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Transfer Coefficients for Free- and Forced-
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AuthorizedABSTRACT

The results of an experimental investigation of heat transfer to a high velocity, vortex air flow are presented. Heat-transfer coefficients have been observed for vortex flow in tubes which are larger than those for linear turbulent flow at a given flow power dissipation per unit surface area. Comparisons are drawn between air in free- and forced-vortex flow within tubes having length-to-diameter ratios of 11.7 and 20.0, respectively, and linear turbulent flow in identical tubes.

For the observed range of flow power dissipation, the free-vortex flow system is characterized by a heat-transfer coefficient one to two hundred per cent larger than that calculated for linear turbulent flow at the same pumping power. The forced-vortex flow system exhibits a superiority over linear flow only at high flow powers.

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NOMENCLATURE

A_1	internal heat-transfer area, ft^2
a^*	ratio of cross-sectional flow area of orifices in vortex generator to cross-sectional flow area of test section.
c_p	specific heat at constant pressure, $\text{Btu/lb.}^\circ\text{F}$
D_1	inside test-section diameter, ft
E	flow power dissipated across test section, hp
G	superficial axial mass velocity, $\text{lb/hr}\cdot\text{ft}^2$
h	internal heat-transfer coefficient, $\text{Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$
j_h	Colburn heat-transfer factor, $N_{St} \cdot N_{Pr}^{2/3}$
k	thermal conductivity of fluid, $\text{Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F/ft}$
L	heated test-section length, ft
ΔP	total pressure drop across vortex generator and test section, lb/in.^2
N_{Pr}	Prandtl modulus, $c_p \mu / k$
N_{Re}	Reynolds modulus, $D_1 G / \mu$
N_{St}	Stanton modulus, $h / c_p G$
q	rate of heat transfer, Btu/hr
r	radius, ft
R	internal test-section radius, ft
ΔT_f	film temperature drop, $^\circ\text{F}$
T_1	inlet air total temperature, $^\circ\text{C}$
V_t	tangential fluid velocity, ft/sec
W	weight rate of fluid flow, lb/hr

NOMENCLATURE (Continued)

Γ circulation constant for vortex flow, ft/sec
 Φ "work function", hp/ft² of internal area
 μ fluid viscosity, lb/hr·ft

Subscripts

V   vortex flow
L   linear flow

INTRODUCTION

One of the shortcomings associated with gas-cooled nuclear reactors is the large power expenditure required for coolant circulation. This can amount to as much as 20% of the total electrical power output of the reactor system. Accordingly, the development of a heat-removal system with minimum pumping requirements is of considerable interest. This, then, constituted one of the primary goals of the authors in investigating the heat-transfer properties of a vortex flow field.

Several interesting hydrodynamic features of vortex flow have indicated its possible application to increase gaseous heat transfer in tubes; namely, higher fluid velocities at the heat-transfer surface for a given gross flow rate, the segregation of the densest fluid layers in the vicinity of the heat-transfer surface, and the possible increase of radial heat transfer arising from the influence of the high centrifugal force field upon the heated fluid layers at the walls. A recent study¹ of boiling burnout heat fluxes in vortex flow with an incompressible fluid (water) has shown a larger ratio of heat transfer to pumping power than linear, forced circulation boiling. The results of this investigation indicate a radial transport mechanism involving the displacement of a less dense fluid (in this case a vapor) inward from the tube wall as the result of the centrifugal forces present in swirling flow. In a homogeneous gaseous medium, where the density differences are small, it was anticipated that high circulation rates would be required. The forced- and free-vortex heat-transfer experiments described in this report appear to have verified this presumption.

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¹Gambill, W. R., and Greene, N. D., "A Preliminary Study of Boiling Burnout Heat Fluxes for Water in Vortex Flow," ORNL-CF 58-4-56, April 12, 1958.

EXPERIMENTAL PROCEDURE

The experimental apparatus assembled for this study is shown in the schematic of Fig. 1. The system consisted of a small diameter test section with associated temperature and pressure measuring devices, a vortex generator, and a filtered air supply. Details of both the forced- and free-vortex test units are depicted in Fig. 2. The forced-vortex generator was purchased commercially while the free-vortex tube was fabricated locally. The air flow rates to the test sections were measured with a Brooks rotameter. Total temperatures were sensed by special thermocouples located at the vortex generator inlets and immediately following the test-section exit in an insulated mixing chamber. Wall temperatures were measured along the exterior, insulated surface of the tubular test section. In the free-vortex experiment the thermocouple output was measured with a Leed and Northrup precision potentiometer; the forced-vortex test section temperature data were recorded by a Honeywell potentiometer. In both experiments, the walls of the test sections were resistance heated by an alternating-current power source. Pressures were indicated by Bourdon-type pressure gages.

DISCUSSION AND RESULTS

Ideally, the motion of the fluid in a forced vortex can be described by solid-body mechanics; i.e., the tangential velocity is directly proportional to the radius,

$$v_t = \Gamma \left(\frac{r}{R} \right), \quad (1)$$

where Γ is a circulation constant describing the strength of the vortex. In a real system (as a Hilsch tube) the tangential velocities decrease rapidly

Fig. 1. The Experimental Apparatus.

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THE FORCED VORTEX GENERATOR

THE FREE VORTEX GENERATOR

Fig. 2. Vortex Test Sections.

with tube length due to friction losses at the tube wall. There is some evidence that within the first several tube diameters downstream from the entrance the peripheral velocities may be supersonic.² A free vortex, on the other hand, obeys the relation,

$$v_t = \Gamma / \left(\frac{r}{R} \right), \quad (2)$$

i.e., the tangential velocity is inversely proportional to the radius. Mathematically, this equation indicates an infinite velocity along the tube centerline. However, for a real gas, the relationship breaks down and it is found that the central core is in solid-body rotation.

An approximation to a free vortex may be created by introducing the fluid to the tube through numerous, small tangential inlets distributed along the length of the tube. With equal and constant pressure drop across each inlet, the incoming fluid cannot be displaced by fluid from an upstream inlet. Thus, in proceeding downstream, the fluid is forced to rotate through a progressively decreasing radius. Since the angular momentum must be conserved, the tangential velocity will increase as the fluid moves inward toward the tube center. The test section used for the free-vortex studies (Fig. 2) was made in this manner.

The results of this study are given in Figs. 3 through 6. The heat-transfer coefficients for forced-vortex and linear flow through tubes of identical geometry are compared in Fig. 3 in terms of a modified Colburn j-factor ($hN_{Pr}^{2/3}/c_p$) and the flow power dissipation. The coefficient was evaluated using the equation,³

$$\frac{hD_1}{k} = 0.0288 N_{Re}^{0.8} \left(\frac{D_1}{L} \right)^{0.05} \quad (3)$$

²Rohlik, H. E., and Wintucky, W. T., "Investigation of Semivaneless Turbine Stator Designed to Produce Axially Symmetrical Free-Vortex Flow," NACA-TN-3980, April, 1957.

³Croft, H., "Thermodynamics, Fluid Flow, and Heat Transfer," McGraw-Hill, New York, p 221, 1938.

Fig. 3. Inside Air Film Coefficients in a Tube for Linear and Forced Vortex Flow.

Fig. 4. Inside Air Film Coefficients in a Tube for Linear and Free Vortex Flow.

Fig. 5. Colburn j -Factors for Linear, Forced-Vortex, and Free-Vortex Heat Transfer in Tubes.

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Fig. 6. Ratio of the Friction Power per Unit of Area of a Heat Transfer Surface to the Friction Power per Unit of Area of a Tubular Surface at the Same Unit Film Conductance vs the Product of the Film Conductance and the Physical Property Factor.

which includes the effect of length-to-diameter ratio. The flow work was calculated using the sum of acceleration loss and friction drop, which was evaluated from standard friction-factor correlations. It is seen that forced-vortex flow is superior to linear flow only for relatively large power input. This result indicates the rapid decay of vorticity and field strength with tube length. The power dissipation value is based on the total head loss; i.e., the loss through both the vortex generator and the test section. The "work function" ϕ , defined as flow work per unit surface area, was calculated, for both linear and vortex flow, from the expression,

$$\phi = \frac{W \Delta P (T_1 + 273)}{20,600 A_1 (\Delta P + 14.7)} \left[\text{hp/ft}^2 \right] \quad (4)$$

In all cases, the heat-transfer coefficient was evaluated as

$$h = \frac{q}{A_1 \Delta T_f} \left[\text{Btu/hr} \cdot \text{ft}^2 \cdot ^\circ\text{F} \right] \quad (5)$$

where A_1 is the total heat-transfer surface area. The film temperature difference, ΔT_f , was taken as the difference between the average outside wall temperature and the average air total-temperature. It was found that the film temperature drop was of such a magnitude that the value of h was substantially independent of the wall-to-air stagnation temperature difference.⁴ This was verified in a series of experiments made with differing film temperature differences. The temperature drop across the test section wall was calculated and found to be negligible in the free-vortex (copper tube) studies. This difference was included in the forced-vortex (brass tube) analysis.

Fig. 4 presents similar results for free-vortex heat transfer. It is noted that the heat-transfer parameter for the vortex flow exceeds that for

⁴McAdams, W. H., et al., "Measurements of Recovery Factors and Coefficients of Heat Transfer in a Tube for Subsonic Flow of Air," NACA-TN-985, June, 1945.

the linear flow by a factor of 2 to 3. This may be explainable by the constancy of the vorticity with length in the free-vortex field. By virtue of the entrance configuration of the free-vortex system, it is estimated that the total fluid flow traverses only 50% of the total heat-transfer surface. On this basis, the heat-transfer coefficients for the free-vortex curve of Fig. 4 would be increased by a factor of 2.

The data of Figs. 3 and 4 are shown in a more conventional form (the Colburn j-factor) in Fig. 5. In this figure, the free-vortex curve (2) and line 1, and the forced-vortex results (4) and line 3 are comparable on the basis of equal length-to-diameter ratios. The inclusion of flow rate as a variable in the correlation brings about a distinct increment in superiority for both free- and forced-vortex flow over linear flow at a given flow power dissipation.

Experimental measurement of linear flow heat-transfer coefficients yielded values (not shown) somewhat smaller than those given by equation 3. In addition, nonboiling water in forced-vortex flow indicated a slight advantage (approximately 4%) for the vortex flow when compared with linear flow as in Figs. 3 and 4.

A further comparison of the heat transfer-flow work characteristics of air in vortex flow and in straight flow is given in Fig. 6, in which the ratio of flow work per unit surface area for vortex or linear flow to that for linear flow is plotted versus the group $\left(h N_{Pr}^{2/3} / c_p \right)$. This comparison, which is similar to one made by Kays and London for extended surfaces versus plain surfaces,⁵ perhaps more clearly shows the small flow-work required to obtain a given heat-transfer coefficient with free-vortex flow.

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⁵Kays, W. M., and London, A. L., "General Discussion Heat Transfer," The Amer. Soc. of Mech. Engrs., London, 1951.

CONCLUSIONS

From these preliminary experimental data, it would appear that a gaseous coolant in a strong centrifugal flow-field induces a greater transfer of heat at a given pumping power than in turbulent linear flow through similar geometries. This increased heat transfer in vortex flow is also characterized by a several-fold decrease in required fluid flow rates.

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