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ON THE THREE-DIMENSIONAL LAMINAR FLOW IN A TEE-JUNCTION

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NOMENCLATURE

d_i, D_m, hydraulic diameter of inlet (i) and main (m) duct;
l_i, length of inlet duct;
L_m, length of main duct;
M_i, momentum at start of inlet duct ($\equiv \rho d_i^2 \bar{V}_i^2$);
M_m, momentum at start of main duct ($\equiv \rho D_m^2 \bar{W}_m^2$);
p, local pressure;
r_s, Stokes radius ($\equiv \mu/\rho\tilde{u}$);
Re, Reynolds number;
u, v, w, Local velocities in x, y, z coordinate directions;
$\tilde{u}$, typical velocity in corner region;
$\bar{V}_i$, bulk velocity at start of inlet duct;
$\bar{W}_m$, bulk velocity at start of main duct;
x, y, z, coordinate directions.

Greek

ρ, density;
μ, molecular viscosity;
Δ, grid spacing.

1. INTRODUCTION

TEE-JUNCTIONS provide passage-ways for either dividing a single fluid stream, or combining two fluid streams. Numerical calculations for dividing laminar flow in tees have been performed in two-dimensions [1, 2]; extension of this type of calculation into the third dimension has yet to be performed. For the case of combining laminar flow, calculations in three dimensions have been performed, Simitovic [3]; however, Simitovic concentrated attention upon the flow in only the receiving (horizontal) duct of the tee: a hole in one side of the horizontal duct was used in lieu of the vertical inlet duct. The fluid entering and combining with the horizontal cross flow was specified to have a plug-shaped velocity profile across the surface of this hole. This practice is common in calculating discrete hole cooling processes; however, in tee-junctions, the diameter ratios d_i/D_m are often much larger (~ 1.0) so that the cross-flow should exert considerable influence upon the velocity distribution within the vertical inlet duct.

In this communication the laminar three-dimensional combining flow in all three-legs of a square cross-sectioned tee-junction is examined by reference to the effects of varying the vertical inlet velocity conditions. The results of the computations are compared to the experimental data of [3]. The effects of varying the inlet velocity profile are provided to ascertain whether the additional expense incurred by calculating the flow development inside the inlet duct is warranted.

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2. DESCRIPTION OF THE PROBLEM, THE GOVERNING EQUATIONS AND THE SOLUTION METHOD

The physical situation considered is shown in Fig. 1. The M's in the figure refer to the momentum influxes at the entrance to the horizontal or main (m) and inlet (i) ducts. Here, $M_i/M_m = 0.25$ and $d_i/D_m = 0.5$.

The equations governing the steady incompressible laminar flow in tee-junctions of square cross-section are put into their finite-difference form using the method of Patankar and Spalding [4]. The equations are then solved using the SIMPLE algorithm; details of which can also be found in [4]. The procedure that is used to solve the equations (that embodies SIMPLE) is called FLIRT; details of this procedure, together with other information regarding the equations and their solutions can be found in [5].

3. COMPUTATIONAL REQUIREMENTS

The geometrical situation shown in Fig. 1 is mapped using orthogonal intersecting grid lines in the manner shown in Fig. 2. The grid is non-uniform in all three coordinate directions, with particular attention being paid to providing large grid densities in all corner regions. The grid density was varied to achieve grid independent results; up to 28 000 grid points were provided for the finest grid here used. An example of the results for a grid independent test is shown in Fig. 3. Further comments regarding the grid will be made shortly. The core requirement of the program was 0.5 M-byte; to achieve a converged solution (normalized residuals <0.005) required on average, 200 iterations. The computer time required will be presented in the next section.

4. PRESENTATION AND DISCUSSION OF RESULTS

The results of some of the calculations that have been performed [5] will now be presented and discussed. Fig. 4 provides plots of axial velocity with distance at, and downstream of, the centre-line of the inlet duct; these plots represent velocities located at the midplane of the horizontal duct ($y/D_m = 0.5$). The four cases that correspond to the

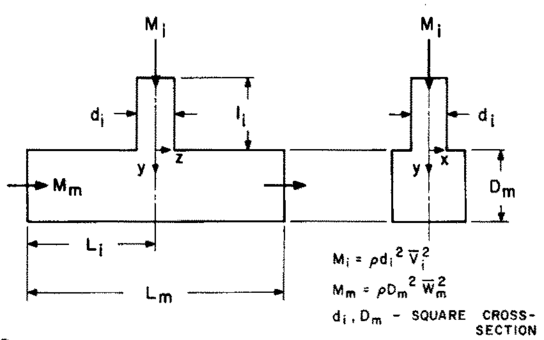

FIG. 1. The general physical situation considered.

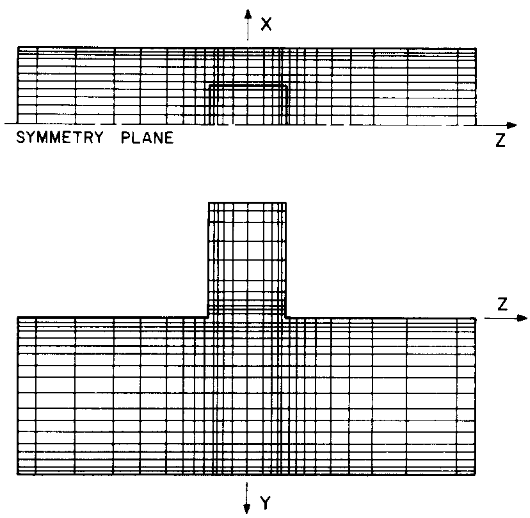

FIG. 2. Finite-difference grid distribution.

results of the calculations refer to those geometrical factors and inlet conditions found in Table 1. The points in the figure are the data of Simitovic [3]. It is apparent from the figure that the predictions are at variance with these data, particularly for case 1; of the other cases, case 4 provides the best overall agreement with these data. Further discussion of the results will be divided into two sections: Experimental, and Numerical.

Table 1. Geometrical factors and inlet conditions for $d_i/D_m = 0.5$, $M_i/M_m = 0.25$

Case no.	Inlet velocity distribution: Main duct	Inlet velocity distribution: Inlet duct	l_i/d_i
1	Plug	Plug	0.0
2	Plug	Plug	2.2
3	Plug	Fully developed	2.2
4	Experiment [3]	Fully developed	2.2

$L_i = 4.5D_m$ $L_m = 9.5D_m$ $Re_{D_m} = 100$

4.1. *Experimental*

The data of Simitovic were obtained using laser doppler anemometry; there is no reason to believe that these data are of poor quality; the experimental set-up, however, appears to have induced asymmetries. This is particularly noticeable in diagrams (a)–(c) of Fig. 4. Moreover, in determining the experimental flow rates, Simitovic reports a maximum error of 5.5%; an examination of the predictions for cases 3 and 4 indicate that they are at most about 5% higher than these data.

4.2. *Numerical*

Of the predictions that are here presented, case 1 of Table 1 duplicates those calculations made by Simitovic. A comparison with these predictions is not made because Simitovic's predictions appear to be not grid independent: at the duct centre line and for $Z/d_i \geq 1.0$, $W/\bar{W}_m$ remains constant, while the data shows an increase in this velocity ratio.

In the literature, concern has been exhibited about grid refinement in those regions where fluid flows around a corner. In [1], the grid spacing Δ, had no effect upon the flow downstream of the corner when the spacing equalled the Stokes radius, r_s. Castro [6] concurred with [1], but noted that Δ/r_s could be "rather greater" than 1 for turbulent flow. Here, it is estimated that, for the finest grid, $\Delta/r_s = 2$.

Turning attention to the computer time requirements, case 1 and cases 2–4 of Table 1 required ~50 min and ~60 min, respectively. These times refer to an IBM 360–195 machine. The 20% increase in time required for calculations to be performed in the inlet duct may be justified for the momentum and diameter ratio here presented; however, as shown in [5], for low momentum ratios ($M_i/M_m \sim 0.1$) and larger diameter ratios ($d_i/D_m \sim 1.0$), the effects of the cross-flow can exert much influence upon the velocity distribution inside the inlet duct.

In view of the above comments, and in particular those concerning the experimental data, a firm conclusion regarding the quantitative accuracy of the predictions cannot be made; however, what is certain is that the use of an inlet duct provides better qualitative agreement with these data than those calculations obtained without an inlet duct.

5. CONCLUSIONS

From what has been presented here, the following conclusions may be drawn:

(a) A numerical procedure is available which can cope

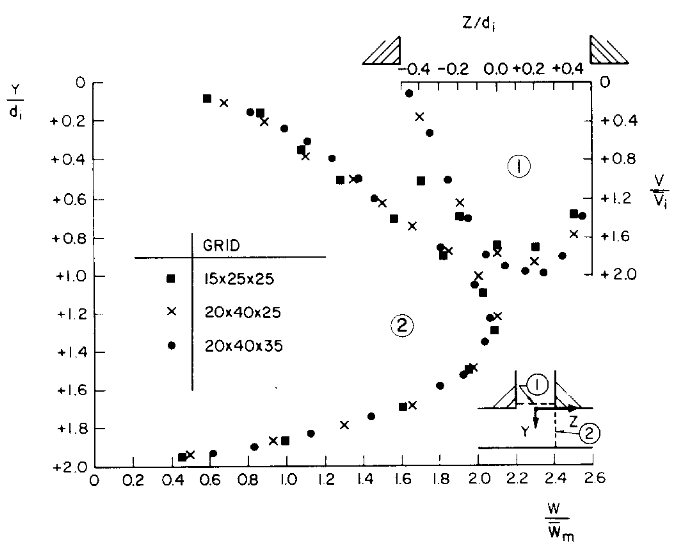

FIG. 3. Sample grid independence test.

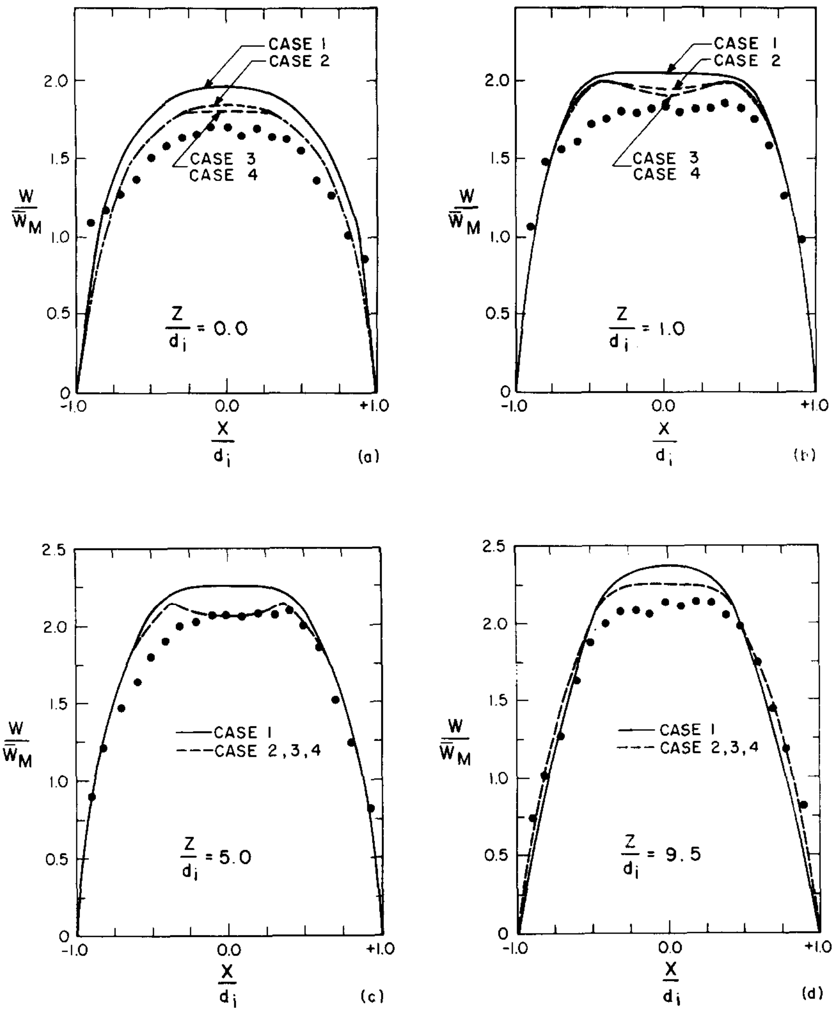

FIG. 4. Comparison of experimental data and predictions.

with the complex flow in all legs of tee-junctions of square cross-sections.

(b) When compared to the available experimental data, agreement is best when the calculations take into account all legs of a tee-junction.

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