GPM

(~20,000 , 90,000 , and 140,000 Reynolds number)

$$W_f(\text{Re} = 20,000) = 0.0027 \quad (\sim 10\% \text{ of } 0.027)$$

$$W_f(\text{Re} = 90,000) = 0.0008 \quad (\sim 5\% \text{ of } 0.0165)$$

$$W_f(\text{Re} = 140,000) = 0.0003 \quad (\sim 2\% \text{ of } 0.0155)$$

As in water test,

$$f = \frac{D_h}{\Delta L} \left[\left(\frac{\Delta P}{\frac{\rho V^2}{2g}} \right)_2 - \left(\frac{\Delta P}{\frac{\rho V^2}{2g}} \right)_1 \right] \quad (2)$$

$$D_h = 0.0559 \pm 0.0002 \text{ ft}$$

$$\Delta L = 4.948 \pm 0.0074 \text{ ft}$$

$$A = 0.00614 \pm 1.74 \cdot 10^{-5} \text{ ft}^2$$

$$g = 32.17 \text{ ft/s}^2$$

Flow rate:

$$Q(\text{ACFM}) = Q(\text{SCFM}) \left(\frac{14.7}{P_3} \right) \left(\frac{T}{530} \right)$$

where $Q(\text{SCFM}) = (8.0/106) \Delta P_{FM} \text{ (IN. H}_2\text{O)}$

$$W_{\Delta PFM} = 0.005 \text{ in. H}_2\text{O}$$

$$W_T = 0.1^\circ\text{F}$$

$$W_{P_3} = 0.10 \text{ psi}$$

$$W_Q = \left[\left(\frac{\partial Q}{\partial \Delta P_{FM}} \right)^2 W_{\Delta PFM}^2 + \left(\frac{\partial Q}{\partial P_3} \right)^2 W_{P_3}^2 + \left(\frac{\partial Q}{\partial T} \right)^2 W_T^2 \right]^{1/2} \quad (14)$$

Velocity $V = Q/A$; as before

$$W_V = \left[\left(\frac{1}{A} \right)^2 W_Q^2 + \left(-\frac{Q}{A^2} \right)^2 W_A^2 \right]^{1/2} \quad (4)$$

ΔP was measured on one of two manometers,

Low end manometer: $W_{\Delta P} = 0.05 \text{ in. H}_2\text{O}$

High end manometer: $W_{\Delta P} = 0.01 \text{ in. H}_g$

$\rho = \frac{P}{RT}$, $R = \text{gas constant for air}$

$$W_{\rho} = \left[\left(\frac{\partial \rho}{\partial P} \right)^2 W_P^2 + \left(\frac{\partial \rho}{\partial T} \right)^2 W_T^2 \right]^{1/2} \quad (15)$$

$$W_{\rho} = \left[\left(\frac{1}{RT} \right)^2 W_P^2 + \left(\frac{-P}{RT^2} \right)^2 W_T^2 \right]^{1/2} \quad (16)$$

Again define $R = \frac{\Delta P}{\frac{\rho V^2}{2g}}$, then

$$W_R = \left[\left(\frac{2g}{\rho V^2} \right)^2 W_{\Delta P}^2 + \left(\frac{-2\Delta P g}{\rho^2 V^2} \right)^2 W_{\rho}^2 + \left(\frac{-4\Delta P g}{\rho V^3} \right)^2 W_V^2 \right]^{1/2} \quad (6)$$

Similarly defining $\Delta L = L_2 - L_1$ and

$\Delta R = R_2 - R_1$ yields

$$f = \frac{D_h}{\Delta L} \Delta R \quad (11)$$

and

$$W_f = \left[\left(\frac{\Delta R}{\Delta L} \right)^2 W_{D_h}^2 + \left(\frac{D_h}{\Delta L} \right)^2 W_{\Delta R}^2 + \left(\frac{-D_h \Delta R}{\Delta L^2} \right)^2 W_{\Delta L}^2 \right]^{1/2} \quad (13)$$

Perform the analysis at 30,000
55,000 Reynolds Number
80,000

$$W_f(\text{Re} = 30,000) = 0.0010 \quad (\sim 4\% \text{ of } 0.025)$$

$$W_f(\text{Re} = 55,000) = 0.0006 \quad (\sim 2.9\% \text{ of } 0.021)$$

$$W_f(\text{Re} = 80,000) = 0.0005 \quad (\sim 2.8\% \text{ of } 0.018)$$

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