Effect of Entrance Shape on Flow Between Parallel Plates

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Whan and Rothfus (3) reported measurements of pressure drop and mean local fluid velocities for transition flow of water in a smooth rectangular duct of large aspect ratio. Their conduit was 0.70 in. high, 14 in. wide, and 20 ft. in horizontal length and was fitted with a bell-shaped entrance 3 ft. long. Velocity profiles were measured by means of a calibrated impact probe situated at the center line of the duct and 3 ft. from the downstream end. Bulk average linear velocities were calculated by integrating under the experimental profiles. Reynolds numbers and Fanning friction factors were expressed in the usual way, based on an equivalent diameter equal to four times the half-clearance between the broad sides of the duct, since infinitely wide parallel planes were approximated. No attempt was made at the time to establish the possible effect of entrance shape on the temporal mean behavior of the fluid.

Additional experiments have now been performed with the same apparatus, unchanged except for replacement of the bell-shaped entrance by a square-edged one. The isothermal flow of water at essentially room temperature has been investigated at seven Reynolds numbers between 3,270 and 9,590. Again both pressure drop and local velocities have been measured in the same manner as before.

Fanning friction factors and ratios of average to maximum velocities for the two types of entrances are compared in the following table. The data have been smoothed so that even values of the Reynolds number can be shown.

Reynolds number	Fanning friction factor		Average velocity ÷ maximum velocity	
	square	bell	square	bell
2,750	—	—	(0.667)	0.667
3,000	—	—	(0.740)	0.738
3,500	0.0094	0.0094	0.798	0.780
4,000	0.0093	0.0093	0.825	0.802
5,000	0.0091	0.0090	0.848	0.828
6,000	0.0090	0.0088	0.853	0.841
8,000	0.0088	0.0086	0.859	0.854
10,000	0.0086	0.0084	0.862	0.860

Velocity ratios, but not friction factors, could be extended smoothly to the critical Reynolds number of 2,750 reported by Whan, and the extrapolated values are shown in parentheses.

Installation of the square-edged entrance appears to have affected the friction factors very little over the investigated range. The ratios of average to maximum velocity on the other hand are at least measurably higher for the square-edged entrance than for the bell. The region of difference is limited however, since both entrances yield about the same values at the extremities of the reported Reynolds numbers. At a given average velocity an increase in the velocity ratio is of course associated with a flattening of the velocity profile over the main-stream portion of the fluid. A small change in the velocity ratio can therefore reflect considerable changes in the local velocity gradients.

Mache (2) has pointed out that in transition flow intermittent turbulent fluctuations appear with greater frequency for a sharp-edged entrance than for a bell-shaped one. Lindgren (*1*) has pictured the transition process in terms of random bursts of turbulence resulting from the collapse of certain entrance disturbances. The present data also suggest that the transient patterns associated with different entrance shapes influence ordinary temporal mean velocity measurements in transition flow at long distances from the entrance. Nothing more definitive can be said about the mechanism however, since no relationship between instantaneous velocities and the action of the impact probe was investigated in the present case. The principal points to be noted are that there does seem to be an effect of entrance shape on the velocity profiles in the transition range and that the velocity data of Whan and Rothfus are therefore limited, strictly speaking, to the case of the bell-shaped entrance.

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Velocity and Temperature Distributions About a Horizontal Cylinder in Free Convection Heat Transfer

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Some interesting results became apparent in the process of correlating velocity and temperature profile measurements taken about a single horizontal tube (*1*).

These measurements were taken at the plane of maximum diameter of a single, tapered, horizontal 1-in. diameter copper tube. Cooling water was passed through the tube while it was immersed in a volume-heated liquid.

All runs were made at the following single set of conditions:

- Bulk temperature, 75.0°F.
- Wall temperature, 63.8°F.
- Volume heat source, 0.10 B.t.u./cu.ft.-sec.
- Grashof number, 2.19×10^5

Velocity profiles were determined by measuring the motion of suspended 0.2 mm oil droplets (sp. gr. = 1.00), with an optical system which provided a 10x magnified image on a polar coordinate screen. The temperature field was obtained with a calibrated thermocouple probe, constructed from No. 36 chromel-alumel wire and supported

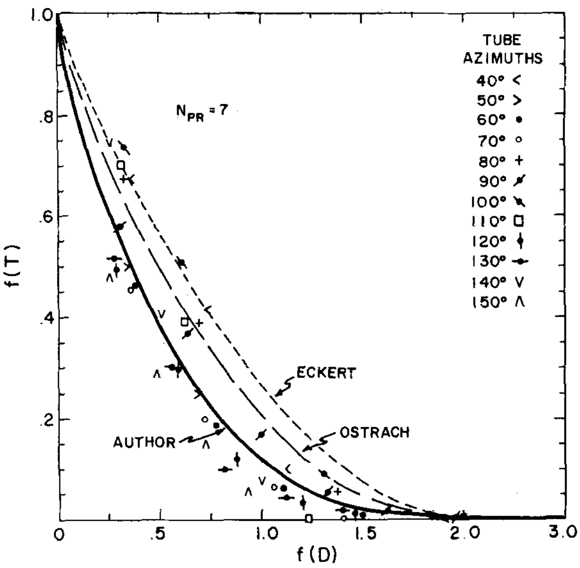

Fig. 1. Velocity profile comparison for water.

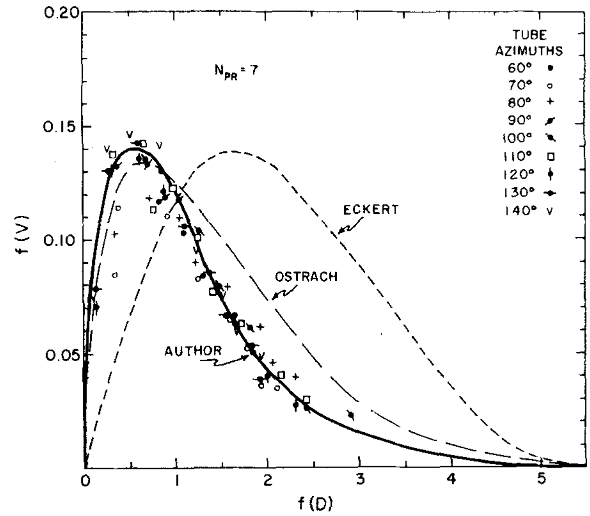

Fig. 2. Temperature profile comparison for water.

by a lever type of jig above the solution surface. Use of the optical system permitted exact positioning very close to the tube surface.

The experimental velocity profile is shown in Figure 1, with Hermann's (2) coordinates.

The dimensionless ordinate is

$$f(V) = \frac{u\,r}{\nu N_{Gr}^{1/2} f(x) g(x)}$$

The dimensionless abscissa is

$$f(D) = \frac{n}{r} N_{Gr}^{1/4} g(x)$$

Hermann's azimuth functions were used to represent the data, taken at the various tube azimuths shown, on a single dimensionless basis.

Ostrach (3) has expressed velocity and temperature distributions about a vertical plate on a dimensionless basis. His calculated vertical plate velocity profile is also shown in Figure 1, as obtained by interpolation for a fluid of Prandtl number 7. The comparison on a dimensionless basis is valid, since Hermann's and Ostrach's solutions of the appropriate boundary-layer equations are similar. In this case

$$f(V) = \frac{UX}{\nu 2 N_{Gr}^{1/2}}$$

and

$$f(D) = \left(\frac{N_{Gr}}{4}\right)^{1/4} \frac{Y}{X}$$

In a number of integral-method solutions of the boundary-layer equations, as mentioned by Eckert (3), the velocity profile is expressed as

$$u = u_x \frac{y}{\delta}\left(1 - \frac{y}{\delta}\right)^2$$

This function, calculated from the experimental maximum velocity and boundary-layer thickness, is also shown in Figure 1.

TEMPERATURE DISTRIBUTIONS

Both Hermann and Ostrach expressed a dimensionless temperature function of an appropriate similarity variable. In Figure 2 the experimental cylinder measurements are shown in dimensionless form, where

$$f(T) = \frac{t_\infty - t}{t_\infty - t_{\text{wall}}}$$

In the same figure Ostrach's vertical plate predictions and the usual temperature profile assumption as given by Eckert (4) are shown:

$$f(T) = \left(\frac{1 - y}{\delta}\right)^2$$

DISCUSSION

Although general conclusions are not justified, since measurements were made at only one set of conditions, some interesting indications are evident.

The measured cylinder velocity profiles, when expressed dimensionlessly, are in good agreement with comparable predictions for a vertical plate. Hermann's azimuth functions appear to represent correctly the velocity and temperature behavior about a horizontal tube.

The experimental behavior, when compared with Ostrach's nonvolume heated plate predictions, is in agreement with the analysis carried out by Randall and Sesonske (5).

The usual assumed velocity distribution for integral solutions, as mentioned by Eckert, is not in agreement with the experiment.

Although the temperature-profile data are considered less reliable than the velocity measurements, there is agreement between the cylinder measurements and the vertical-plate predictions.

Eckert's assumed temperature distributions are observed to differ little from the experiments. The thermal boundary-layer thickness is however not equal to that of the velocity boundary-layer but agrees with the relation

$$\delta_T = \frac{\delta_V}{N_{Pr}^{1/2}}$$

NOTATION

- $f(x)$ = azimuth function of Hermann (*1*)
- $g(x)$ = azimuth function of Hermann (*1*)
- N_{Gr} = Grashof number based on diameter
- n = normal distance from surface
- r = cylinder radius
- U = velocity component parallel to surface
- u = tangential velocity
- X = distance parallel to surface
- Y = distance normal to surface
- ν = kinematic viscosity

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