HEAT TRANSFER PERFORMANCE OF SHORT TWISTED TAPES*

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MASTER

ABSTRACT

The use of a short twisted tape extending over a portion of a tube's heated length to enhance the heat transfer to a gas flowing within the tube was investigated with emphasis on the vortex decay region downstream from the tape. This problem is of particular significance to the cooling of high-performance gas-cooled nuclear reactors. Heat-transfer and pressure-drop data were obtained using nitrogen gas to cool tubes heated by an electric current passing axially through the tube wall. Tape twists (measured as the ratio of the tube length per 180-degree tape twist to the tube diameter) varied from 2.38 to 8.05; Reynolds moduli ranged from 20,000 to 380,000; and wall-to-gas absolute temperature ratios from 1.0 to 2.1.

Heat-transfer results in the tape region and in the vortex decay region were correlated fairly well with empirical expressions; for fully developed swirl flow, Nu was from 35 to 85% greater than values obtained with straight flow at the same Re.

The decaying swirl flow exhibited considerably smaller friction factors than fully developed swirl flow, suggesting that an empty tube region following a twisted tape may provide heat-transfer enhancement at a relatively small pressure drop penalty.

* Research sponsored by the U. S. Atomic Energy Commission under contract with the Union Carbide Corporation.

In terms of heat-transfer coefficients evaluated at constant pumping power, the tubes containing twisted tapes performed up to 22% better than empty tubes.

INTRODUCTION

The performance of gas-cooled reactors is often limited by the heat transfer in the reactor core. A search for ways to modify the flow and/or the surface in order to enhance the heat transfer without an excessive increase in the pressure drop has been underway at the Oak Ridge National Laboratory (1) for several years — one of the more promising configurations has been twisted tapes inserted in smooth tubes (2). Most previous work has been concerned with tapes that extended for the entire length of the test section and, by comparison, relatively little is known about the heat-transfer improvement resulting from short twisted tapes where the swirl flow extends over a region considerably longer than the tape itself. Hence, this study (3) was undertaken to investigate the performance of such an arrangement in more detail.

The coolant for these tests was nitrogen gas at about 200 psig; Reynolds numbers reached 380,000, while the peak heat flux approached 150,000 Btu/hr·ft² at wall temperatures up to 1500°F. Under these conditions, the nitrogen tests simulated thermally and dynamically the heat transfer and flow of interest for advanced helium-cooled reactors that might operate in the pressure range from 1500 to 2000 psig with heat fluxes up to 500,000 Btu/hr·ft² at wall temperatures reaching 1500°F.

NOTICE

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EXPERIMENTAL APPARATUS

A schematic arrangement of the test equipment for once-through operation with nitrogen at 200 psig is shown in Figure 1. For clarity, not all pressure taps and thermocouples are pictured. The nitrogen gas was supplied to the test section at a pressure of about 200 psig and ambient temperature. The pressure drops between taps along the length of the various test sections were measured with a U-tube manometer. The inlet gas temperature was measured with a sheathed chromel-alumel thermocouple located in the inlet line. The outlet temperature was measured downstream of a mixing section to insure a representative mixed-mean temperature. The outer wall temperature of the test section and of the connected piping were measured with chromel-alumel thermocouples spot-welded to these surfaces. In the tape region, the thermocouples were positioned on the tube circumference at least 45-deg away from the nearest tape edge. A low-voltage electric current was supplied to the test section for resistance heating.

The test sections were made from Inconel 600 tubing, 0.675-in.-OD and 0.12-in. wall thickness. The twisted tapes were formed from Inconel 600 shimstock 0.015-in.-thick and slightly narrower than the tube inside diameter. After a tape had been inserted into a tube, the tube diameter was reduced about 0.003 in. by a drawing operation, producing light contact between the tape edges and the tube. The degree of contact between the tape and the tube was sufficient to hold the tape in place but, inferring from the local temperature distribution, not enough to conduct a significant amount of heat from the tube wall to the tape. The tubes were X-rayed to obtain accurate values of the tape twist ratios and also

Figure 1. Test Facility Schematic Diagram.

to assist in locating the pressure taps. In the tape region, the pressure taps were spaced at an integral number of tape twists and oriented with the hole axis at right angles to the tape surface. The dimensions of the several test sections and the locations of the pressure taps and thermocouples are shown in Figure 2.

HEAT-TRANSFER RESULTS AND DISCUSSION

Study of the local heat-transfer and pressure-drop variations along the test sections, necessitated about 60 measurements per run; a computer program was used to reduce both the heat-transfer and pressure-drop data.

For calculational purposes, the test sections were divided into 23 segments. The boundaries of the segments coincided with the locations of the thermocouples attached to the outside surface (Figure 2). The actual tube inside diameter rather than the hydraulic diameter was used, since this allowed a more direct comparison of the relative heat-transfer performance of tube regions with and without twisted tapes.

Runs with a heat balance discrepancy greater than 5% were discarded; normally the difference was less than 3%. The test section electrical and thermal conductivities used in the calculations were corrected for axial but not for radial temperature variations (4).

The properties of nitrogen were computed using two computer programs, TABHP and TABTP, taken from Reference 5.

Straight-Flow Heat Transfer

In the straight-flow entrance region, the local Nusselt modulus was maximum at the entrance to the heated length and diminished as the flow proceeded downstream. Fully developed straight flow was approached at

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TEST SECTION R-I, ID = 0.429 in., OD = 0.675 in.

• TEMPERATURE ELEMENTS

TEST SECTION R-II, ID = 0.421 in., OD = 0.658 in.

TEST SECTION T-I, Y = 2.38 in., ID = 0.421 in., OD = 0.658 in.

TEST SECTION T-II, Y = 4.42 in., ID = 0.421 in., OD = 0.658 in.

TEST SECTION T-III, Y = 8.05 in., ID = 0.4175 in., OD = 0.6555 in.

Figure 2. Test Sections.

about 50 diameters from the entrance; the Nusselt modulus for fully developed flow in empty tubes, can be approximated to within $\pm 5\%$ by:

$$Nu_{\infty} = 0.023 Re^{0.8} Pr^{0.4} (T_w/T_b)^{-0.5} \quad (1)$$

The -0.5 exponent on the temperature ratio is in agreement with the recommendations of Kays (6) for turbulent flow heat transfer to gas. Equation (1) is, of course, the Dittus-Boelter equation modified by a function of T_w/T_b .

Heat Transfer in Twisted Tape Region

As expected, the tape with the tightest twist ($Y = 2.38$) showed the greatest heat-transfer improvement. The local Nusselt modulus rose very rapidly as the flow entered the tape. The maximum was reached within about eight diameters, amounting to an enhancement of 70 to 87% above the Nusselt modulus calculated for straight flow. Constant slope of the T_w and Nu_x plots was observed beyond $X/D = 25$, suggesting fully developed flow.

For $Y = 4.42$, the relative increase of the Nusselt modulus in the tape region was less than for $Y = 2.38$. A local peaking of the heat-transfer coefficient, observed near the tape entrance for $Y = 2.38$, was absent at $Y = 4.42$. A typical Nusselt modulus profile in the tape region for $Y = 4.42$ is shown in Figure 3. Table 1 summarizes the heat-transfer results in the tape region for the three tape-twist ratios investigated.

In order to remove the effects of the tape ends and the tape length, the heat-transfer performance in the fully developed flow region was correlated. After several trials, the following relationship was found which represents the data to within $\pm 8\%$:

$$Nu_{\infty} = 0.023 Re^{0.8} Pr^{0.4} (T_w/T_b)^{-0.5} \left(1.105 + \frac{0.59 Re^{0.05}}{Y^{0.6}} \right) \quad ; \quad (2)$$

Figure 3. Typical Local Nu Profile for $Y = 4.42$.

Table 1. Summary of Heat Transfer Results in Tape Region

Y =	2.38			4.42			8.05		
Re x 10 ⁻⁴	2	15	30	2	15	30	2	15	30
Number of Diameters to Point of Maximum Nu _x	3	5	5	5	5	5	8	8	8
Number of Diameters to Establish Fully Developed Flow	3	26	26	5	5	5	8	8	8
Enhancement Predicted by Equation (2), Nu _∞ /Nu _{straight flow} (%)	73	81	84	50	56	58	35	39	40

the fluid properties are evaluated at the local bulk temperature. The final factor in Equation (2) represents the contribution of the twisted tape; for a tube containing a straight tape ($Y = \infty$), this reduces to the modified Dittus-Boelter equation. The results are shown in Figure 4 in terms of $Nu_{\infty} / [Pr^{0.4} (T_w/T_b)^{-0.5}]$; this representation eliminates data scatter caused by variations in gas temperature.

Comparison of the heat-transfer coefficients averaged over the length of the tape region agreed quite satisfactorily with Kidd's results (7), although for the tightest twist ratio slightly poorer performance was indicated by the present data.

Heat Transfer in Decaying Swirl Flow

The heat-transfer enhancement at positions downstream from the tape depended on the Reynolds modulus, the tape twist ratio, and the distance from the end of the tape. Increasing Re, or decreasing the tape pitch, improved the performance relative to that calculated for straight flow at the same Re.

Figure 4. Comparison of Nu_w Correlation with Experimental Results.

The local Nusselt moduli in the region of decaying swirl flow, beyond about one diameter from the tape, were correlated to within $\pm 10\%$ by the following expression, which was obtained after several trials:

$$\text{Nu}_x = 0.023 \text{ Re}^{0.8} \text{ Pr}^{0.4} (T_w/T_b)^{-0.5} \psi_1 \cdot \psi_2 ; \quad (3)$$

where,

$$\psi_1 = (0.7 + 0.000042 \text{ Re}) / (1.0 + 0.000039 \text{ Re}) \quad (4)$$

expresses the influence of the Reynolds modulus on the persistence of the heat-transfer enhancement and

$$\psi_2 = 1.0 + 1.05 / [Y + 0.0025 Y (Z/D)^2]^{0.6} , \quad (5)$$

the effect of the distance from the tape trailing edge on the persistence. When the product $\psi_1 \times \psi_2 = 1.0$, Equation (3) reduces to the modified Dittus-Boelter equation, Equation (1). Equation (3) is applicable only if the product $\psi_1 \times \psi_2$ is greater than one; in tube regions where $\psi_1 \times \psi_2 \leq 1.0$, straight flow has been essentially reestablished. Gas properties in Equation (3) were evaluated at the local bulk temperature, and Z is the distance from the trailing edge of the tape to the point under consideration. As in Equations (1) and (2), the effect of gas temperature variations is accounted for quite well by the ratio $(T_w/T_b)^{-0.5}$.

The agreement between the observed heat-transfer performance in the vortex flow decay region and Equation (3) is shown in Figure 5. The local heat-transfer performance is expressed in terms of $F_x(\text{Nu})$:

$$F_x(\text{Nu}) = \text{Nu}_x / [\text{Pr}^{0.4} (T_w/T_b)^{-0.5} \psi_1 \psi_2] ; \quad (6)$$

the solid line represents the predictions of Equation (3).

The improvement in Nusselt moduli in the vortex decay region, over the Nusselt moduli for straight flow, is summarized in Table 2. A comparison

Figure 5. Local Heat Transfer in Decaying Swirl Flow.

Table 2. Ratio of $Nu_x/Nu_{\text{straight flow}}$ in Vortex Decay Region Based on Correlating Equation (3)

Y =		2.38			4.42			8.05		
Re x 10 ⁻⁴		2	15	30	2	15	30	2	15	30
Distance beyond	5	1.39	1.64	1.68	1.22	1.45	1.48	1.12	1.32	1.35
tape trailing	20	1.22	1.44	1.48	1.11	1.31	1.34	1.04	1.22	1.25
edge in	40	1.07	1.27	1.30	1.01	1.19	1.22	1.00	1.14	1.17
diameters	60	1.00	1.18	1.21	1.00	1.13	1.16	1.00	1.08	1.11

of the observed effect with other experimental results was not possible, since to the writer's knowledge other heat-transfer data for the vortex decay region with gaseous, tape-induced, swirlflow are not available.

PRESSURE DROP RESULTS AND DISCUSSIONS

For the purpose of analyzing the pressure drop measurements, the test sections were divided into segments with boundaries defined (as shown in Figure 2) by the location of the pressure taps.

The observed pressure drops were corrected to remove the contributions from changes in axial flow velocity. Accelerations were due to changes in the gas density and in the channel flow area. In swirl flow, $\Delta P_{\text{friction}}$ reflects losses due to the change in angular momentum, friction, and pressure forces; thus:

$$\Delta P_{\text{friction}} = \Delta P_{\text{measured}} - \Delta P_{\text{density}} - \Delta P_{\text{flow area}} \quad (7)$$

The pressure drop due to gas density change was calculated by Equation (8) taken from Reference 8:

$$\Delta P_{\text{den.} + y} = \frac{G^2}{g_c} \left(\frac{1}{\rho_2} - \frac{1}{\rho_1} \right) \quad (8)$$

where the subscripts 1 and 2 refer to the entrance and exit, respectively, of the region being considered.

Slight changes in channel flow area were caused by welding shrinkage at the terminal lugs and by the presence of the twisted tape. The associated pressure drops were calculated as follows (Reference 8):

$$\Delta P_{\text{flow area}} = \frac{G^2}{2g_c \rho} [1 - (A_2/A_1)^2] \quad , \quad (9)$$

where A_1 and A_2 are the cross-sectional areas preceding and following the area change.

The Blasius friction factors were calculated from:

$$f_I = \frac{2g_c D}{\rho_I V^2 (X_2 - X_1)} \Delta P_{\text{friction}} \quad (10)$$

$$f_{II} = \frac{2g_c D}{\rho_{II} V^2 (X_2 - X_1)} \Delta P_{\text{friction}} \quad (11)$$

where f_I is based on fluid properties evaluated at the gas temperature T_I :

$$T_I = 0.5 (T_w - T_b) + T_b \quad (12)$$

and f_{II} at the temperature T_{II} :

$$T_{II} = 0.3 (T_w - T_b) + T_b \quad . \quad (13)$$

X is the axial position along the tube; and again, the subscripts 1 and 2 correspond to the entrance and exit of a segment. In each case, the velocity was taken as the average gas bulk velocity in the specific segment. The bases for the characteristic fluid temperatures, Equations (12) and (13), are discussed in the sections which follow.

Straight Flow

For straight flow good agreement was obtained between the measured local Blasius friction factors calculated from Equation (10) and values predicted by Equation (14) obtained from Reference 9:

$$f = 0.0056 + 0.5 \text{ Re}^{-0.32} \quad (14)$$

both plotted against a modified Reynolds modulus based on gas density and viscosity evaluated at temperature T_I and velocity determined at the gas bulk temperature. These same bases apply to the calculation of the friction factors. This procedure was recommended by Humble et al., Reference 10.

Swirl Flow

After trying several approaches, the overall friction factors for the tape region were computed from Equation (11) and plotted against the modified Reynolds modulus; results are shown in Figure 6. This scheme is identical to that used for straight flow, except that for swirl flow the characteristic fluid temperature is given by Equation (13). This latter equation differs from Equation (12) only in the coefficient 0.3; this value was preferred for swirl flow since it eliminated data scatter due to variations in gas temperature. The measured results were correlated to within $\pm 10\%$ by the following equation:

$$f_{II} = (1.0 + 5.15 Y^{-0.8}) f_{pt} \quad , \quad (15)$$

shown by the solid lines in Figure 6. The factor, f_{pt} , is the friction factor for the plain (empty) tube and is calculated from Equation (14) with gas density and viscosity evaluated at T_{II} and the velocity based on T_b . Equation (15) predicts the following increases over empty-tube friction

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Figure 6. Comparison of Correlated Swirl Flow Friction Factor with Experimental Results.

factors: 97% for $Y = 8.05$, 150% for $Y = 4.42$, and 258% for $Y = 2.38$; these percentages are independent of the Reynolds modulus. The pressure loss due to establishing the swirl flow is included in the friction factor.

The current results lie below Kidd's data (7); thus, at a Reynolds modulus of 100,000, the present friction factors are about 40% below values given in Reference 7 for $Y = 2.38$, 15% for $Y = 4.42$, and 5% for $Y = 8.05$. In part, these discrepancies may be due to the greater significance of entrance losses with the somewhat shorter tapes utilized by Kidd.

Agreement with isothermal friction coefficients reported by Thorsen and Landis (11) for air flowing through 1.0-in.-diam tubes with twisted tapes was fairly good. Friction factors obtained by these investigators generally showed the same dependence on Re as found in this study. However, for $Y = 2.38$ and 4.42, Equation (15) predicts swirl flow friction factors 10% higher than given in Reference 11 and, for $Y = 8.05$, 10% lower.

Decaying Swirl Flow

One effect observed for all the runs was the sudden reduction of the friction coefficient beyond the tape exit. The absence of the tape in this region probably brought this about in several ways. Firstly, the elimination of the axial component of the shear forces associated with the tape diminished the pressure drop. Secondly, the empty tube beyond the tape had a larger hydraulic diameter than within the tape region; thus, the effective Reynolds modulus increased while the wall shear stresses diminished. Finally, some recovery of the rotational energy from the fluid probably was achieved. Consequently, the friction factor averaged over the first ten diameters beyond the tape generally retained only one-third

of the increase above smooth tube values (as observed in the tape region). At 30 diameters, only about one-fourth of the increment persisted for $Y = 2.38$ and 4.42 ; while the pressure drop had returned to straight flow values beyond this axial position for a tape with $Y = 8.05$.

PERFORMANCE COMPARISON BETWEEN SWIRL-FLOW AND STRAIGHT-FLOW HEAT TRANSFER

One measure of the heat-transfer improvement afforded by a twisted tape is the comparison of the heat-transfer coefficient at a given pumping power for a tube containing a tape with that for an empty tube requiring the identical pumping power. For this comparison, it was assumed that the inside diameter, length, and the average fluid and wall temperatures were identical in the enhanced and in the empty tubes. The effectiveness, E , can then be written as:

$$E = \frac{\bar{h}_{\text{swirl flow}}}{\bar{h}_{\text{straight flow}}} \bigg|_{\text{pumping power}} \quad (16)$$

If E is greater than 1.0, swirl flow is obviously attractive. In Equation (16), $\bar{h}_{\text{swirl flow}}$ is the experimental average heat-transfer coefficient for the region of interest; for straight flow, the heat-transfer coefficient was obtained from Equation (1), the modified Dittus-Boelter equation. The pumping power requirements for swirl flow were based on the experimentally obtained pressure drops and included the tape entrance and exit losses. For the hypothetical empty tubes in the comparison, a flow rate was chosen that required the identical pumping power. The friction factors for the pressure drop in straight flow were determined from:

$$f_I = \frac{0.184}{Re_I^{0.2}} \quad (17)$$

adapted from Reference 9. The quantity E was evaluated for all experiments involving swirl-flow heat transfer, both for tube segments extending over the tape region only and for tube segments incorporating the tape region and 50 diameters of the adjacent vortex decay region.

These values, as well as the results obtained considering only the tape region, are summarized in Table 3.

Table 3. Summary of Swirl Flow Effectiveness

$Y =$	2.38			4.42			8.05		
$Re \times 10^{-4}$	2	15	30	2	15	30	2	15	30
E (tape only)	1.16	1.19	1.22	1.06	1.11	1.16	1.13	1.10	1.10
E (tape + 50 diam)	1.17	1.22	1.29	1.09	1.15	1.22	1.12	1.12	1.13

On the basis of constant pumping power, it appears therefore that the combination of a twisted tape followed by a swirl flow decay region may give an average heat-transfer coefficient ~30% higher than for straight flow.

The heat-transfer performance of the tape region for $Y = 2.38$ agrees well with E values at constant pumping power reported by Gambill and Bundy (12) for swirl flow heat transfer to air. However, for the more loosely twisted tapes at Reynolds moduli $> 20,000$, the current E values are about 10% larger. It is noted that for water, E values of 1.2 to 1.4 were reported for Y of about 2.5 in References 12 and 13.

CONCLUSIONS

The following main conclusions were drawn from the present study:

1. While the heat-transfer performance of the twisted tapes observed in these tests is generally consistent with previous results, the enhancement achievable in gaseous flows has been verified over a wider range of temperatures, heat fluxes, and Reynolds moduli than has been reported before.

2. The heat-transfer enhancement associated with a swirl flow increases sharply as the flow enters the tape region, suggesting that quite short tapes may provide significantly improved heat transfer. Local variations in heat transfer and pressure drop, as a function of axial position, were similar; this is consistent with the analogy between heat and momentum transport.

3. Gaseous swirl flow induced by a twisted tape persisted for a considerable distance beyond the tape for small Y and large Re . Since the pressure drop in this region diminished quickly as the flow progressed, it appears that a plain-tube region downstream from a twisted tape can extend the zone of enhanced heat transfer at a relatively small pressure drop penalty.

LIST OF SYMBOLS

A	Flow area normal to tube axis
D	Inside diameter
E	Effectiveness, $\bar{h}_{\text{swirl flow}}/\bar{h}_{\text{straight flow}}$, at constant pumping power
f	Blasius friction factor
f_I	Modified friction factor based on bulk velocity and properties evaluated at T_I
f_{II}	Modified swirl flow friction factor based on bulk velocity and properties evaluated at T_{II} , including losses due to both frictional and pressure forces
F_x	Non-dimensional function of local Nusselt modulus in vortex decay region
g_c	Gravitational constant
G	Mass velocity
$\bar{h}_{\text{swirl flow}}$	Average experimental heat-transfer coefficient in swirl flow
$\bar{h}_{\text{straight flow}}$	Heat-transfer coefficient given by modified Dittus-Boelter equation
k	Thermal conductivity
Nu_x	Local Nusselt modulus, evaluated at position X, hD/k
Nu_{∞}	Nusselt modulus in asymptotic fully-developed flow
Pr	Prandtl modulus
Re	Reynolds modulus, $\rho VD/\mu$
T_b	Bulk gas temperature, °R
T_w	Inside tube wall temperature, °R
T_I	Characteristic fluid temperature, $0.5 (T_w - T_b) + T_b$, °R
T_{II}	Characteristic fluid temperature, $0.3 (T_w - T_b) + T_b$, °R
V	Gas bulk velocity
W	Mass rate of flow

- X Axial position along length of tube
- Y Tape twist ratio, the number of tube diameters per 180-deg twist
- Z Distance beyond trailing edge of tape

Greek Symbols

- ΔP Pressure drop
- μ Fluid viscosity at T_b
- μ_I Fluid viscosity at T_I
- μ_{II} Fluid viscosity T_{II}
- ρ Bulk density at T_b
- ρ_I Bulk density at T_I
- ρ_{II} Bulk density at T_{II}

Subscripts

- 1 Entrance
- 2 Exit

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