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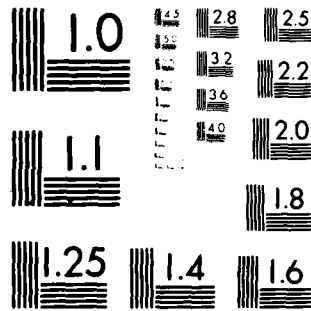
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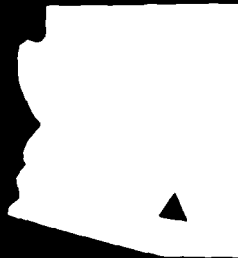
M. A. Habib and D. M. McEligot

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Calculations of the flow and heat transfer of a swirling turbulent flow after a sudden pipe expansion are reported. The calculations were obtained by the numerical solution of the time-averaged forms of the continuity, momentum and thermal energy equations together with equations for the turbulence kinetic energy and its rate of dissipation. For a sudden expansion without swirl, the predictions produced satisfactory agreement with available data for the Nusselt number. In the swirling case, there were no heat transfer measurements available for comparison. However, existing measurements of a swirl flow at an expansion without heat transfer were used to test the validity of the flow field calculations.

The predicted effects of varying the swirl number from zero to 1.0 on the heat transfer behavior are presented for a range of Prandtl numbers from 0.7 to 10. The expected effects of the swirl on the velocity and temperature fields are also reported. The results predict, for example, that the maximum Nusselt number is increased as the swirl number increases and its position moves towards the inlet section. They also show an increase in the Nusselt number as the Prandtl number increases.

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Sixth Annual Summary Report

CONVECTIVE HEAT TRANSFER FOR SHIP PROPULSION

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CONVECTIVE HEAT TRANSFER FOR SHIP PROPULSION

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Abstract

Calculations of the flow and heat transfer of a swirling turbulent flow after a sudden pipe expansion are reported. The calculations were obtained by the numerical solution of the time-averaged forms of the continuity, momentum and thermal energy equations together with equations for the turbulence kinetic energy and its rate of dissipation. For a sudden expansion without swirl, the predictions produced satisfactory agreement with available data for the Nusselt number. In the swirling case, there were no heat transfer measurements available for comparison. However, existing measurements of a swirl flow at an expansion without heat transfer were used to test the validity of the flow field calculations.

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NOMENCLATURE

Symbol	Definition
C_R	Constant in turbulence model, for correction for stream-line curvature, equation (11)
D	Diameter of pipe downstream of the expansion = $2R_o$
d	Diameter of the pipe upstream of the expansion = $2r$
G	Generation of turbulence kinetic energy K
H	Step height = $(D - d)/2$
H	Enthalpy
K	Turbulence kinetic energy = $\frac{1}{2} \overline{u_i u_i}$
Nu	Nusselt number
Nu_{fd}	Nusselt number of fully developed pipe flow
Pr_l	Molecular Prandtl number
Pr_t	Turbulent Prandtl number
q_w	Wall heat flux
R	Radius of downstream pipe
r	Radial coordinate
Re	Reynolds number; Re_d , based on diameter upstream; Re_D , based on diameter downstream
R_f	Flux Richardson number
S	Swirl number, $\int_0^{r_o} \overline{U} \overline{W} r^2 dr / r_o \int_0^{r_o} \overline{U}^2 r dr$
T_b	Bulk temperature, $\frac{\int \overline{U} T r dr}{\int \overline{U} r dr}$
T_{in}	Inlet temperature

NOMENCLATURE--Continued

Symbol	Definition
T_w	Wall temperature
$\bar{U}$	Axial mean Velocity
$\bar{U}_{av}$	Bulk velocity = $\int \rho \bar{U} r dr / \int \rho r dr$
u	Fluctuations of axial velocity
$\bar{V}$	Radial mean velocity
v	Fluctuations of radial velocity
$\bar{W}$	Tangential mean velocity
w	Fluctuations of tangential velocity
x	Axial coordinate
X_o	Axial distance downstream from the expansion
y	Transverse coordinate measured from the wall
ϵ	Rate of dissipation of turbulence kinetic energy
κ	von Karman constant for turbulence model, equation (10)
τ_w	Wall shear stress in the axial direction, equation (9)

Subscripts

o	Wall
p	Node point nearest the wall

CHAPTER I

INTRODUCTION

1.1 Background

Current Naval propulsion plants are powered by variations of the Rankine cycle (steam) or the open gas turbine cycle (air and combustion products) plus some diesel engines in small ships. Alternative power systems suggested include the closed gas turbine cycle and cycles involving dissociation of the working fluid in either a Rankine or a gas cycle. These latter two are believed to offer the potential of substantial improvement in the power-to-weight ratio of the propulsion plant. The present and proposed studies consider basic problems in convective heat transfer and flow friction that are important in all of the above.

Convective heat transfer provides the dominant thermal resistance in several components of conventional steam power plants as well as in all heat transfer components in gaseous cycles. For example, heat transfer from the condensing steam to the cooling water in the main condenser of a Naval ship is via the condensation heat transfer coefficient which is high, then conduction conductance through thin tubing of high conductivity, and finally via the moderate heat transfer coefficient of the cooling water flowing through smooth circular tubes. Thus, the overall thermal resistance is dominated by the convective thermal resistance of the cooling water inside the tubes. One can expect significant reductions in tube

length and, therefore, size and weight of the condenser and overall plant if the convective heat transfer coefficient of the cooling water is increased appreciably.

A common geometry recurring in compact Naval propulsion plants is a change in duct size in the primary fluid loop. As a consequence of the upstream plumbing, the fluid is often swirling about the axis in the piping. Heat losses from the primary fluid and thermal stresses in the component depend on the convective heat transfer from the fluid to the component as it undergoes this geometrical transition. The idealized problem is a study of heat transfer in a sudden expansion with swirl flow, the subject of the present report. The same situation also occurs at the entrance to some heat exchanger tubes.

The flow downstream of a sudden expansion consists primarily of a central jet, with or without rotation due to swirl, surrounded by a recirculating donut-shaped vortex induced by the separation as the jet leaves the central tube. This jet gradually spreads as the flow proceeds downstream until it reattaches to the downstream wall and fills the larger tube. Eventually a fully developed velocity profile can be expected to be established if the tube length is sufficiently long. The centrifugal body forces induced by swirl modify the recirculating vortex, the development of the jet and the reattachment location.

Significant enhancement of heat transfer rates generally occurs when fluids are transported through sudden enlargements in pipes (Fig. 1) or duct cross-sections or when they are subjected to angular tangential

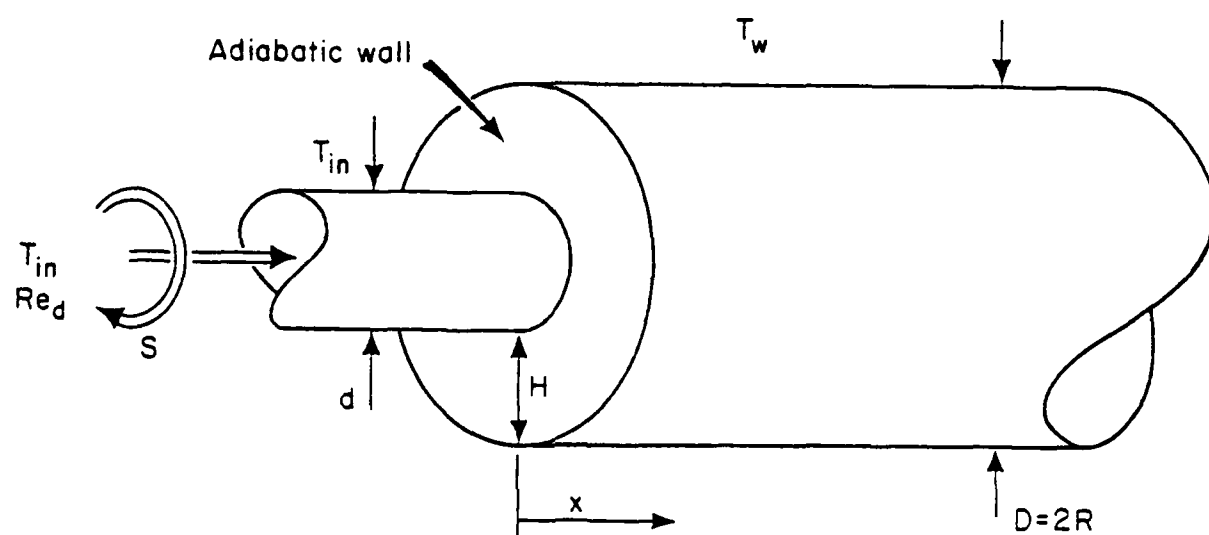


Figure 1. Idealized geometry.

momentum due to twisted tapes or the application of vane-generated swirl. In both cases, the levels of turbulence kinetic energy are increased and the heat transfer rate generally increases as well. High shearing rates created by both separation and swirling effects are associated with high generation rates for the kinetic energy of turbulence; in addition the rate of dissipation of turbulence kinetic energy decreases due to an increase in the length scales. The two effects lead to elevated magnitudes of the kinetic energy of turbulence. All these effects are expected to reduce the thickness of the viscous sublayer through which heat must pass largely by molecular diffusion and so will increase the heat transfer rate.

1.2 Previous work

The problem of heat transfer to separated and reattached subsonic turbulent flows downstream of abrupt pipe expansions has been studied experimentally by a number of investigators such as Ede, Hislop and Morris [1956], Krall and Sparrow [1966] and Zemanick and Dougall [1970]. Reviews of the heat transfer literature for separated and reattached flows are reported by Eckert et al. [1971] and by Fletcher, Briggs and Page [1974]. The experimental data of Krall and Sparrow [1965] for water, and of Zemanick and Dougall [1970] for air are for typical geometries representing axisymmetric abrupt enlargement of tubes. Both teams found a maximum in the local heat transfer coefficient at the assumed point of flow reattachment. Adequate descriptions of wall heat transfer at solid boundaries in regions of separated and

reattached flows have been limited by the lack of a complete solution for the fluid flow field. Spalding [1967] presented a power law relation between the Stanton and the Reynolds numbers for heat transfer from a wall adjacent to a region of separated flow; he employed a one-dimensional model of the flow near a wall. Recently, Chieng and Launder [1980], in calculations of the turbulent heat transfer downstream of an abrupt pipe expansion, found that the Nusselt number is overpredicted by 20% in flows with high Reynolds numbers and that its maximum value occurs about one step height upstream of the calculated reattachment point, relative to the data of Zemanick and Dougall.

For swirling flow, a number of studies have been done to investigate the effect of swirl on heat transfer parameters. Experiments performed by Thomas [Eckert et al., 1971], with swirling air flow in a tube indicated that, for turbulent flow and Reynolds numbers less than 10^5 , the swirl flow transfers heat more efficiently than a non-swirling flow. Zaherdadeh and Jagadish [1975] concluded that an augmentation of up to 80% was obtained, with a constant wall temperature boundary condition, with the use of tangential vane type swirl generators.

1.3 Present work

Calculations of the effects of separation and swirl on heat transfer parameters have rarely been reported and the combined effects of both apparently have not been studied. Thus, we attempt the prediction of such cases in contrast to the "postdiction" usually employed in numerical studies. In the present paper, a calculation method is provided and

is applied for the prediction of flow and temperature fields in the swirling, separated and reattached flow in the vicinity of a sudden expansion of a pipe.

An idealized version of the geometry is shown as Figure 1. The expansion ratio is 2 and the Reynolds number upstream of the expansion is taken as 5×10^4 for the calculations. The effects of swirl on the heat transfer parameters are studied in the range of swirl number from 0.0 to 1.0 and of Prandtl number from 0.7 to 10. For confidence testing, the calculations are compared with the measurements of Zemanick and Dougall [1970] for zero swirl number and those with swirl, but without heat transfer, are compared to the data of Beltagui and MacCallum [1976].

The governing equations and boundary conditions are described in the following section which is ended by a description of the solution procedure. The results are then presented and discussed. The paper ends with a summary of conclusions.

CHAPTER 2

ANALYSIS

Equations representing conservation of mass, energy, turbulence kinetic energy and its dissipation rate plus the momentum equations were solved in axisymmetric coordinates with an extension of the TEACH computer program developed originally by Gosman and Pun [1974]. In the present study, the version of Habib and Whitelaw [1980], which treats double coaxial jet flows, was modified to give predictions of swirl flow downstream of an abrupt pipe expansion and to yield the temperature field and heat transfer parameters by adding the solution of the thermal energy equation.

The equations modeled and the turbulence model are as follows:
continuity,

$$\frac{\partial \bar{U}_j}{\partial x_j} = 0 \quad (1)$$

momentum,

$$\frac{\partial}{\partial x_j} (\rho \bar{U}_i \bar{U}_j + \rho \overline{u_i u_j}) = - \frac{\partial P}{\partial x_i} \quad (2)$$

thermal energy,

$$\frac{\partial}{\partial x_j} (\rho \bar{U}_j H) = \frac{\partial}{\partial x_j} \left(\frac{\mu_{eff}}{\sigma_H} \frac{\partial H}{\partial x_j} \right) \quad (3)$$

turbulence kinetic energy,

$$\frac{\partial}{\partial x_j} (\rho \bar{U}_j k) = \frac{\partial}{\partial x_j} \left(\frac{\mu_{eff}}{\sigma_k} \frac{\partial k}{\partial x_j} \right) + G - \rho \epsilon \quad (4)$$

and dissipation of turbulence kinetic energy

$$\frac{\partial}{\partial x_j} (\rho \bar{U}_j \epsilon) = \frac{\partial}{\partial x_j} \left(\frac{\mu_{eff}}{\sigma_\epsilon} \frac{\partial \epsilon}{\partial x_j} \right) + C_1 \frac{\epsilon}{k} G - C_2 \frac{\epsilon^2}{k} \quad (5)$$

$$\text{with } \mu_{eff} = \mu_l + \frac{c_\mu \rho k^2}{\epsilon} \quad (6)$$

$$G = -\rho \overline{u_i u_j} \frac{\partial \bar{U}_i}{\partial x_j}, \quad k = \overline{u_i u_i} / 2$$

and

$$-\rho \overline{u_i u_j} = \mu_{eff} \left(\frac{\partial \bar{U}_i}{\partial x_j} + \frac{\partial \bar{U}_j}{\partial x_i} \right) - \frac{2}{3} \rho k \delta_{ij}$$

$$\text{where } \delta_{ij} = 0 \text{ if } i \neq j \text{ and } \delta_{ij} = 1 \text{ if } i = j. \quad (7)$$

Fluid properties were idealized as constant except in some cases with air where the density was allowed to vary due to pressure variation.

In addition to the set of equations listed above, the following inlet and boundary conditions are considered. In the case of non-swirling flows, data for fully developed pipe flow were taken as initial values for mean velocity and turbulence kinetic energy. The inlet dissipation rate was then determined from its model,

$$\epsilon = c_u \frac{K^{3/2}}{\ell} \quad (8)$$

where ℓ is the length scale that characterizes the energy containing eddies, taken as $\ell = 0.03 r_o$.

For swirl, the initial axial and tangential mean velocities were interpolated from the data of Habib [1980]. The turbulence kinetic

energy was taken as a function of the total mean kinetic energy and the dissipation rate was determined via equation (8) with λ assumed to be given as $r_0 (0.03 + 0.07S)$.

All axial gradients, $\partial/\partial x$, were presumed zero at the exit plane of the tube well downstream. Axial symmetry was specified so $\bar{V} = 0$ and the radial gradients of all other quantities were zero at the centerline. No slip and impermeable wall conditions were specified on all solid surfaces.

Wall functions [Launder and Spalding, 1972] corresponding to the dependent variables were taken as

$$\tau_w = \frac{\mu_\ell \bar{U}_p}{y_p} \frac{y_p^+}{U_p^+} \quad (9)$$

where $U_p^+ = \frac{1}{\kappa} \ln Ey_p^+$ and $y_p^+ = C_\mu^{1/4} K_p^{1/2} y_p / \nu_\ell$

$$\text{and} \quad q_w = \frac{\mu_\ell (H_w - H_p)}{y_p} \frac{y_p^+}{U_p^+ + P_f} \quad (10)$$

$$\text{where } P_f = C_f \left(\frac{Pr_{,\ell}}{Pr_{,t}} - 1 \right) \left(\frac{Pr_{,t}}{Pr_{,\ell}} \right)^{1/4}$$

The value of the kinetic energy of turbulence near the wall, K_p , is calculated from the transport equation for K with the flux of energy to the solid wall set to zero. The corresponding value of ϵ was calculated from equation (8) with $\lambda = C_\mu^{1/4} \kappa y_p$.

The constants used in the above equations are in accord with those of Khalil and Whitelaw [1974] and Pope and Whitelaw [1976]. They are summarized as follows:

$$\begin{aligned} C_\mu &= 0.09, & \sigma_K &= 1.0, & \sigma_\epsilon &= 1.22, & \sigma_H &= 0.9, & C_1 &= 1.45, \\ C_2 &= 1.9, & C_f &= 9.24, & \kappa &= 0.42, & E &= 9.8 \end{aligned} \quad (10)$$

Modification of C_1 to account for streamline curvature is described in Section 3.2 later.

The solution procedure described by Patankar and Spalding [1972] was used to solve the above equations. From trial values of the pressure distribution, the momentum equations can be solved to obtain an estimate of the velocity field at the first iteration. However, there is no guarantee that the resultant velocity field will satisfy the continuity equation, therefore, after each calculation over the solution domain, adjustments are made to the pressure and velocity fields to satisfy continuity. The procedure is repeated until convergence is obtained. The energy equation is then solved and the temperature field is presented.

Most of the computations were done with a grid of 25 x 18 nodes with more points concentrated near the walls and regions of separation and high velocity gradients. The computer time (per iteration per grid node) using the CDC CYBER 175 was 9.5×10^{-4} Sec to solve the velocity field and 1.2×10^{-4} Sec to obtain the temperature field. (These time estimates typically corresponded to \$24/run and \$6/run, respectively, at University rates). The storage requirements of the program were 37,000 octal words.

A grid independence test was conducted during the comparisons to the data of Zemanick and Dougall [1970]. It showed that an increase in the number of nodes from 20×17 to 27×21 affects the predicted Nusselt number only in the region of its maximum (see Figure 2) and then less than three percent. The effect was negligible in the downstream region. For most of the predictions presented in the next section a grid of 25×18 nodes was employed so they are expected to have better numerical accuracy than the former case.

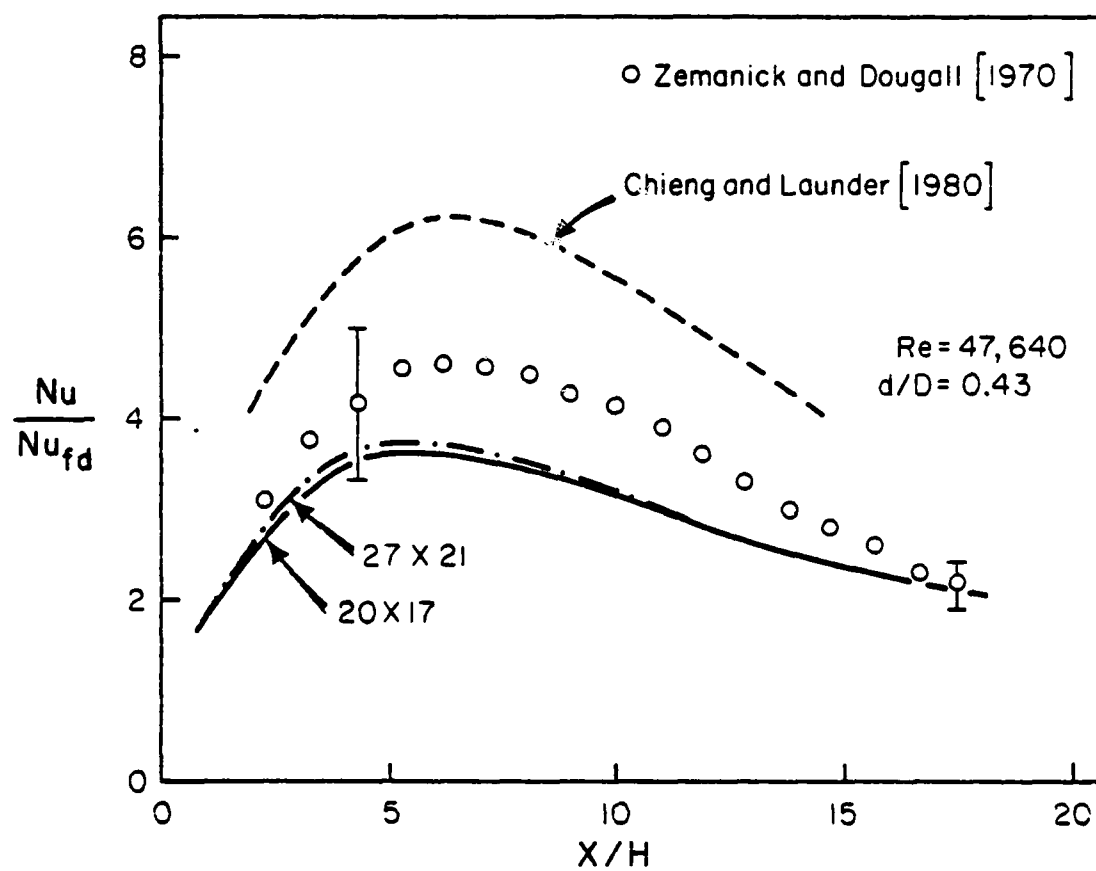


Figure 2. Comparison for heat transfer at a sudden expansion without swirl.

CHAPTER 3

PREDICTIONS

In this section results from the computer program are first compared to the available data to test its validity. Then, the predictions of the separated flow with both swirl and heat transfer are presented.

3.1 Zemanick and Dougall - heat transfer without swirl

The results are first compared with the experiment of Zemanick and Dougall [1970] for zero swirl. Since inlet velocity profiles were not presented, the initial conditions were taken as those of fully developed pipe flow in the smaller tube. In the run chosen for comparison, the expansion ratio was 0.43 and the Reynolds number calculated downstream from the expansion was 47,800. In the experiment resistive heating was employed so the appropriate thermal boundary condition would be a specified wall heat flux. To correspond with the procedure of the present program, the wall temperature distribution which was measured was entered as the thermal boundary condition and the wall heat flux was calculated via equation (10).

As noted by Zemanick and Dougall [1970] there were some difficulties with their experiment so it is not ideal for a close comparison, but it is one of the few with data readily available. Axial conduction in the tube was significant and the thickness of the copper electrode at the expansion extended about a step height in the downstream direction. In addition it may be noticed that their correlation of local

Nusselt numbers in the fully developed region downstream is about 15% higher than commonly accepted values [McEligot, Magee and Leppert, 1965]. Unfortunately, quantitative estimates of the experimental uncertainties [Kline and McClintock, 1953] were not presented. If one assumes an uncertainty of 1°C in measuring the wall temperature, typical of premium grade thermocouple wire, it can be shown that this effect alone would lead to an uncertainty of about twenty percent in the Nusselt number near the entry and approximately five percent downstream; these estimates are shown by brackets on Figure 2. Thus, it is best to restrict the comparison to trends rather than absolute magnitudes.

Figure 2 shows a comparison between the calculated and measured distributions of Nusselt numbers; the curves represent the numerical predictions. It is shown that the agreement is generally good in the upstream and downstream locations with the present calculations underpredicting the maximum Nusselt number by about 20%. The location of the maximum Nusselt number is found to be one step height upstream from the corresponding peak in the experiments. By examination of the predicted velocity distributions it may also be found that the calculated maximum Nusselt number occurs two step heights upstream from the calculated reattachment point. This direction was also indicated by Chieng and Launder [1980]; however, they indicate the difference in location to be one step height only.

The discrepancy between the calculations and the experiment may be attributed partly to the numerical accuracy in the calculation

procedure. Increasing the number of grid points to reduce effects of the numerical diffusion leads to a slight increase in Nusselt number in the entry region as shown on Figure 2. However, as noted earlier, the increase in the number of nodes from 20×17 to 27×21 gives a change of less than 3% which is essentially negligible for the present purposes.

3.2 Beltagui and MacCallum - adiabatic flow with swirl

For swirl, data with heat transfer are not available to compare to predictions; however, the adiabatic measurements of Beltagui and MacCallum [1976] were considered to test the flow field calculations and to provide guidance. It is expected that when the model provides predictions which are satisfactory for the flow field calculations, then the heat transfer calculations which depend on them will consequently be reasonable too.

Predictions were calculated for the conditions of the experiment by Beltagui and MacCallum [1976] with $Re = 2 \times 10^5$, $S = 0.67$ and $d/D = 0.413$ as expansion ratio. Their measurements do not include inlet velocity profiles; therefore initial values for the present calculations were interpolated to $S = 0.67$ from the data of Habib [1980].

Results based on the standard $K-\epsilon$ model of Section 2 are shown as curves with centerline markings (dash-dot) in Figure 3(a) for axial mean velocity profiles and Figure 3(b) for circumferential mean velocity distributions. Dashed lines represent the experimental observations. It should be noticed that the ordinate scale changes as effects decay downstream.

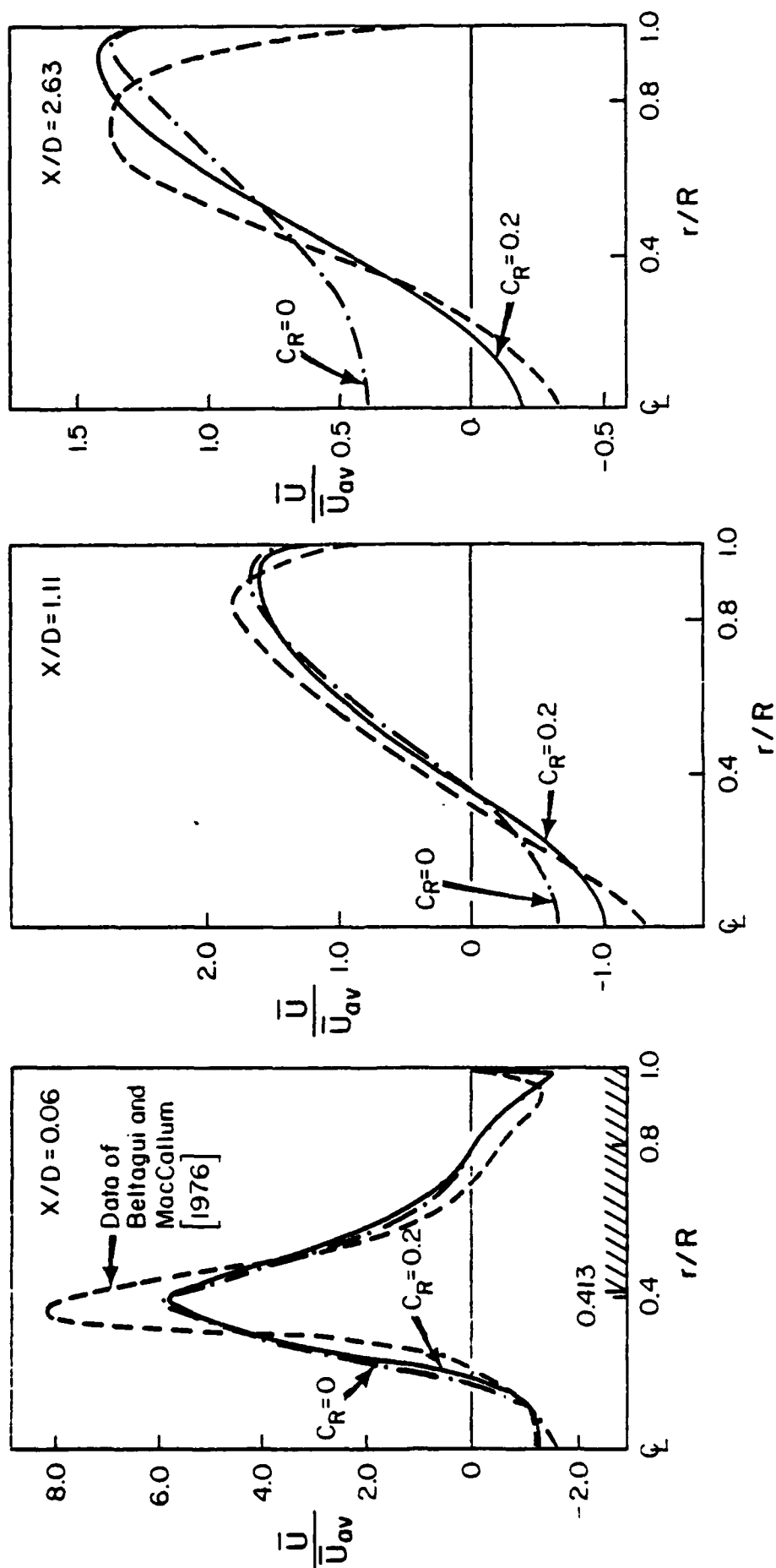


Figure 3(a). Mean velocity distributions for swirl flow without heat transfer, $S \approx 0.67$, $Re = 2 \times 10^5$ and $d/D = 0.413$ as flow parameters.

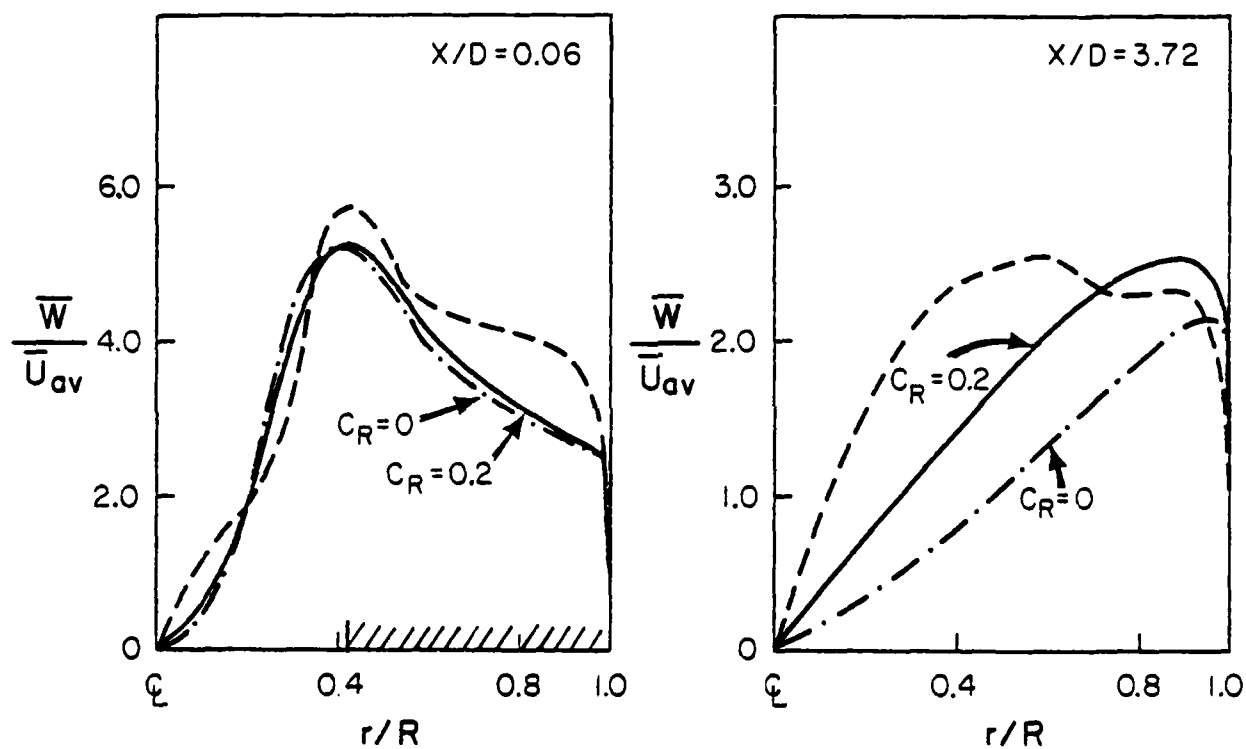


Figure 3(b). Mean velocity distributions for swirl flow without heat transfer (cont'd).

One effect of this relatively high swirl is that the central jet rapidly takes the form of a circular sheet which moves radially outward and attaches to the wall of the outer tube in a short distance. Consequently, the surrounding vortex becomes significantly smaller in size than without swirl and is seen as a small reversed flow region at only the first axial location. Along the centerline of the larger tube a reversed flow also appears showing the existence of another recirculating vortex of size comparable to the diameter of the smaller tube. The qualitative trends are in agreement with the data. The axial velocity profiles agree well except near the centerline where the vortex is predicted to be shorter than measured (e.g., compare centerline velocities at $X/D = 2.63$). Also the tangential velocity in the central region is predicted to decay more quickly than observed.

The discrepancies are partially explained by Khalil and Whitelaw [1974] and Habib and Whitelaw [1980], who showed that the standard $K-\epsilon$ model often yields underpredicted lengths for the central recirculation zones of swirling flows. The work of Pope and Whitelaw [1976] for flows behind a baffle and of Habib and Whitelaw [1981] for confined flows with a sudden expansion indicates that the dissipation equation is the weakest of the governing equations since its modelling is only based on assumptions. As shown by Bradshaw [1973], body forces due to streamline curvature can have a strong effect on turbulence and the equations of turbulence models should account for these effects. Recently, Rodi [1979] and Morse [1980] found that it is essential to

correct one of the empirical constants in the dissipation equation as a function of Richardson numbers to achieve agreement with experiments.

To improve the present calculations, we replaced C_1 , the empirical constant in the dissipation equation, by $C_1(1 + C_R R_f)$ where R_f is the flux Richardson number [Bradshaw, 1973]

$$R_f = \frac{2 \frac{\bar{W}}{r} \frac{\partial \bar{W}}{\partial r}}{\left(\frac{\partial \bar{U}}{\partial r}\right)^2 + \left(\frac{\partial \bar{W}}{\partial r}\right)^2} \quad (11)$$

The use of $C_R = 0.2$ was found to increase the length of the recirculation zone from about 1.7D to 3.6D. This value was chosen to provide the best agreement with the measured size of the central recirculation zone.

Predictions with $C_R = 0.2$ are presented as solid curves in Figure 3. The axial velocity profiles show considerably better agreement with the data in the central recirculating region, as would be expected from the manner of adjustment, and slightly better agreement near the wall. The predicted circumferential velocity distribution is also improved near the wall, the region with the greatest effect on the heat transfer, but as the centerline is approached the values appear to be under-predicted. Comparison of the predicted velocities with $C_R = 0$ and $C_R = 0.2$ shows that modifying C_1 with this functional dependence on R_f tends to decrease the rate of diffusion in the downstream sections and to suppress the development of the jet.

3.3 Heat transfer with swirl at a sudden expansion

Calculations of heat transfer to a swirling flow were made after an abrupt enlargement of $D/d = 2$ for an inlet Reynolds number (Re_d) of 50,000 and with constant wall temperature imposed after the expansion. It was idealized that there was no heat transfer through the annular step between the inner and outer tubes. For flows at the different swirl numbers studied the inlet distributions of velocities and kinetic energy of turbulence were interpolated from the data of Habib [1980]. The effects of varying swirl number and Prandtl number were examined separately.

Effects of swirl were studied with the Prandtl number taken as 0.7 as for common gases. Predictions are shown in Figures 4 through 7. Figure 4, presenting the development of velocity and temperature distributions, provides information which helps understanding the later figures. On these figures the solid curves represent the reference case: zero swirl.

In Figure 4 the velocity profiles show that at $S = 0.1$ the flow is only slightly modified, at $S = 0.4$ there are significant differences and for $S = 1$ the general pattern of the flow changes. These runs might be considered as showing low, medium and high swirl, respectively. As the swirl (or angular momentum) increases, the reattachment point of the separated region in the corner moves successively closer to the entrance and the central jet spreads more rapidly towards the wall. After a short distance the velocity profile for $S = 0.4$ resembles a

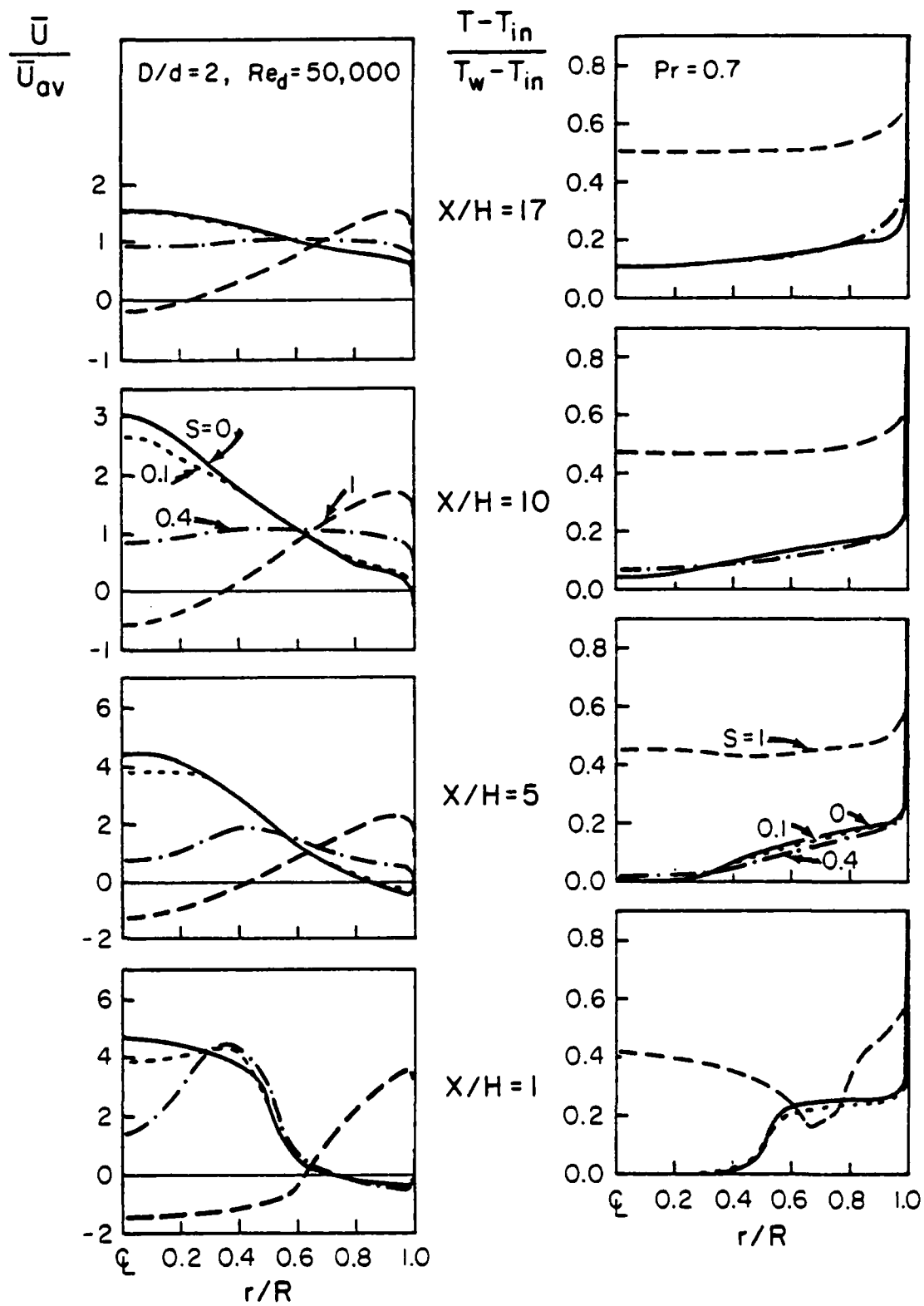


Figure 4. Development of mean velocity and temperature profiles for different swirl numbers.

plug flow. The velocity near the heated wall increases with S in this entry region.

At $S = 1$ the flow pattern resembles that of Beltagui and MacCallum [1976] shown earlier. For this high swirl number the flow exhibits a long central recirculation zone while the outer recirculation zone is decreased to $0.13H$. Flow development is delayed as the central jet becomes a wall jet with the reversed flow at the centerline persisting to beyond $X/H = 17$. These features are also demonstrated by Habib [1980].

The corresponding variation of the non-dimensional wall shear stress or friction factor is shown on Figure 5 (note the change of scale for $S = 1$). Negative values occur due to the reversed flow near the wall. The value of zero indicates the location of the reattachment point or position where the dividing mean streamline meets the wall. As S increases this point progresses upstream as noted from examination of the velocity distribution; the points of maximum shear stress likewise move upstream. The magnitudes of the extremes also increase with swirl; an effect of the centrifugal forces apparently is to drive the higher velocity central flow towards the wall, thus steepening the velocity gradient. For fully developed, turbulent flow in a tube at $Re_D = 25,000$ the friction factor is expected to be about 0.006 [Kays, 1966] so all the flows must go considerably further downstream before becoming fully developed.

The temperature profiles in Figure 4 show approximately isothermal zones across the recirculating regions, thereby demonstrating high transport

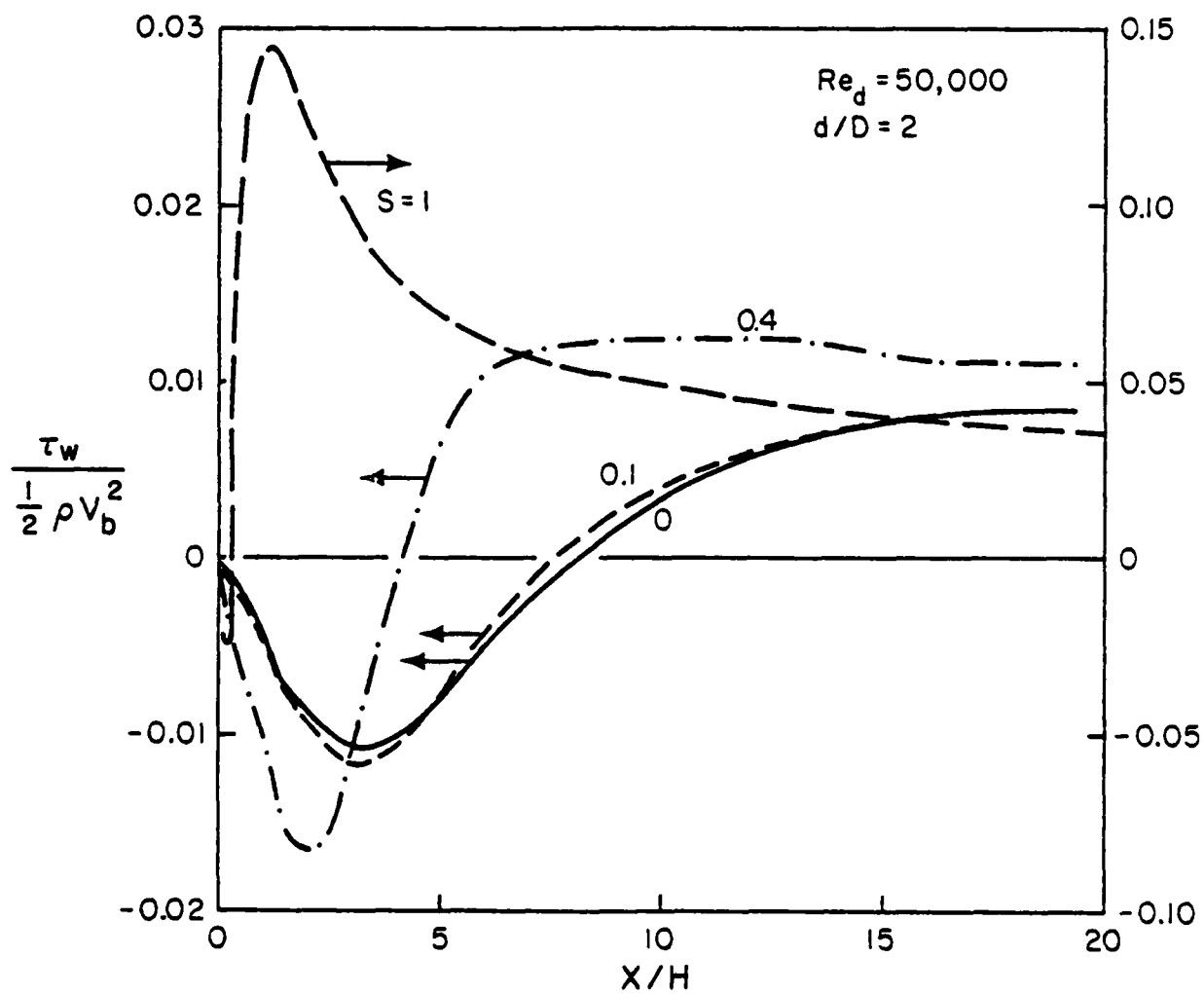


Figure 5. Development of wall shear stress for swirl flow.

rates due to the turbulent mixing in these regions. For the range $S \leq 0.4$ there is only a slight change in temperature distributions as the swirl is increased.

The interesting case is for high swirl flow. Within 5 step heights (or 1.25 D) the central recirculating flow is approximately isothermal. By $X/H = 1$ this reversed flow has carried thermal energy forward from further downstream and, thereby, has raised the centerline temperature substantially. The small recirculating region near the wall has done the same for that region, giving the appearance of a cooled wake at an intermediate radius. However, before $X/H = 5$ the central region has expanded and has eliminated the wake appearance in the temperature distribution.

Predicted local Nusselt numbers are shown in Figure 6 normalized by

$$Nu_{fd} = 0.020 Re^{0.8} Pr^{0.4}$$

a correlation for fully developed turbulent flow with a constant wall temperature [Kays, 1966]. As for the wall shear stress, increasing swirl increases the maximum value and moves it upstream towards the entry. The higher velocity gradient corresponds to flow at a higher Reynolds number so the viscous layer, which dominates the thermal resistance, is thinner and the Nusselt number is increased. Comparison to Figure 5 shows that for each swirl number the maximum Nusselt number occurs near the point of reattachment, which was shown earlier to move upstream as swirl increases.

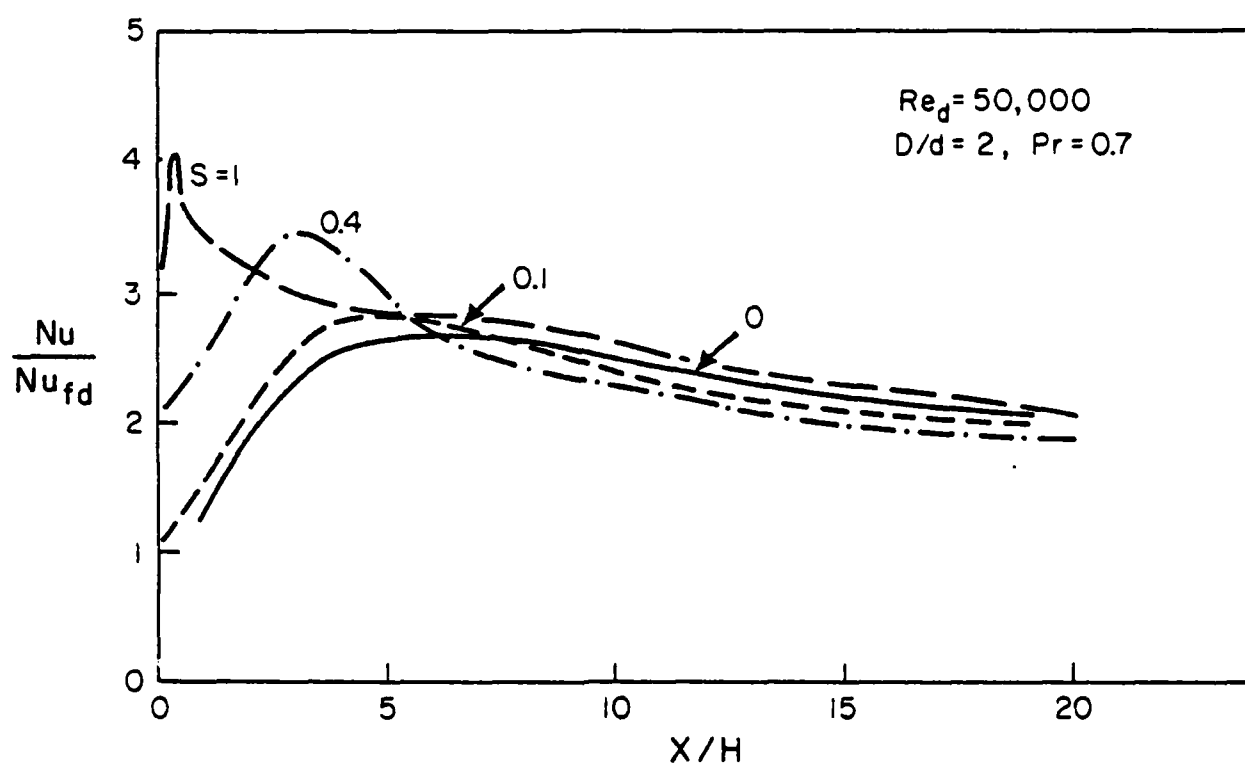


Figure 6. Effect of swirl on development of local Nusselt numbers.

For the high swirl case, the increase in Nu_{max} due to the combined effects of separation at the annular step and swirl is four times while for separation alone the improvement is 2.5, relative to fully developed flow. Thus, the major increase is due to the separation. Further downstream, the low and moderate swirl cases show a slight decrease in Nusselt number as swirl number increases. This result may be the consequence of swirl reducing the length of the outer separated zone so that the development of the normal tube flow has progressed further at a given location, i.e., for given X , $X - X_{reattachment}$ increases with S . For $S = 1$ the strong wall jet phenomenon continues to dominate the velocity profile and, apparently, maintains a slightly higher Nusselt number than for zero swirl.

For applications such as heat exchangers, the designer is interested in the resulting bulk temperature variation of the fluid as it is heated (or cooled). Figure 7 presents this information and it is seen that an effect of swirl is to increase the bulk temperature and its rate of increase slightly. After viewing the temperature profiles, it may seem surprising that the increase is not greater for high swirl. The difficulty is that, though the temperature is approximately uniform, the velocity changes sign in the large reversed flow in the central region. In calculating the bulk temperature via its definition ($\int \bar{U} T dr / \int \bar{U} dr$) the enthalpy flow rate there is subtracted from the enthalpy flow rate in the positive direction giving a net value which is smaller than expected. Therefore, while local heat transfer rates are increased substantially by swirl the overall improvement is not proportionally as large.

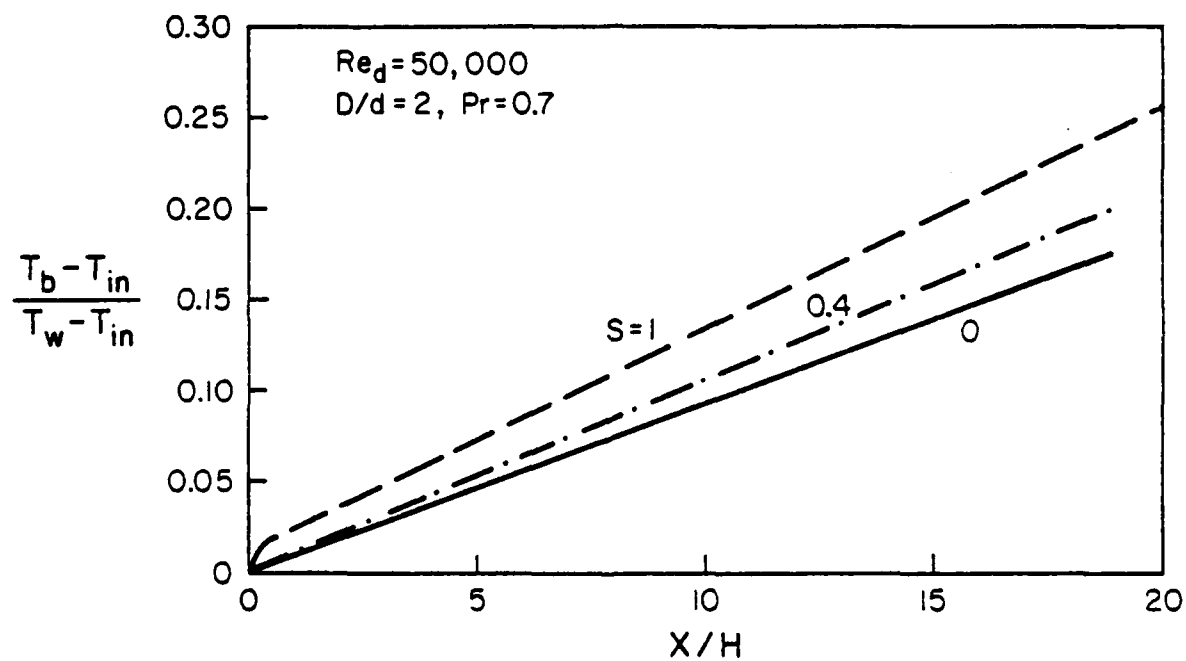
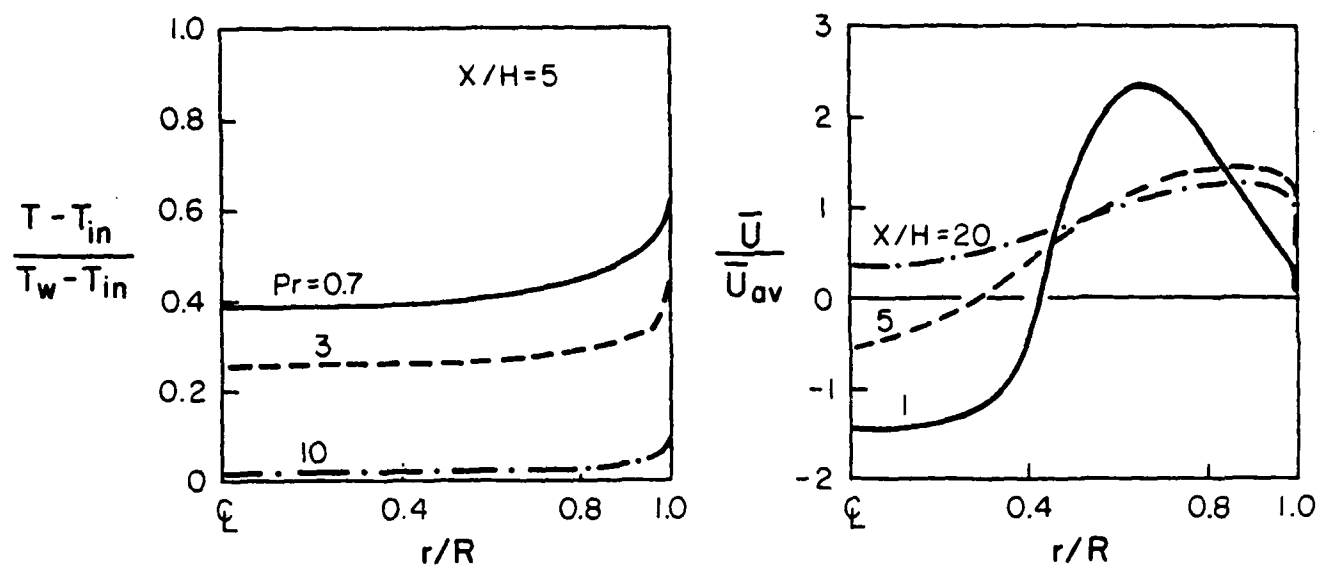


Figure 7. Effect of swirl on overall heating.

Predictions of the effects of Prandtl number on heat transfer were calculated for $S = 0.7$, again at $Re_d = 50,000$ and $D/d = 2$. Examination of the expected velocity profiles in Figure 8 shows this situation to be a high swirl case with a substantial recirculating vortex in the central region. In comparison to the profile for $S = 1$ at $X/H = 1$ in Figure 4, one finds the same shape except the maximum does not fall as close to the wall in this case with lower swirl. Lower circumferential momentum at the entry causes the main streamlines of the annular jet to have less rapid divergence from the axis.

Temperature profiles were calculated for $Pr = 0.7, 3$ and 10 , corresponding to common gases and liquid water at about 60°C and 10°C , respectively. Results are also shown on Figure 8. The general behavior is in agreement with the earlier predictions for $S = 1$.

At each axial location the temperature profiles have the same general shape for all Prandtl numbers. Only the magnitudes change significantly. Thus, the total thermal energy transferred to the flow is greater as the Prandtl number decreases. The similarity of the shapes may be taken as an indication that convection dominates the thermal energy transfer processes in these regions despite the increased thermal conductivity implied by lower Prandtl numbers. The levels of the curves then depend on the heat fluxes at the wall where the thermal conductivity does dominate the thermal resistance. Thus, accurate prediction of heat transfer parameters requires accurate knowledge of the behavior of the viscous layer in this flow as well as in simpler ones.



$Re_d = 50,000 \quad D/d = 2 \quad S = 0.7$

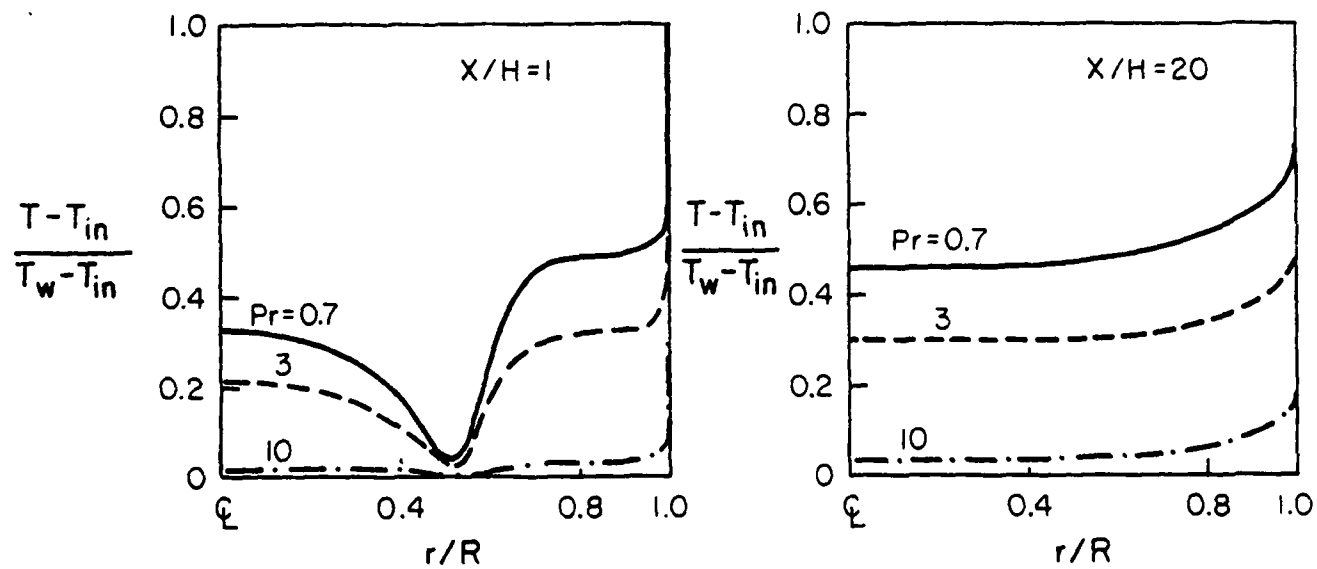


Figure 8. Effect of Prandtl number on temperature distributions.

The axial distribution of local Nusselt numbers is presented as Figure 9; the curve for $Pr = 0.7$ is consistent with the trends of Figure 6 earlier. Again, as with the temperature profiles, the shape of the Nusselt number distribution is primarily dependent on the flow parameters-- Re , S and D/d --and is relatively independent of the Prandtl number. However, the level or magnitude of the results increases as the Prandtl number increases so, as a general conclusion, one may note that the local Nusselt numbers increase with the Prandtl number as in most flows.

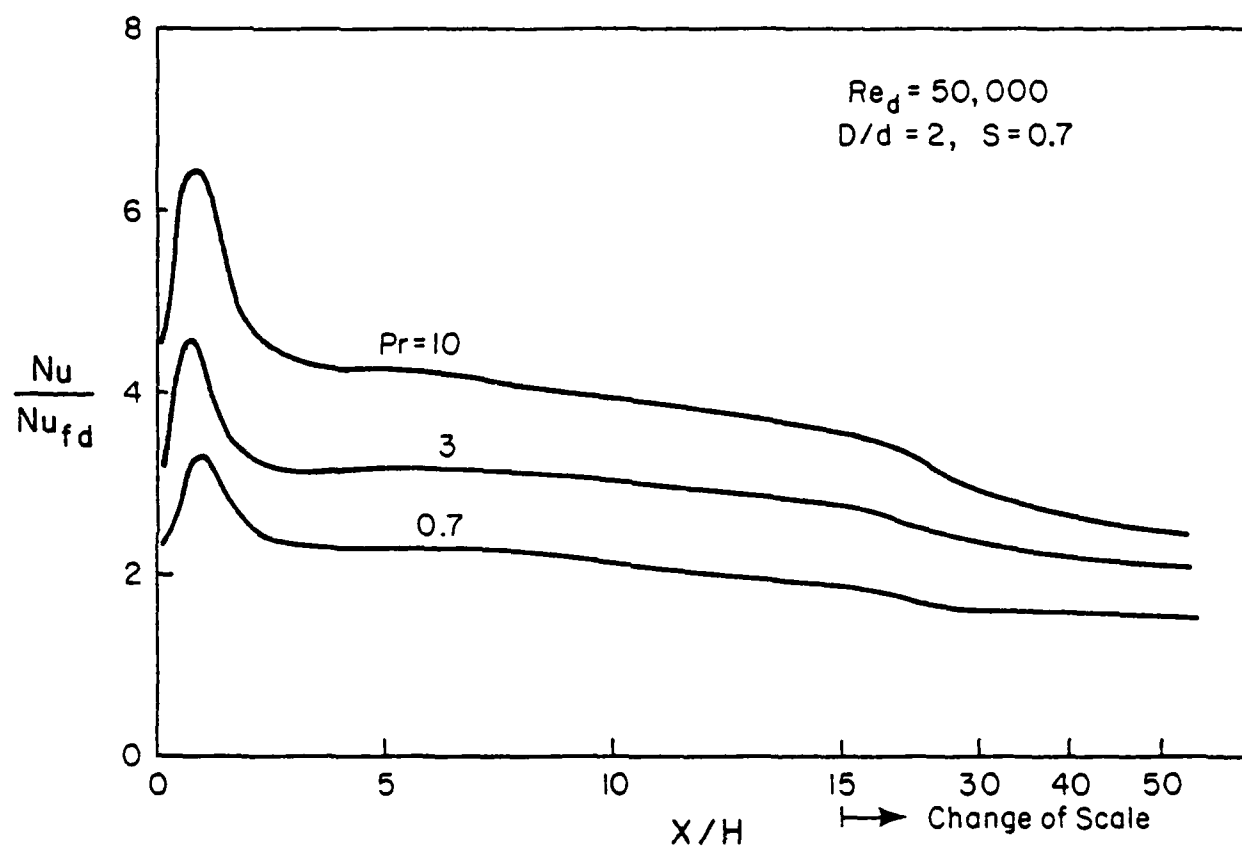


Figure 9. Effect of Prandtl number on local Nusselt number.

CHAPTER 4

CONCLUSIONS

The following important conclusions can be extracted from the preceding results:

- 1) For heat transfer without swirl, generally good agreement is obtained between measured and calculated Nusselt numbers.
- 2) For an adiabatic flow with swirl, predictions could be obtained in agreement with available data by modifying the turbulence model to account for effects of streamline curvature.
- 3) For swirl flow with heat transfer, (a) the maximum local Nusselt number is predicted to increase and its location moves towards the entry as the swirl number increases, and (b) the local Nusselt number increases as the Prandtl number increases.

APPENDIX. COMPUTER PROGRAM*

1. Comments concerning parameters used in Block data of Teach T.CDC

IGET: This parameter indicates continuation of calculations
 from previous output saved on magnetic tape

 ≡1 Does not read or write any output

 >1 Writes data on tape 8

 >2 Reads from Tape 7 and writes on Tape 8

IHT Parameter controls heat transfer calculations

 ≡1 There is a heat transfer calculation (the thermal
 energy equation is solved)

 ≡0 No heat transfer calculations

ICP Parameter used to restrict the program to calculations for a
 constant property fluid

 ≡0 Flow field calculations only

 ≡1 Energy equation is solved alone provided that the flow
 is known from a previous set of iterations. In this
 case (i.e., ICP = 1) IHT should be set to 1 also.

 The program may consider changes in density when the
 fluid is air. For this case set IHT = 1 and ICP = 0.

IABN Defines North wall (i.e., larger tube) as adiabatic

 ≡0 There is heat transfer through the North wall

 ≡1 There is no heat transfer through the North wall

IABW Sidewall (end wall) boundary situation

 ≡0 There is heat transfer through the side wall.
 (This option was not tested.)

 ≡1 There is no heat transfer through the side wall

*Questions concerning the details of the program should be directed to
Dr. M. A. Habib, Power Section, Mechanical Engineering Department, Cairo
University, Cairo, Egypt.

CRF	Constant C_R used with equation (11) for streamline curvature correction. (It is only useful in the swirl cases.)
$\equiv 0$	There is no correction
$\equiv 0.2$	There is a correction (with $C_R = 0.2$)
MAXIT	Maximum iterations allowed. (It will write the results if IGET>2)
INDPRI	Defines the number of iterations after which the program shows a sample of the calculations
PO	Atmospheric pressure, N/m^2
CP1	Specific heat, $J/Kg\ C^\circ$
GASCON	Universal gas constant = $8314\ J/Kgmole\ C^\circ$
WFLUID	Molecular weight, $Kg/Kgmole$
COND	Thermal conductivity, $Watt/m\ C^\circ$
NOTAIR	
$\equiv 0$	The fluid is air or a comparable gas
$\equiv 1$	The fluid is not air and the properties for the fluid must be inserted. These are identified as CPFL, CONDFL, VISFL, DENFL)
PRLAM, PRH	Laminar and turbulent Prandtl numbers
DENFL	} The properties of the fluid to be used if it is not considered to be air. (NOTAIR should be equal to 1 to consider these.)
CPFL	
VISFL	
CONDFL	
ICHF	Indicates whether there is a constant heat flux boundary condition along the larger tube.
$\equiv 0$	Not constant heat flux; the program needs a specified wall temperature distribution
$\equiv 1$	Constant heat flux boundary condition

QADD Wall heat flux in the case of a constant heat flux boundary condition, J/m^2s . (To be used only if ICHF = 1)

TIN Inlet temperature, K°

SWNO Swirl number, defined as the axial flow of tangential momentum divided by the axial momentum rate, $\int_0^R \bar{U} \bar{W} r^2 dr / R \int_0^R \bar{U}^2 r dr$, where R is the nozzle radius.

$\equiv 0$ No swirl

$\equiv$ any value Magnitude of swirl number

UFUIN The maximum inlet velocity (on the centerline); important only if SWNO = 0. The program then treats the inlet as fully developed pipe flow and UFUIN is the value at the centerline.

RLARGE }
 & }
 DNOZLE }

Geometry



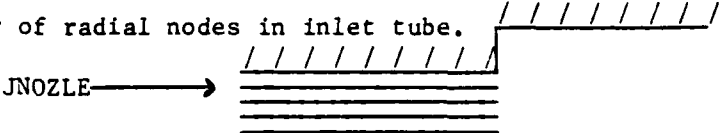
IDATA Parameters specify if the input velocities, turbulence kinetic energy and dissipation will be supplied via READ statements or will be calculated by the program as fully developed pipe flow.

$\equiv 0$ For swirl number = 0 (SWNO = 0)

$\equiv 1$ For others. In this case, at the University of Arizona Computer Center, the program Teach.Vel should be run first; its output should be saved on Tape 4; then the present program is run and it will automatically read $\bar{U}$, $\bar{W}$, K , ϵ from Tape 4.

NI, NJ Number of nodes in x and y directions, respectively

JNOZLE Number of radial nodes in inlet tube.



X Locations of nodes in the x-direction

Y Location of nodes in the radial direction (starts from centerline)

TWALN North wall temperature, K

TWALW Side wall temperature, K

2. Program listing (following pages)

BLICK DATA RECAP:

[illegible]

```

BLOCK DATA SUKDAT. 74/1/1, 71-1) PAGE 4.8+528
DATA MINJ/22.21/
DATA JHJL/24/
DATA X/233333.0035.0045.0055.0065.0075.0085.0095.011,
1.013.02.03.04.05.06.07.08.09.10.11.12.13.14.15,
2.16.17.18.19.20.21.22.23.24.25.26.27.28.29.30.31,
3.32.33.34.35.36.37.38.39.40.41.42.43.44.45.46.47,
4.48.49.50.51.52.53.54.55.56.57.58.59.60.61.62.63,
5.64.65.66.67.68.69.70.71.72.73.74.75.76.77.78.79,
6.80.81.82.83.84.85.86.87.88.89.90.91.92.93.94.95,
7.96.97.98.99.100.101.102.103.104.105.106.107.108.109,
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69.903.904.905.906.907.908.909.910.911.912.913.914.915,
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05/19/81 12.05.45

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TPYUGPAF MAIN

[illegible]

[illegible]

05/18/81 12.04.45

FIN 4.04520

741.15 19150 124C6

PROGRAM MAIL.

Line	Code	Text	Address
136		IF(UJIN)=UFUJIN**1.5/(ALAMDA**KLARGE)	136
137		EDATIN=TEATIN**1.5/(ALAMDA**KLARGE)	137
138		HIN=CP*ITIN	138
139		DO 64 J=1,NUM1	139
140		IF(J.GT.JNOZLE) GO TO 2117	140
141		U(I,J)=UFUJIN*(1-ABS(Y(I,J)))/(Y(JNOZLE+1))**AB	141
142		V(I,J)=VFUJIN*(1+CONS1*((ABS(Y(I,J)))/(ONZLE/2))**AN2))	142
143		W(I,J)=WFUJIN	143
144		FO(I,J)=FE(I,J)**1.5/(ALAMDA**KLARGE)	144
145		R1=ONZLE/2	145
146		ALP=RU*10.75*ALM	146
147		ALP=CMU**0.75*ALM	147
148		ALM=0.03*RI	148
149		ALP=ALM	149
150		DATA OF CHATKURVEI (1953)	150
151		U(I,J)=UFUJIN	151
152		U(I,J)=UJIN	152
153		U(I,J)=CHDATE(I,J)**1.5/ALP	153
154		VIS(I,J)=DEMT(JJ)**2*(CMU/ED(I,J))**VISCOS	154
155		OFN(I,J)=DEF	155
156		H(I,J)=HIN	156
157		GO TO 3	157
158		CONTINUE	158
159		IF(I,J)=SMALL	159
160		ED(I,J)=SMALL	160
161		V(I,J)=SMALL	161
162		H(I,J)=SMALL	162
163		CONTINUE	163
164		V(I,J)=SMALL	164
165		H(I,J)=SMALL	165
166		GO 71 J=1,N1	166
167		U(I,NJ)=SMALL	167
168		V(I,NJ)=SMALL	168
169		W(I,NJ)=SMALL	169
170		FO(I,NJ)=SMALL	170
171		GO(1,NJ)=SMALL	171
172		U(I,NJ)=SMALL	172
173		IF(IABN=NC)GO TO 71	173
174		IF(IABN)=TVAL(I)	174
175		H(I,NJ)=CPL*THAL(NI)	175
176		CONTINUE	176
177		IF(IABN=NC)GO TO 74	177
178		DO 72 J=2,NJ	178
179		IF(IJLE=JNOZLE)GO TO 72	179
180		U(I,J)=FWALW(I)	180
181		V(I,J)=FWALW(I)	181
182		H(I,J)=CPL*THAL(IJ)	182
183		CONTINUE	183
184		CONTINUE	184
185		CONTINUE	185
186		CONTINUE	186
187		CONTINUE	187
188		CONTINUE	188
189		CONTINUE	189
190		CONTINUE	190
191		CONTINUE	191
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231		CONTINUE	231
232		CONTINUE	232

05/18/81 12.05.45

NIJ 4-0467

74/1175 701-0124CE

PROGRAM MAIN

```

0900 CONTINUE
C-----PROGRAM CONTROL AND MONITOR
      JMON=5
      INCALU=.TRUE.
      INCALV=.TRUE.
      INCALW=.TRUE.
      INCALP=.TRUE.
      INCALK=.TRUE.
      INCALD=.FALSE.
      INCALH=.FALSE.
      IF (.NOT. .EQ.) INCALH=.TRUE.
      INPRD=.TRUE.
      IF (.SMND.EQ.0.0) INCALW=.FALSE.
      IF (.TICP.EQ.0.0) GO TO 2227
      INCALU=.FALSE.
      INCALV=.FALSE.
      INCALW=.FALSE.
      INCALP=.FALSE.
      INCALK=.FALSE.
      INCALD=.FALSE.
      INPRD=.FALSE.
      INFORM=1.
2227 CONTINUE
      C
      HEUV =0.3
      UPELV =0.3
      UPEVIS =1.0
      UPFKK =0.7
      UPFLC =0.7
      URTHL =0.8
      UPEFF =0.8
      UREFHO=0.4
      UPINPD =0.4
      SLPMAX=1.0E-4
      C
      CALL PROPS
C*****
C CHAPTER 2 2 2 2 INITIAL OPERATIONS 2 2 2 2 2 2
C
      AIRI1 =6.0
      TEMPI =6.0
      PRUZLE =SVJ(JNITUE+1)
      DO 2019 J=2,JNOZLF
        AIRI1=DEVAL(J)*SNS(J)+UL(J)*S2+AIRI
        TEMPI=DEVAL(J)*J+SNS(J)+O(L,J)+TERPI
2019 CONTINUE
      SLLA=SMUD
      SLLA=SMPF
      AIRI=AIRI+SLPMAX*AIRI/TEMPI
      DO 1 J=C,JNOZLC
        W(L,J)=AIRI*(J)
12 W(L,J)=W(L,J)

```

FTN 4.04920

707-0 YPAC

PIVOT AM MAIN

```

349  ALFAL=0.25*(DUMZLE/2)*.03/ KLRAGE**3
350  CL 12.1=3.0*W1
351  DO 14 J=2,N1,M1
352  UC(J)=ALF*ALFAL*Y(J)
353  14 UC(J)=ALF*ALFAL*Y(J)
354  READ(4)(CU(1,J),I=2,INTLE)
355  READ(4)(W(1,J),I=2,INTZLE)
356  READ(4)(TE(1,J),I=2,JNZLE)
357  DO 3282 J=2,JNZLE
358  UC(2,J)=U(1,J)
359  W(2,J)=W(1,J)
360  V(2,J)=V(1,J)
361  T(2,J)=TE(1,J)
362  ED(2,J)=ED(1,J)
363  VIS(1,J)=CMU*DE*(1,J)*TE(1,J)*2/ED(1,J)+VISCO
364  CONTINUE
365  3281 ANFUEL=C*U
366  ARFUEL=C*U
367  IF (IGT-GL2*READ(7)F,YPLUS,NITER
368  DO 1577 J=2,N1
369  T(1,J)=TMAX(I)
370  W(1,J)=WMAX(I,NJ)
371  1577 CONTINUE
372  DO 1576 J=1,JNZLE
373  T(1,J)=TIN
374  W(1,J)=CPI*(1,1)
375  CONTINUE
376  MAXITER=NITER
377  NITER=NITER
378  UC(4018-J-2,JNZLE
379  ARFUEL=ARFUEL*(J)*SNS(J)*DINF
380  ANFUEL=ANFUEL*(J)*SNS(J)*DINF
381  CONTINUE
382  FLOWIN=AMFUEL
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6
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44-38861-101

[illegible]

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3613 J=2-1*UNW(I)/(.5*DENSI*(AUVEL+VELRAT))*2)
3614 DO 401 (I=1,400) YV(JNDLE+I)=1
3615 WRITE(40311,100)S2,S5,I,S5C2,YPLUSN(I)
401 CONTINUE
3616 DO 1508 (I=1,400) YVPLUSW(I)=J-2*NJMI
3617 HELXN(I)=GAMN(I)*(1+(I+J)-HEL(NJMI))
3618 CONTINUE
3619 DO 1509 J=2,NJMI
3620 IF (J.LE.JNDLE) GO TO 1910
3621 HELXY(J)=GAMW(J)*(1+(J-1)-HEL(J))
3622 CONTINUE
3623 AMAS=0.
3624 DO 1912 J=2,NJMI
3625 UP=0.5*(U(I+J)+U(I+J+1))*DEN(I,J)
3626 AMAS=AMAS+UP*(1.0)*DEN(I,J)*SNS(J)
3627 ENR=ENR+UP*(1.0)*DEN(I,J)*R(J)*SNS(J)
3628 CONTINUE
3629 AMAS=0.
3630 AMAS=AMAS
3631 ENR=0.
3632 DO 1913 J=2,NJMI
3633 UP=0.5*(U(I+J)+U(I+J+1))*DEN(I,J)
3634 APAS=APAS+UP*(1.0)*SNS(J)
3635 ENR=ENR+UP*(1.0)*DEN(I,J)*R(J)*SNS(J)
3636 CONTINUE
3637 TEP=ENR/AMAS
3638 ALF=X(MI)-X(I)
3639 DELTD=TD-I
3640 DO 1914 I=2,NJMI
3641 TOLUK(I)=(X(I)-X(I+1))/ALF*DELTD*TD
3642 VISFAC=4.15*(TOLUK(I)-310.73)+1912)*1.E-8/VISCO
3643 MOLF(I)=FLXN(I)/TALN(I)-TOLUK(I)
3644 TOTF(I)=TALN(I)-TOLUK(I)
3645 C GRUO=62421E9*H(I)-273.15*3237E-5
3646 HUS(I)=MOLF(I)/ALF*273.15*3237E-5
3647 CONTINUE
3648 PF=4.9*FLXN/VISCO*0.1456
3649 RI=PF*TM/TOL*(NTMI)/VISFAC
3650 HUSDED=0.058*RI*PLAM*56
3651 TOTTRU=(NTMI)/TALN(MI+1)+0.575
3652 DO 1915 I=2,NJMI
3653 HUS(I)=HUS(I)/HUSDED
3654 CONTINUE
3655 WRITE(406)
3656 DO 407 I=2,NJMI
3657 XH=X(I)/RLAGE-YV(JNDLE+I)
3658 WRITE(40511,100)XH,HELXN(I),MOLF(I),HUS(I),HUSN(I)
3659 WRITE(410)
3660 DO 411 I=2,NJMI
3661 HUSDED=42300,RIUK(I),TOTF(I),TALN(I)
3662 CONTINUE
3663 WRITE(411,100)XH,3237E-5
3664 CONTINUE

```


[illegible]

[illegible]


```

SUBROUTINE PRUPS      74/175  30T=0 TRACE      05/13/81  12.05.45
60  DO 202 J=2,NJM1
    DEN(J)=DEN(J)+URFV(J)*VISC(J)
202 CONTINUE
65  DO 100 I=2,NJM1
    DO 102 J=2,NJM1
    VISC(I,J)=E0*0.1 GOTO 102
    IF (DEN(I,J) .EQ. 0.1) GOTO 102
    VISC(I,J)=DEN(I,J)*2*CMU/ED(I,J)*VISCOS
    GO TO 100
102 VISC(I,J)=VISCOS
C-----UNDER-RELAX-VISCOSITY
101 VISC(I,J)=URFV(I,J)*VISC(I,J)+(1.-URFV(I,J))*VISOLD
    VISC(I,J)=VISC(I,J)
    IF (DEN(I,J) .EQ. 0.1) DEN(I,J) = 0.10
100 CONTINUE
    RETURN
    END

```

```

PRUPS 20
PRUPS 21
PRUPS 22
PRUPS 23
PRUPS 24
PRUPS 25
PRUPS 26
PRUPS 27
PRUPS 28
PRUPS 29
PRUPS 30
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PRUPS 34
PRUPS 35
PRUPS 36
PRUPS 37
PRUPS 38
PRUPS 39
PRUPS 40
PRUPS 41
PRUPS 42

```



```

SUBROUTINE CALCU      74/175  021=0 TRACE      05/18/81  12.05.44
110  C-----UNDEP-RELAXATION
      AP(1:J)=AP(1:J)/UPFU
      SU(1:J)=SU(1:J)+AP(1:J)*U(1:J)
      DU(1:J)=DU(1:J)+UPFU
      301 CONTINUE
      300 CONTINUE
      C CHAPTER 4 4 4 SOLUTION OF DIFFERENCE EQUATION 4 4 4 4 4
      C
      ON 400 N=1,HISMPH
      400 CALL LISULV(3,2,0)
      RETURN
      END
      CALCU      77
      CALCU      79
      CALCU      80
      CALCU      81
      CALCU      83
      CALCU      84
      CALCU      85
      CALCU      86
      CALCU      87
      CALCU      88
      CALCU      89

```

[illegible]

[illegible]

LINE	CODE	STATEMENT	OPERATOR	REGISTER	ADDRESS	DATA
111	DU	301 J=3,NMI				
112	DU	301 J=3,NMI				
113	AP	100 J=3,NMI				
114	AP	100 J=3,NMI				
115	AP	100 J=3,NMI				
116	AP	100 J=3,NMI				
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180	AP	100 J=3,NMI				
181	AP	100 J=3,NMI				

05/13/81 12.05.45

FIN 4.0+52H

SUBSIDIARY CAPITAL 741176 731250 TRADING

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	90	91	92	93	94	95	96	97	98	99	100
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C-----
DO 101 J=2,NM1
  COMPUTE AREAS AND VOLUME
  AREA=SV(J)+SV(J+1)
  AREA=AREA+SV(J)+SV(J+1)
  AREAEM=AREA*(J)*SN(S(J))
  VOL=RC(J)*SN(S(J))*SV(J)
C-----
CALCULATE COEFFICIENTS
DEN=0.5*(DEN(I,J)+DEN(I,J+1))
DENS=0.5*(DEN(I,J)+DEN(I,J+1))
DENF=0.5*(DEN(I,J)+DEN(I,J+1))
DENP=0.5*(DEN(I,J)+DEN(I,J+1))
GO TO 49
48 DEN=-DEN(I,J)
49 GO TO 51
50 DENE=-DEN(MT,J)
51 IF(I.EQ.NJM1) GO TO 53
52 DENN=-DEN(I,N1)
53 IF(I.LO.2) GO TO 54
54 GO TO 55
55 DENT=DEN(I,1)
  CONTINUE
  AN(I,J)=DENH*AREAN(V(I,J)+1)
  AV(I,J)=DENH*AREAS(V(I,J))
  AE(I,J)=DENH*AREAE(V(I,J))
  AL(I,J)=DENH*AREAL(V(I,J))
C-----
CALCULATE SOURCE TERMS
CN=DEN*V(I,J)+I*AREAN
CE=DEN*V(I,J)+J*AREAS
CF=DEN*U(I,J)+J*AREASM
CW=DEN*U(I,J)+J*AREAM
CMP=CN-CS+CF-CW
SP(I,J)=0.0
SUM(I,J)=CMP
C-----
COMPUTE SUM OF ABSOLUTE MASS SOURCES
RESORM=RESORM+ABS(SMP)
101 CONTINUE
100 CONTINUE
CHAPTER 2 2 2 2 2 2 2 PROBLEM MODIFICATIONS 2 2 2 2 2 2 2
CALL MOOP
CHAPTER 3 3 3 3 3 FINAL COEFFICIENT ASSEMBLY 3 3 3 3 3 3 3
DO 300 I=2,NM1
  DO 301 J=2,NJM1
    AP(I,J)=AV(I,J)+AS(I,J)*AE(L,J)+AW(I,J)-SP(I,J)
    SP(I,J)=0.0
  301 CONTINUE
  300 CONTINUE
CHAPTER 4 4 4 4 4 SOLUTION OF DIFFERENCE EQUATIONS 4 4 4 4 4 4 4

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SUBROUTINE CALCP      74/175  OPT=0 TRACE      05/18/81  12.05.45
FTN 4.0+528
115  C CHAPTER 5 5 5 CORRECT VELOCITIES AND PRESSURE 5 5 5 5
      DO 400 N=1, NSMPP
      400 CALL LISOLV(2,2,PP1)
120  C-----VELOCITIES
      DO 500 J=2,NJM1
      500 J=2,NJM1
      IF (J.NE.2) V(1,J)=V(1,J)+8*V(1,J)+16*V(1,J)+8*V(1,J)
125  301 CONTINUE
      C-----PRESSURES (WITH PROVISION FOR UNDER-RELAXATION)
      PPREF=PP(1,PPREF)
      DO 502 I=2,NJM1
      502 I=2,NJM1
      P(1,I)=8*PP(1,I)+URFP*(PP(1,I)-PPREF)
130  302 CONTINUE
      302 CONTINUE
      END
135

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77  CALCP
78  CALCP
79  CALCP
80  CALCP
81  CALCP
82  CALCP
83  CALCP
84  CALCP
85  CALCP
86  CALCP
87  CALCP
88  CALCP
89  CALCP
90  CALCP
91  CALCP
92  CALCP
93  CALCP
94  CALCP
95  CALCP
96  CALCP
97  CALCP

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SUBROUTINE CALCU      74/175  321=0  IPACE      FIN 4,0+528      05/14/81  12.05.46
60  G=0.5*(DEN(I,J)+DEN(I+1,J)+U(I+1,J))
    G=0.5*(DEN(I,J)+DEN(I-1,J)+U(I,J))
    CR=ONVAREEN
    CS=0.2*AREAEW
    CF=0.2*AREAEW
    CW=0.2*AREAEW
    C-----CALCULATE DIFFUSION COEFFICIENTS
    GAMN=0.5*VIS(I,J)+VIS(I+1,J)/PRTE
    GAMN=0.5*VIS(I,J)+VIS(I-1,J)/PRTE
    GAMN=0.5*VIS(I,J)+VIS(I+1,J)/PRTE
    DN=GAMN*AREAN/DVPS(I)
    DS=GAMN*AREAS/DVPS(I)
    DE=GAMN*AREAEW/DXEP(I)
    DA=GAMN*AREAEW/DXPM(I)
    C-----SLURC
    SMP=CN-CS*CF-CW
    CP=MAX(10.0,SM)
    CPO=CP
    DUDX=(U(I+1,J)-U(I,J))/SEMI
    DUDY=(V(I+1,J)-V(I,J))/SNS(I)
    LUT(I,J-1)=U(I,J)+U(I+1,J)/4.-(U(I,J)+U(I+1,J)+
    DUDX*(V(I+1,J)-V(I,J))/SNS(I))
    DUDX=(V(I+1,J)-V(I,J))/SNS(I)
    DUDY=(U(I+1,J)-U(I,J))/4.-(V(I,J)+V(I+1,J)+V(
    GEN(I,J)=(2.+(DUDX**2)+DUDY**2)*VIS(I,J)
    VDR=V(I,J)/RV(I)
    IF(I.EQ.2) VDR=0.0
    IF(I.EQ.3) GEN(I,J)=GEN(I,J)+0.5*(VDR*V(I,J+1)/
    IF(I.EQ.4) RV(I,J)**2
    C-----GENR11=GENR11
    IF(I.SNO.EQ.0.0)GO TO 102
    DMDX=(U(I+1,J)+U(I,J))/2.-(U(I,J)+U(I-1,J))/2.)/SNS(I)
    DMDY=(V(I+1,J)+V(I,J))/2.-(V(I,J)+V(I-1,J))/2.)/SNS(I)
    WDR=U(I,J)/I(J)
    GEN(I,J)=GEN(I,J)+VIS(I,J)*(DMDX**2+(DMDY-HDR)**2)
    102 CONTINUE
    C-----GENR22=GENR22
    RIF(I,J)=J-0.5*(1.-GENR11/GENR22)
    GENR11=DUDY**2+DUDY*DUDY
    RIF(I,J)=2.*(WDR*DUDY/GENR11
    IF(IASR(I,J).GT.1)/CRF*IF(I,J)=
    10.25*(RIF(I,J)+RIF(I+1,J)+RIF(I-1,J)+RIF(I,J-1))
    PUE(I,J)=GEN(I,J)/ED(I,J)
    C-----ASSEMBLE MATN COEFFICIENTS
    AN(I,J)=MAX(CAS(0.5*CN),DN)-0.5*CN
    AE(I,J)=MAX(CAS(0.5*CS),DS)-0.5*CS
    AN(I,J)=MAX(CAS(0.5*CF),DE)-0.5*CF
    SU(I,J)=CP+I(J)
    SUK(I,J)=SU(I,J)
    SP(I,J)=SP(I,J)+P
    SPN(I,J)=SP(I,J)
    SPAT(I,J)=SP(I,J)-CD*CHI*DEN(I,J)+2*TE(I,J)*VUL/VIS(I,J)
    101 CONTINUE
    100 CONTINUE

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SUBROUTINE CALCIE      74/174  OPT=0 TRACE      05/18/81  12.05.44
11> C CHAPTER 2 2 2 2 2 2 PROBLEM MODIFICATIONS 2 2 2 2 2 2
C CALL MODEL
C CHAPTER 3 FINAL COEFFICIENT ASSEMBLY AND RESIDUAL SOURCE CALCULATION 3
C
  NLSOPK=0.0
  DO 300 I=2,NINI
    AP(I,J)=AN(I,J)+AS(I,J)*AC(I,J)+AW(I,J)-SP(I,J)
    PLSOR=AN(I,J)+AS(I,J)*TE(I,J)-1.0*TE(I,J)+TE(I,J)*SU(I,J)
    VOL=K(I)*SEW(I)*SNS(I)
    SURVOL=GRAT*VOL
    IF (SP(I,J).GT.0.0) PLSOR=PLSOR/SURVOL
    NLSOPK=NLSOPK+PLSOR*TRF508
    C-----UNDERRELAXATION
    AP(I,J)=SJ(I,J)+(1.0-URFK)*AP(I,J)+TE(I,J)
    300 CONTINUE
  130 C
  135 C CHAPTER 4 4 4 4 4 SOLUTION OF DIFFERENCE EQUATIONS 4 4 4 4 4
C
  DO 400 N=1,NLSOPK
    400 CALL LISOLV(2,2,TE)
  RETURN
  END

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CALCIE 104

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[illegible]


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SUBROUTINE CALCD      74/1/5  0210 TRACE      05/18/81  12.05.45
110      CHAPTER 4 4 4 4 4 SOLUTION OF DIFFERENCE EQUATIONS 4 4 4 4
C      00 400 N=1, NSWP0
      400 CALL LSOLV(2,2,FD)
      RETURN
      END
120

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 CALC CD
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 CALC CD
 CALC CD

SUBSTITUTED CALCIUM

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C0  GL=0.5*IDEN(I,J)+DEH(I,I,J)+U(I,I,J)
C1  GM=0.5*IDEN(I,J)+DEH(I-1,J)+J(I,J)
C2  GN=GN+AREAN
C3  GO  CE=GE+AREAEW
C4  C-----CALCULATE DIFFUSION COEFFICIENTS
C5  DIMS =0.5*(VIS(I,I)+VIS(I,J-1))/PRH
C6  DEHN =0.5*(VIS(I,I)+VIS(I,J-1))/PRH
C7  DHM =0.5*(VIS(I,J)+VIS(I+1,J))/PRH
C8  DHM=FMH*AREAN/DYHP(I,J)
C9  DS=DEHN+AREAS/DYPS(I,J)
C10 D=DEHN+AREAF/XPW(I)
C11 DI=DEME+AREAF/XEP(I)
C12 C-----SOURCE TERMS
C13 SMP=CN*CS+CE*CH
C14 CPD=CP
C15 C-----ASSEMBLE MATN COEFFICIENTS
C16 ANCI(I,J)=MAXI(ATS(I,0.5*CN),DM)-0.5*CM
C17 ACII(J)=MAXI(ATS(I,0.5*CS),DI)-0.5*CS
C18 ACII(J)=MAXI(ATS(I,0.5*CE),DEI)-0.5*CE
C19 AHI(I,J)=MAXI(ATS(I,0.5*CH),DJ)+0.5*CH
C20 SUI(I,J)=CPD+I(I,J)
C21 SPFI(I,J)=CP
C22 C-----
C23 101 CONTINUE
C24 100 CONTINUE
C25 C
C26 CHAPTER 2 2 2 PRTLEY MODIFICATIONS 2 2 2
C27 C
C28 CALL MODH
C29 CHAPTER 3 FINAL COEFF. ASSEMBLY AND RESIDUAL SOURCE CALCULATIONS 3
C30 C
C31 RESORH=0.0
C32 DC 300 I=2,NIMI
C33 DO 301 J=2,NJMI
C34 T(I,J)=H(I,J)/CPI
C35 AP(I,J)=AV(I,I)+AS(I,I)+AE(I,J)+AW(I,J)-SPFI(J)
C36 H(I,I)=H(I,I)
C37 HMI(J)=H(NIMI,I)
C38 TINI(J)=T(NIMI,I)
C39 PLSOR(I,J)=H(I,I)+H(I,J)+AS(I,J)+H(I,J-1)+AE(I,J)+H(I+1,J)+
C40 I*U(I,J)+H(I-1,J)-AP(I,J)+H(I,J)+SUI(I,J)
C41 U(I,J)=V(I,I)+SUS(I,J)+FMI(I)
C42 SUBVOL=GE*AT*V(I)
C43 I(I-SPFI(J),GI(J))=SORV(J) RESOR=RESOR/SORVOL
C44 PLSOR=PLSOR/PLSORSI
C45 C-----ORDER RELAXATION
C46 AP(I,J)=AP(I,J)/JHPH
C47 SUI(I,J)=SUI(I,I)/(1.-H*F(I+AP(I,J))+I(I,J)
C48 301 CONTINUE
C49 300 CONTINUE
C50 C
C51 DO 400 N=1,NSUP4
C52 CALL LISOLV(2,2,H)

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115 SUBROUTINE CALCH 74/175 OPT=0 TRACE 05/18/81 12.05.46
 RETURN
 END
 CALCH 77
 CALCH 78

[illegible]

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SUBROUTINE PROMOD 74/175 DPE=0 TRACE FIN 4.8*28 05/18/81 12.05.44
  J=NJMI
  DC=1333 I=2,NJMI
  DENU=0FNI(1,J)
  SORTK=0FNI(1,J)
  YPLUSN(1)=DENUSORTK*CDTERH*YP/VISCOS
  ON 210 I=1,NJMI
  SORTK=0FNI(1,J)
  DENU=0.5*IDEN(1,J)*FNI(1,J)
  YPLUSN(1)=YPLUSN(1)+YPLUSN(1-1)
  IF(YPLUSN(1).GE.11.63) GO TO 211
  TMULT=DENU*CDTERH*CAPPA/ALOG(ELLOG+YPLUSN)
  GO TO 212
  211 TMULT=VISCOS/YP
  212 CONTINUE
  210 AN(1,J)=0.0
  C-----SIDE WALL
  I=2
  DO 205 J=2,NJMI
    AREA=8*CV(1)*SIN(1/J)
    CH=DEH(1,J)+U(2,J)*AREA
    DM=V(1,J)+AREA*DXPH(1,J)
    DN=AREA*DM*FNI(1,J)*CH*(1.0-HEW(1,J)*CM)
    A(1,J)=DM*(1.0-HEW(1,J)*CM)
    AREA=RV(1,J)*SEW(1,J)
    AREA=PV(1,J)*SEW(1,J)
    GN=0.5*IDEN(1,J)+DEN(1,J-1)*V(1,J)
    GM=DN(1,J)+V(1,J)
    CH=AREA*AN(1,J)+DM*V(1,J)*GN*0.5*SEW(1,J)/SEW(1,J)
    CS=AREA*AS(1,J)+DM*V(1,J)*GN*0.5*SEW(1,J)/SEW(1,J)
    V(1,J)=2.5*V(1,J)+V(1,J)*V(1,J)+V(1,J)*V(1,J)
    V(1,J)=0.2*V(1,J)+V(1,J)*V(1,J)+V(1,J)*V(1,J)
    DN=V(1,J)*AREA/DM*V(1,J)
    DS=V(1,J)*AREA/DM*V(1,J)
    AN(1,J)=AREA*(AS(1,J)+U(1,J)*0.4)-0.5*CN
    A(1,J)=AREA*(AS(1,J)+U(1,J)*0.2)*CS
    IF(A(1,J).GT.0) A(1,J)=0.0
  205 CONTINUE
  C-----SYMMETRY AXIS
  DO 203 J=2,NJMI
    AS(1,J)=0.0
  203 AS(1,J)=0.0
  2150 U(1,J)=U(1,J)
  C-----OUTLET
  APDENT=0.0
  FLOW=0.0
  DO 204 J=2,NJMI
    IF(U(1,J).GT.0) A(1,J)=0.5*IDEN(1,J)+9.0
    APDENT=APDENT+APDENT
    FLOW=FLOW+APDENT*U(1,J)
    U(1,J)=FLOW/(FLOW+APDENT)
  204 U(1,J)=U(1,J)
  219 U(1,J)=U(1,J)

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PROMOD 20
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PROMOD 76

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SUBROUTINE PROMOD      74.7.75  0210 TRACE      FFN 4.0+20      05/19/81  17.05.45
C-----
ENTRY MODTCL
TOP CALL
CDTERM=CMU*0.25
VI=V(INJ)-V(INJ1)
J=NJNL
DO CLC 1=2,NJ1
  DENV=DEMT(1)
  SORTK=SORTE(F(I,J))
  VOL=AC(J)+SNS(J)+SEM(I)
  YPLUSN(I)=JFNU+SORTK+CDTERM*YP/VISCOS
  YPLUSA=YPLUSN(I)
  IF(YPLUSA.LE.11.63) GO TO 809
  TMULTW(I)=DENURCDE*P*SORTK+CAPP/ALOG(ELUG*YPLUSA)
  GO TO 808
809
  TMULTW(I)=VIS(I,J)/YP
  TAURU(I)=TMULTW(I)*.5*(U(I,J)+U(I+1,J))
  TAUN(I)=TAURU(I)
  GENC0=ABS(TAUN(I))*2.5*(U(I,J)+U(I+1,J))/YP
  DUDY=(U(I,J)+U(I+1,J))/2*(V(I,J)+V(I+1,J))/4.-(U(I,J)+U(I+1,J))+
  1001-1+U(I+1,J))/4.)/SNS(I)
  GENPES=GEN(I,J)-VIS(I,J)+DUDY**2
  IF(ASHO.EQ.0)GO TO 515
  TAUN(I)=TMULTW(I)*W(I,J)
  GENC0=GENC0+ABS(TAUN(I))*2.5*(U(I,J)+U(I+1,J))/YP-ABS(TAUN(I))*W(I,J)/R(J)
  DUDY=(U(I,J)+U(I+1,J))/2*(V(I,J)+V(I+1,J))/4.-(U(I,J)+U(I+1,J))+
  W(I,J)-W(I+1,J))/2
  GENPES=GENPES-VIS(I,J)+DUDY*WDR**2
  TAUN(I)=SORTK*TAURU(I)+2*TAUN(I)**2
  GENC(I,J)=GENPES*GENC0
515 CONTINUE
  YPLUSN(I)=LE.11.63) GO TO 811
  D1TERM=DEMT(I,J)*(CMU*0.75)*SQRTK*ALOG(ELUG*YPLUSN(I))/(CAPP*YP)
  GO TO 612
611 CONTINUE
  D1TERM=DEMT(I,J)*(CMU*0.75)*SORTK*YPLUSN(I)/YP
612 CONTINUE
  SP(I,J)=GEN(I,J)+VOL+SURO(I,J)
  SP(I,J)=D1TERM*VOL+SURO(I,J)
610 AN(I,J)=G.J
C-----
XP=XTSTEP)-XU(ISTEP)
I=1,151P
DO 620 J=1,NJZL+1
  DENV=DEMT(J)
  SORTK=SORTE(F(I,J))
  VOL=AC(J)+SNS(J)+SEM(I)
  XPLUSN(J)=DENV+SP(I)+CDTERM*XP/VISCOS
  XPLUSA=XPLUSN(J)
  IF(XPLUSA.LE.11.63) GO TO 817
  TMULTW(J)=JFNU+CDTERM*SORTK+CAPP/ALOG(ELUG*XPLUSA)
  GO TO 818
817
  TMULTW(J)=VIS(I,J)/XP
  TAURU(J)=TMULTW(J)*.5*(V(I,J)+V(I+1,J))
  GENC0=ABS(TAURU(J))*2.5*(V(I,J)+V(I+1,J))/XP
  DUDY=(V(I,J)+V(I+1,J))/2*(V(I,J)+V(I+1,J))/4.-(V(I,J)+V(I+1,J))+
  11-1+V(I+1,J))/4.)/SEM(I)
  GENPES=GEN(I,J)-VIS(I,J)+DUDY**2
  TAUN(J)=TMULTW(J)*W(I,J)
  GENC0=GENC0+ABS(TAUN(J))*2.5*(V(I,J)+V(I+1,J))/XP-ABS(TAUN(J))*W(I,J)/R(J)
  DUDY=(V(I,J)+V(I+1,J))/2*(V(I,J)+V(I+1,J))/4.-(V(I,J)+V(I+1,J))+
  W(I,J)-W(I+1,J))/2
  GENPES=GENPES-VIS(I,J)+DUDY*WDR**2
  TAUN(J)=SORTK*TAURU(J)+2*TAUN(J)**2
  GENC(J,J)=GENPES*GENC0
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SUBROUTINE PROPGD      747.75      DT=0.1PAC      05/14/81      12.05.45
400      C-----SIDE WALL
      XP=X(ISTEP)-XHT(STEP)
      L=15*STEP
      TERM=ICMU*(.75)/(CAPPA*XP)
      DT727=J*JNMLE-NMI
      SU(1,J)=GREAT*TERM*TF(1,J)*1.5
      SP(1,J)=GREAT
405      CONTINUE
      DO 1205 J=2,NMI
      AREA=XCV(J)*SC(J)
      CM=DIS(1,J)*OUT2*AREA/OUTPW(2)
      DM=VIS(1,J)*POPC*AREA/OUTPW(2)
      AM(2,J)=AMAX1(ABS(.5*CM),DM)*.5*CM
      IF (J.GT. JNMLE) AM(2,J)=0.0
410      CONTINUE
      XP=X(NMI)-X(NIEMI)
      TERM=ICMU*(.75)/(CAPPA*XP)
      IF (JOUT.EQ.NMI) GO TO 700
      DO 1206 J=INUT,NMI
      SU(1,J)=GREAT*TERM*TF(1,J)*1.5
      AL(1,J)=0.0
      SP(1,J)=GREAT
415      CONTINUE
      DO 700 SYMMETRY AXIS
      DD 730 I=2,NMI
      DD 11 I=DD(1,2)
      AN(1,NMI)=0.0
      AS(1,2)=0.0
      RETURN
420      C-----
425      C-----
430      C-----
      ENTHALPY
435      ENTRY MODH
      CDTERM=CMH*0.25
      YP=YVINJ)-YINJ(1)
      XP=X(2)
      PFUN=0.2*(PRLAM/PPH-1.0)*(PRH/PRLAM)*0.25
      IF (IAUN.E3.0) GO TO 902
      J=NJML
      DO 1902 I=2,NMI
      AN(1,I)=0.
440      CONTINUE
      IF (IAUN.EQ.1) GO TO 904
      J=NJML
      DO 903 I=2,NMI
      AN(1,I)=0.
      AREA=V(J)*SEAL(1)
      GAMMA(1)=VISCOS/PRLAM/YP
      YPLUS=VPLUS(1)
      IF (YPLUSA.L.11.5) GO TO 1903
      UPLUS=ALGCTC*G*YPLUSAL/CAPPA
      GAMMA(1)=G*UPLUS/J*CHEN*SQRT(TE(1,J))/(PRH*(UPLUS*PFUN))
      IF (ICUF.E3.0) GO TO 1710
      H(1,J)=H(1,J)+GADU/GAMMA(1)
445      1902
      902
450      903
455      904

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PRUMOD 362
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	SUBROUTINE	PRGMOD	74/17	DT-3	TPACH	FIN	4.3*528	05/18/81	17.05.65
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T(I,J)=I(I,J+1)/CPI
TVALN(I)=T(I,J+1)
CONTINUE
1710
1703 TIRM=GAMMA(I)*AREA
      SUC(I,J)=SU(I,I)*TER4*(I,J+1)
      SP(I,J)=SP(I,J)-TER4
903 CONTINUE
908 IF(IABN.EQ.0)GO TO 904
      I=2
      DO 1904 J=2,NM1
        IF(IJ.LS.JNDZ)GO TO 1904
        AM(I,J)=C.
        IF(IABN.EQ.1)GO TO 907
        I=2
        DO 905 J=2,NJ1
          AM(I,J)=0.
          AREA=RCJ)*SNS(I)
          GAMMA(I)=VISCIS/PRLAM/KP
          XPLUSA=XPLUSW(I)
          IF(XPLUSA.LE.11.63)GO TO 1905
          UPLUS=ALOG(FCIG*XPLUSA)/CAPPA
          GAMMA(I)=DEHT(I)*CONERN*SOKT(ITE(I,J))/(PRH*(UPLUS+PFUN))
1905 CONTINUE
      TIRM=GAMMA(I)*AREA
      SUC(I,J)=SU(I,I)*TER4*(I-L,J)
      SP(I,J)=SP(I,J)-TER4
905 CONTINUE
907 DO 906 I=2,N1
      H(I,1)=H(I,2)
      AS(I,2)=0.
906 CONTINUE
908 END
  
```

[illegible]

A 10x10 grid of 100 small squares, each containing a single digit from 0 to 9. The digits are arranged in a pattern that resembles a random distribution, with some digits appearing more frequently than others. The grid is used for a statistical exercise.

[illegible][illegible]

220 140 110 100 90 80 70 60 50 40 30 20 10

[illegible]

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ARIZONA UNIV TUCSON ENGINEERING EXPERIMENT STATION
CONVECTIVE HEAT TRANSFER FOR SHIP PROPULSION.(U)
APR 81 M A HABIB, D M MCLEIGOT

F/O 20/13

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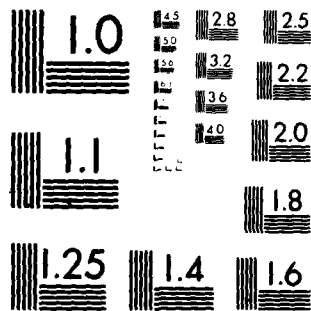
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MICROCOPY RESOLUTION TEST CHART
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[illegible]

A 10x10 grid of 100 small, stylized, abstract shapes arranged in a regular pattern. The shapes are composed of various geometric elements like lines, dots, and curves, creating a complex, textured appearance. The overall effect is a dense, repeating pattern of intricate, hand-drawn-like motifs.

A series of seven rows of repeating patterns made of small black dots and dashes, arranged horizontally.

A series of seven rows of repeating patterns of letters and symbols:

- Row 1: A sequence of 'N' characters.
- Row 2: A sequence of 'O' characters.
- Row 3: A sequence of '+' characters.
- Row 4: A sequence of 'W' characters.
- Row 5: A sequence of 'f' characters.
- Row 6: A sequence of 't' characters.
- Row 7: A sequence of 'e' characters.

A series of seven rows of repeating patterns made of small black marks on a white background. The patterns include horizontal lines of dots, vertical bars, and various geometric shapes like circles and triangles.

A series of seven rows of symbols, including letters, numbers, and special characters, arranged in a grid-like pattern. The symbols are:

 Row 1: N N N N N N N N N N N N N N N N N N

 Row 2: 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0 0

 Row 3: + + + + + + + + + + + + + + + +

 Row 4: W W W W W W W W W W W W W W W W

 Row 5: C C C C C C C C C C C C C C C C

 Row 6: - - - - - - - - - - - - - - - -

 Row 7:

NNNNNNNNNNNNNNNNNNNNN
 0000000000000000000000
 ++++++
 WWWWWWWWWWWWWWWWWWWWW
 NNNNNNNNNNNNNNNNNNNNN
 ~~~~~~  
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[illegible]

A series of five rows of black dots arranged in a grid-like pattern, likely representing a barcode or a data matrix.

[illegible]

**PAGE TWO**

[illegible]

A 10x10 grid of 100 small circles, arranged in 10 rows and 10 columns. The circles are black and white, forming a pattern that resembles a barcode or a stylized letter 'E'.

210 07 05 32  
211 10 11 10  
212 11 12 11  
213 12 13 12  
214 13 14 13  
215 14 15 14  
216 15 16 15  
217 16 17 16  
218 17 18 17  
219 18 19 18  
220 19 20 19

**SECRET**

[illegible]

TEMPERATURE

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2

[illegible]

S.S. COEFF.

XU(1)

|    |           |    |    |    |           |
|----|-----------|----|----|----|-----------|
| 2  | 7.428E-02 | 0. | 0. | 0. | 1.052E+01 |
| 3  | 2.228E-01 | 0. | 0. | 0. | 1.349E+01 |
| 4  | 3.714E-01 | 0. | 0. | 0. | 1.930E+01 |
| 5  | 5.943E-01 | 0. | 0. | 0. | 2.663E+01 |
| 6  | 8.914E-01 | 0. | 0. | 0. | 3.051E+01 |
| 7  | 1.189E+00 | 0. | 0. | 0. | 3.658E+01 |
| 8  | 1.634E+00 | 0. | 0. | 0. | 4.297E+01 |
| 9  | 2.228E+00 | 0. | 0. | 0. | 5.963E+01 |
| 10 | 2.971E+00 | 0. | 0. | 0. | 5.497E+01 |
| 11 | 3.863E+00 | 0. | 0. | 0. | 5.811E+01 |
| 12 | 5.051E+00 | 0. | 0. | 0. | 7.868E+01 |
| 13 | 6.537E+00 | 0. | 0. | 0. | 5.627E+01 |
| 14 | 8.320E+00 | 0. | 0. | 0. | 5.067E+01 |
| 15 | 1.040E+01 | 0. | 0. | 0. | 6.305E+01 |
| 16 | 1.293E+01 | 0. | 0. | 0. | 3.584E+01 |
| 17 | 1.590E+01 | 0. | 0. | 0. | 3.078E+01 |
| 18 | 1.931E+01 | 0. | 0. | 0. | 2.798E+01 |
| 19 | 2.303E+01 | 0. | 0. | 0. | 2.653E+01 |
| 20 | 2.748E+01 | 0. | 0. | 0. | 2.564E+01 |
| 21 | 3.224E+01 | 0. | 0. | 0. | 2.490E+01 |
| 22 | 3.714E+01 | 0. | 0. | 0. | 2.415E+01 |
| 23 | 4.308E+01 | 0. | 0. | 0. | 2.341E+01 |
| 24 | 5.051E+01 | 0. | 0. | 0. | 2.277E+01 |
| 25 | 5.943E+01 | 0. | 0. | 0. | 2.230E+01 |
| 26 | 6.834E+01 | 0. | 0. | 0. | 2.202E+01 |
| 27 | 7.428E+01 | 0. | 0. | 0. | 2.192E+01 |

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| NO. | X/H      | HEAT FLUX | HEAT TRANSFER COEFF. | NUSEFF NUMBER | NORM. NU.ND. |
|-----|----------|-----------|----------------------|---------------|--------------|
| 2   | .743E+01 | .130E+03  | .543E+02             | .495E+02      | .528E+00     |
| 3   | .223E+00 | .178E+03  | .690E+02             | .630E+02      | .670E+00     |
| 4   | .371E+00 | .291E+03  | .101E+03             | .92E+02       | .993E+00     |
| 5   | .594E+00 | .420E+03  | .133E+03             | .121E+03      | .129E+01     |
| 6   | .891E+00 | .579E+03  | .169E+03             | .154E+03      | .164E+01     |
| 7   | .119E+01 | .763E+03  | .206E+03             | .188E+03      | .200E+01     |
| 8   | .163E+01 | .101E+04  | .249E+03             | .227E+03      | .242E+01     |
| 9   | .223E+01 | .130E+04  | .294E+03             | .268E+03      | .286E+01     |
| 10  | .297E+01 | .167E+04  | .337E+03             | .307E+03      | .320E+01     |
| 11  | .386E+01 | .189E+04  | .365E+03             | .335E+03      | .355E+01     |
| 12  | .505E+01 | .192E+04  | .379E+03             | .345E+03      | .368E+01     |
| 13  | .654E+01 | .186E+04  | .377E+03             | .343E+03      | .366E+01     |
| 14  | .832E+01 | .199E+04  | .355E+03             | .324E+03      | .345E+01     |
| 15  | .104E+02 | .178E+04  | .317E+03             | .288E+03      | .308E+01     |
| 16  | .129E+02 | .172E+04  | .277E+03             | .253E+03      | .269E+01     |
| 17  | .159E+02 | .210E+04  | .245E+03             | .223E+03      | .238E+01     |
| 18  | .193E+02 | .242E+04  | .223E+03             | .203E+03      | .217E+01     |
| 19  | .230E+02 | .259E+04  | .206E+03             | .188E+03      | .200E+01     |
| 20  | .275E+02 | .319E+04  | .192E+03             | .175E+03      | .187E+01     |
| 21  | .322E+02 | .330E+04  | .178E+03             | .162E+03      | .173E+01     |
| 22  | .371E+02 | .328E+04  | .163E+03             | .148E+03      | .158E+01     |
| 23  | .431E+02 | .306E+04  | .148E+03             | .135E+03      | .144E+01     |
| 24  | .509E+02 | .287E+04  | .137E+03             | .125E+03      | .133E+01     |
| 25  | .594E+02 | .275E+04  | .129E+03             | .118E+03      | .126E+01     |
| 26  | .683E+02 | .271E+04  | .126E+03             | .115E+03      | .123E+01     |
| 27  | .743E+02 | .277E+04  | .126E+03             | .115E+03      | .122E+01     |

| NO. | IBULK     | IDIFF     |
|-----|-----------|-----------|
| 2   | .3097E+03 | .2398E+01 |
| 3   | .3097E+03 | .2985E+01 |
| 4   | .3097E+03 | .2873E+01 |
| 5   | .3097E+03 | .3154E+01 |
| 6   | .3098E+03 | .3429E+01 |
| 7   | .3098E+03 | .3704E+01 |
| 8   | .3098E+03 | .4066E+01 |
| 9   | .3099E+03 | .4416E+01 |
| 10  | .3099E+03 | .4953E+01 |
| 11  | .3100E+03 | .5177E+01 |
| 12  | .3101E+03 | .5076E+01 |
| 13  | .3102E+03 | .4951E+01 |
| 14  | .3104E+03 | .5600E+01 |
| 15  | .3106E+03 | .5624E+01 |
| 16  | .3108E+03 | .6210E+01 |
| 17  | .3110E+03 | .8558E+01 |
| 18  | .3113E+03 | .1087E+02 |
| 19  | .3116E+03 | .1255E+02 |
| 20  | .3120E+03 | .1638E+02 |
| 21  | .3124E+03 | .1857E+02 |
| 22  | .3128E+03 | .2016E+02 |
| 23  | .3133E+03 | .2066E+02 |
| 24  | .3140E+03 | .2103E+02 |
| 25  | .3147E+03 | .2127E+02 |
| 26  | .3155E+03 | .2152E+02 |
| 27  | .3160E+03 | .2201E+02 |
|     |           | .3121E+03 |
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|     |           | .3380E+03 |

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