the adiabatic efficiency which is considered to closely approximate the true performance because the delivery pressure in the LN_2 tests was less than 16 MPa. With regard to LH_2, not only the adiabatic efficiency, but also the polytropic efficiency, is presented in Fig. 4. The polytropic efficiency fairly well agreed with the head and efficiency obtained in the LN_2 test. Based on the above mentioned fact, the polytropic efficiency can be considered to show nearly true pump head and efficiency.

The efficiency of about 80% at the designed flow ratio ($Q/Q_d = 1.0$) with the LOX pump mentioned above is considered to be fairly high compared with those of the previously developed rocket pumps as shown in Fig. 5.[5] Figure 5 shows the stage efficiency of the rocket pumps with the main impeller, the diameter of which was more than 195 mm. The efficiency of about 75% obtained with the present LH_2 pump can also be considered as reasonable as shown in Fig. 5.

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Numerical Prediction of a Turbulent Evaporating Fuel Spray in a Recirculating Flow

Xi-Qing Chen* and José Carlos Fernandes Pereira†
Instituto Superior Técnico/Technical University of Lisbon, 1096 Lisbon Codex, Portugal

Introduction

INVESTIGATION of turbulent evaporating sprays is vitally important in various industrial applications, such as industrial combustors, gas turbines, etc. With the advent of high-speed and sophisticated computers, numerical modeling has been playing a more and more important role. Up to now, various mathematical and physical models have been developed to predict the flow characteristics of sprays. Due to the intrinsic existence of potential multivaluedness arising as droplet trajectories cross, the Lagrangian separated-flow methods are widely used. Currently used spray combustion models have been reviewed.[1–5]

The Lagrangian approach[6] treats the dispersed phase by representing different droplet "parcels," each of which is assumed to possess the same size range, diameter, and velocity. The droplet motion is governed by the Lagrangian form of transport equations. In tracking each parcel of droplets, the loss or gain of droplet mass, heat, and momentum within each Eulerian cell can be obtained, which will be used as the coupling source or sink terms in the corresponding finite difference equations for the gas phase. Lagrangian treatment of droplet transport possesses the advantages of no numerical diffusion and easy treatment of multivaluedness. But if this approach is used to track the droplet motion in a deterministic way, the advantage of no numerical diffusion will turn out to be its disadvantage as well, which gives a discernible underprediction of droplet dispersion. Confronted with this problem, many researchers are presently employing the stochastic separated-flow (SSF) method to treat the gas-turbulence influence on the droplet motion through particle-eddy interactions.[7] This method has wide applications and gives encouraging results in a variety of flow situations,[8–10] including flows with and without evaporation.

While the SSF models have found wide acceptance and been applied with great success in most spray combustion modeling, they usually neglect the fact that turbulent velocity fluctuations are correlated temporally and directionally, and vanish only when the time elapsed is large and cross-correlations are zero. To account for this physical phenomenon in practical spray combustion modeling, Berlemont et al.[11] and Zhou and Leschziner[12] have recently proposed two similar models to incorporate the turbulent temporal correlations and anisotropy, both of which share the same philosophy, but start from different routes.

In this Note, a comprehensive spray evaporation model, based on an Eulerian model of the gas field and a Lagrangian model of the droplet field in conjunction with the stochastic description of gas turbulence effect on the droplet motion, is applied to a turbulent evaporating spray in a recirculating flow and validated by comparison between predictions and measurements. Unlike many previous numerical predictions, this Note has been able to avoid the usual problem of a lack of detailed initial droplet-size and velocity-distribution conditions, and incorporated the turbulent temporal and directional correlations. In the present study, we have adopted Zhou and Leschziner's methodology to include turbulent temporal and directional correlations in the numerical modeling, which has proved to be an improvement over the conventional particle-eddy modeling in simple flows.[13,14]

Physical Model Description

We consider an evaporating spray of liquid isopropyl alcohol issuing into a coflowing turbulent, heated, airstream. An annular air jet enters the test section of 194-mm i.d. The inlet air and liquid isopropyl alcohol temperatures are 353 and 304 K, respectively. The detailed droplet-size and velocity distribution data were measured with phase-Doppler by Sommerfeld.[15]

Governing Equations for the Gas Field and Droplet Field

The equations for the continuous gas field are based on the modified two-equation turbulent eddy-viscosity model by incorporating two-phase coupling source terms. Application of the Reynolds time-averaging process to the Eulerian form of conservation equations for each dependent variable results in the mean flow equations for all dependent variables.

The droplet field is calculated by tracking the droplet parcels throughout the computational domain. The initial droplet-size distribution of the spray is divided, according to the given experimental probability density function (PDF), into an adequate number of discrete parcels, each of which represents a set of droplets belonging to the same size range and possessing the same initial conditions. Due to the small ratio

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*Ph.D. Student, Mechanical Engineering Department, Av. Rovisco Pais.

†Associate Professor, Mechanical Engineering Department, Av. Rovisco Pais. Member AIAA.

of the gas density to the droplet density, only drag force was considered in writing the equations of motion for each droplet parcel. The details of formulating the gas- and droplet-phase equations can be found elsewhere.[10]

Turbulent Temporal and Directional Correlations

The Lagrangian stochastic model treats the droplet dispersion by applying the instantaneous gas velocity in the droplet equations of motion through the concept of droplet-eddy interactions.[7] The interaction time is determined by minimizing two time scales, the eddy life time and eddy transit time. The instantaneous gas velocity is determined by summing its mean value, which is obtained by solving the gas-phase mean flow equations, over a fluctuating value, which is obtained by randomly sampling a Gaussian PDF with a standard deviation of $\sqrt{2k/3}$. Here k is the turbulent kinetic energy.

Although there are a variety of methods to prescribe the droplet-eddy interaction time, they almost all neglect the turbulent temporal and directional correlations existing in turbulent flows. This approximation is not adequate because turbulent fluctuations are correlated in both time and direction. The correlations vanish only when a large time step is used and cross-correlations are zero. Recognizing this deficiency of conventional modeling, Berlemont et al.[11] and Zhou and Leschziner[12] have recently proposed two similar methodologies to incorporate the temporal and directional correlations. They are different in that the Berlemont's method considers temporal and directional correlations among many time steps, while the Zhou and Leschziner's method considers the temporal and directional correlations only between two successive time steps. In this Note, we follow Zhou and Leschziner's methodology to account for the temporal and directional correlations of turbulent fluctuations. The detailed description of the methodology can be found elsewhere.[12]

Droplet Evaporation Model

In the spray considered here, droplet evaporation with an initial temperature of 304 K occurs in a lower temperature environment. The maximum inlet gas temperature is 353 K, while the droplet boiling temperature at the atmospheric pressure is 355 K. The evaporation is mass transfer controlled. Therefore, it is adequate to consider the droplet preheating process during its evaporation. Among the various evaporation models accounting for the droplet transient heat transfer, the simplest one may be the infinite-conduction evaporation model. Other evaporation models, i.e., the conduction-limit model, the vortex model, etc., are more sophisticated, where the droplet temperature varies not only temporally along its trajectory but also spatially inside the droplet itself.

In this Note, the infinite-conduction evaporation model is selected owing to its simplicity and inclusion of transient heat transfer. Droplet mass evaporation rate was calculated by the classical d^2 law.[16] To take into account the convective effect of the gas phase on droplet evaporation, the Nusselt and Sherwood numbers were modified using the correlations.[3] In the calculations of Schmidt and Prandtl numbers, etc., all the gas-film thermophysical properties except the gas density, which is evaluated by the freestream value, are evaluated by using the $\frac{1}{3}$-rule.

The Lagrangian stochastic model requires tracking a relatively large number of trajectories to obtain the stochastically significant averages. In the study, 3000 initial droplet trajectories are tracked. Solution to the droplet field will give the history of the location, velocity, size, and temperature for each droplet parcel.

Numerical Solution Procedure

In this study, a high-order accurate convection-discretizing scheme (QUICK)[17] was used for the gas-phase solution. Due to the different locations existing in the Eulerian and Lagrangian calculations, two different interpolations are carried out in this Note. In the Lagrangian calculations, the gas-phase properties are interpolated from the Eulerian locations to the Lagrangian locations by a second-order-accurate scheme. In the Eulerian calculations, the droplet source or sink terms are distributed from the Lagrangian locations to the Eulerian locations in terms of the droplet residence time in the Eulerian cell crossed.

Results and Comparisons

Before calculating the complex turbulent evaporating spray, we have tested the necessity and feasibility of incorporating the turbulent temporal correlations through a simple two-phase nonevaporating flow. To this end, the measurements (labeled as L50) of Vames and Hanratty[18] were chosen. The gas-phase data (i.e., the mean velocity profile) and the turbulent properties were prescribed using the measurements of Laufer[19] for the fully developed pipe flow. Shown in Fig. 1 is the comparison of radial mean-squared dispersion, based on an average of 3000 droplet trajectories, between the measurements and predictions with and without turbulent temporal correlations.

Figure 1 indicates that the temporal correlations should be taken into account after a certain time (9 ms for this problem) in order to predict the mean-squared displacement. Earlier than 9 ms, there is little difference between the dispersions with and without temporal correlations. This phenomenon is consistent with the conclusion drawn by Friedlander[20] that the initial spreading of particles should depend little on the correlation coefficient, but much on the intensity of turbulence. This simple test case shows that the turbulent temporal correlations should be accounted for in modeling the droplet-phase dispersion.

For the present developing flow, an axial length of 1.3 m is chosen to make the computational domain bounded longitudinally. The calculations are based on a grid of 60 × 53 in the axial and radial directions, respectively, the choice of which is in terms of numerical experiments. This grid yields similar results to those obtained using 85 × 70 control volumes.

Figure 2 shows the mean axial droplet velocity profiles at different axial locations. The first profile at $X = 25$ mm shows very good agreement with the measurements due to the prescribed inlet measurements at $X = 3$ mm. Further downstream, the predictions continue to show excellent agreement with the measurements.

Figure 3 compares the predicted mean-droplet-diameter profiles with those measured experimentally. In the first two axial locations, $X = 25$ mm and 50 mm, the mean diameter around the centerline is slightly overpredicted, indicating that there are more small droplets going into the central recirculating region due to their lower inertias. When the flow recirculation dies out, the prediction improves. This can be observed from the first three axial locations. The largest discrepancies are in the last profile where the experimental data show a uniform distribution, while the predictions are nonuniform, either due to the presence of smaller number of total

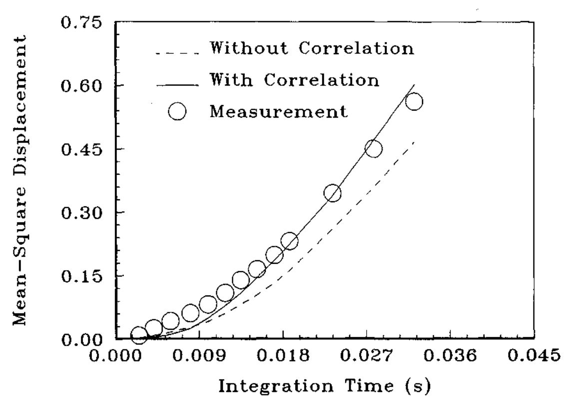

Fig. 1 Comparison of measured and predicted radial dispersion (cm²).

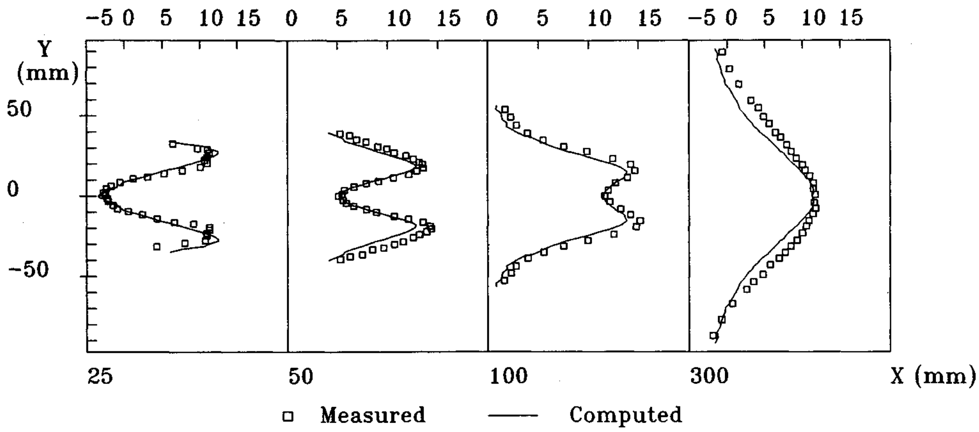

Fig. 2 Comparison of the mean axial droplet velocity (m/s).

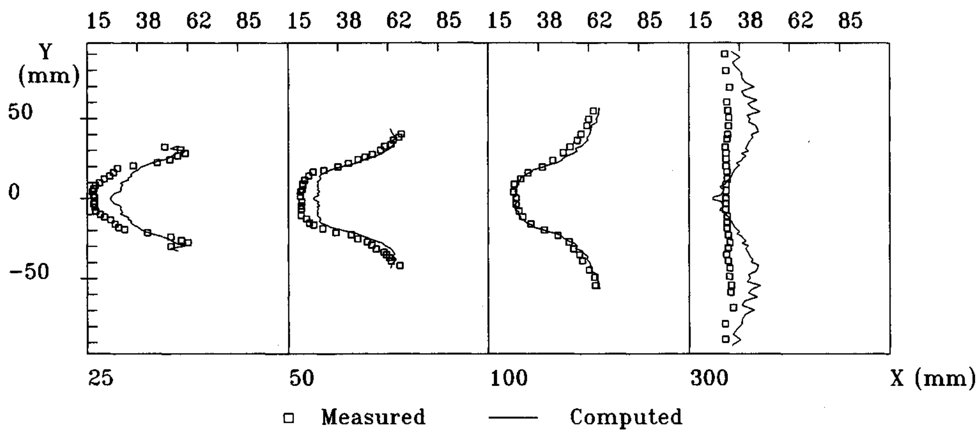

Fig. 3 Comparison of the mean droplet diameter (μm).

droplets resulting from evaporation or as a direct consequence of the prediction of mean radial velocities at this location. Similar good agreement with the data can also be observed for the droplet mass fluxes.

Conclusions

A comprehensive spray model is applied into a turbulent evaporating spray with flow recirculations. The conventional Eulerian-Lagrangian stochastic separated-flow model was improved in the present spray modeling by taking into account the turbulent temporal and directional correlations. The present numerical predictions were carried out on the detailed initial droplet-size and velocity-distribution conditions, thus eliminating the ambiguity of prescribing those conditions in many other predictions. Very good agreement was obtained in the prediction of droplet axial velocities and droplet mean diameters. It is expected that the present Eulerian-Lagrangian SSF model could be further improved by extending it to differential second-order Reynolds stress closures to take advantage of the turbulent anisotropy in directional correlations.

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Effect of Impinging Jet Excitation on Curved Surface Heat Transfer

P. J. Disimile*
University of Cincinnati, Cincinnati, Ohio 45221

Introduction

THE overall efficiency of gas turbine engines is directly dependent upon the turbine inlet temperature of the working fluid. In today's engines, metallurgical limitations restrict the fluid temperature to approximately 1300°C, which is below the adiabatic flame temperature of most hydrocarbon fuels. For over 10 yr, engineers have sought ways to circumvent these design restrictions by developing more advanced methods of turbine blade cooling. One of the more generally accepted methods for enhanced cooling utilizes impinging jet heat transfer. In this approach, highly disturbed fluid emanating from the compressor is transmitted through a complex array of ductwork before impinging onto the blade surface. Yet little attention has been given to the state of the fluid impinging on the internal blade structure.

Previous research has indicated that changes in heat transfer are possible under harmonic excitation. Gutmark et al.[1] demonstrated that low level harmonic excitation of a jet impinging on a flat plate did enhance cooling. However, 18 yr earlier, Nevins and Ball[2] investigated the average heat transfer between a flat plate and a pulsating impinging jet and reached the opposite conclusion. They concluded that, for the range of variables covered, the heat transferred to the jet was independent of both frequency and amplitude of the disturbance. These studies reveal a major difference between an excited jet, in which disturbances have a negligible effect upon the steady mass flow rate, and a pulsating jet in which the mass flow is unsteady.

Time independent studies in the open literature include work by Gardon and Cobonque,[3] who examined the average heat transfer coefficients as well as their variations from point-to-point on a cooled flat surface using impinging jets. Metzer et al.[4] studied the heat transfer between a single jet and a concave cylindrical surface. Tabakoff and Clevenger[5] compared the thermal effectiveness of three different systems of air jets impinging on the inside surface of a half-circular cylinder. Hrycak[6] recently measured average and local heat transfer coefficients from a row of impinging jets to concave cylindrical surfaces.

When examining the fundamental fluid dynamics of the exiting jet, one notes that in the natural (unexcited) free shear layer, the fluid which separates from the nozzle boundary layer tends to "roll up" and form discrete vortices. If periodic excitation is applied to the shear layer, it is found to cause a two-dimensional undulation of the separating boundary layer. This undulation causes the agglomeration of several of these discrete vortices into a large, isolated vortical structure downstream from the separation lip. That is, excitation has been found to organize and generate larger vortical structures than would otherwise exist in a naturally occurring shear layer. Moreover, Hussain[7] demonstrated that under certain conditions excitation can also suppress the turbulent intensity of an axisymmetric jet, extending as far downstream as eight jet diameters. Since there are no experiments that investigated the effects of a simple periodic disturbance on blade cooling, the following program was conducted.

Experimental Setup

The implementation of this experiment required a test facility which would simulate a nominal surface arrangement within the turbine blade. This facility consisted of a 0.180-mm-thick nickel-chromium plate curved to form a long semicylinder with a 12.70-cm arc length which served as the inner blade contour. This contour was mounted in a wooden cell (43.2 cm wide × 44.5 cm deep × 30.5 cm high), and insulated with styrofoam (Fig. 1). The air jet emanated from a rectangular nozzle mounted above the plate with a 3.18-mm width and an aspect ratio of 44. Upstream of the nozzle was a settling chamber containing a 21-cm loudspeaker, oriented in the streamwise direction. This device provided periodic excitation of the exiting flow. The plate was heated by a resistive heating element, providing a constant heat flux per unit area. The plate temperatures were measured by nine miniature, fast responding, copper-constantan (type T) thermocouples located between the plate and the heater. To minimize lead conduction errors, the leads were placed parallel to the axis of the semicylinder. This ensured that the thermal gradient along the leads was approximately zero.

The data taking was carried out using a computer-controlled data acquisition system. Specifically, 1000 samples were acquired for each of the nine thermocouples at time increments of 125 ms. Using the pressure difference obtained across a sharp-edge orifice plate the mass flow rate was calculated.

Results

In the first phase of the experiment, the heat loss through the insulated test cell was determined over the temperature

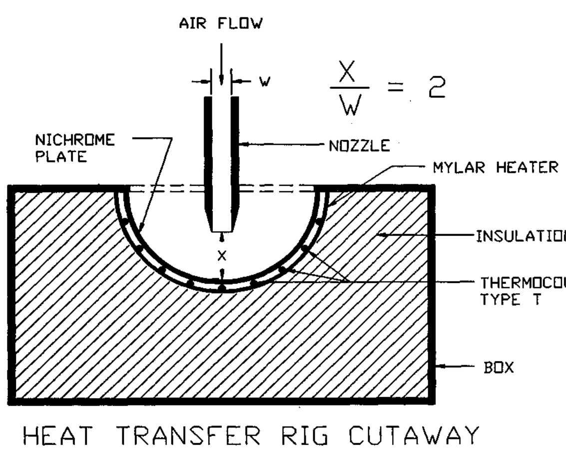

Fig. 1 Schematic of test cell.

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*"The Bradley Jones" Associate Professor, Department of Aerospace Engineering, M.L. #70. Member AIAA.