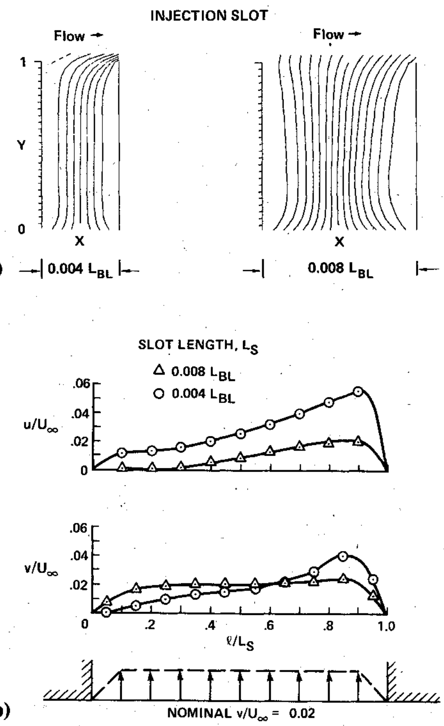
a)



b)

Fig. 3 Injection applied through the bottom of the slot: a) streamline plots for fluid injected into the boundary layer; b) distributions of the two velocity components along the top of the slot for the two slot sizes.

tangential and normal velocity distributions along the open top for the two slots in Fig. 2a. For the shorter slot, the u distribution is flat, while the v distribution shows the largest (negative) velocity close to the upstream corner, which then decreases in the downstream direction. For the larger slot, the u distribution is curved and the maximum value is 2.5 times greater than for the previous case, since the shear layer has more distance over which to develop. For comparison, the maximum values of u indicated in Fig. 2b are twice as large as the corresponding values for the case of the slot closed off at the bottom (the shear-layer-driven cavity).[6] This increase, then, represents the net streamwise acceleration of the lower part of the shear layer due to the fluid withdrawn at the bottom of the slot.

Figure 3 addresses the problem of injecting air through the slot into the boundary layer. The streamline plots (Fig. 3a) show that the injected fluid turns much more severely upon merging with the boundary layer for the case of the smaller slot. Examination of the velocity component distributions at the interface (Fig. 3b) lends further support for that statement. For the larger slot, u is practically zero over a third of the span, and substantially less over the whole span than the corresponding values for the smaller slot. Again, comparing these results with those for slots closed off at the bottom,[6] it is found that injecting through the larger slot substantially retards the bottom portion of the shear layer, while injecting through the smaller slot initially retards it but substantially accelerates it over the downstream third of the span. It is appropriate to note that (as in the case of suction) the rate of mass transfer per unit span of the slot is the same for the two slot sizes. The distributions of v velocity in Fig. 3b again show near uniformity for the larger slot, while for the shorter slot v increases in the downstream direction and peaks close to the downstream corner.

Discussion and Conclusion

The results presented in the previous section indicate that the influence of the injected (or withdrawn) fluid on the shear layer development in the vicinity of a slot is significant. Both u and v velocity components are present at the interface. The presence of the u velocity component (usually neglected in analysis of boundary layers with mass transfer at the wall) can have a significant influence on the development of the boundary layer, affecting the ability of the boundary layer to separate or stay attached. It also affects the displacement thickness, thus influencing the interaction of the boundary layer with the (inviscid) freestream.

The flowfield in the interface region was seen to be affected by the slot size. The larger slot was more effective in accelerating the bottom portion of the shear layer when in the suction mode and in blowing off the shear layer when in the injection mode. The slot efficiency could be improved by rounding off the corners and by appropriately inclining the slot with respect to the flow.[8] Thus recirculation bubbles such as the one shown in Fig. 2a could be suppressed.

Acknowledgments

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Sensitivity of Chamber Turbulence to Intake Flows in Axisymmetric Reciprocating Engines

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Introduction

In reciprocating internal-combustion engines, combustion controls efficiency and emissions and turbulence controls combustion. Turbulence is generated during intake but there are sources and sinks of it during compression. Since com-

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bustion starts toward the end of the compression stroke, the state of turbulence near top dead center (TDC) is of practical interest.

During intake, a jet issues into the chamber from the valve, or port, opening and sets up a highly nonhomogeneous and nonisotropic flow. If the intake and chamber are purposely designed so as not to generate strong bulk flows, the initial nonuniformities are smoothed out during compression and, when TDC is reached, bulk flows are negligible and turbulence tends to be homogeneous and isotropic except near the walls. That is, the TDC flowfield is insensitive to the details of the intake process. Thus some combustion computations[1-3] are initiated near TDC from spatially uniform turbulent diffusivities that are related only to a few engine parameters, such as speed, load, and compression ratio, with considerable savings in computation time. Although this behavior was suggested by measurements,[4] definite experimental proof has not yet been reported and several researchers have concentrated on the details of the intake flow.[5,6] It would appear that intake details should be important when strong intake bulk flows are present because such flows persist to TDC. But then flows in the intake and exhaust manifolds should also be computed since they influence the intake process and computation times become impractical. Thus the question of the sensitivity of the TDC field to intake conditions remains theoretically and practically important.

Here we summarize results of a theoretical sensitivity study. Details are available in Ref. 7. The model is for unsteady axisymmetric flows and includes a k-ϵ submodel for turbulence. In the first part of this study,[8] we reported its equations, method of solution, and several comparisons with measurements.

Results

A pancake-type combustion chamber and a cup-in-piston flat-head chamber were considered again as in Fig. 1a of Ref. 8. Only one engine speed (rpm = 1700), one load ($\eta_v = 100\%$), and one compression ratio (CR = 21) were used, since the dependency of turbulence on those parameters was investigated in the first study. Initial bulk flows were simulated by imposing an initial solid-body rotation with a swirl ratio (SR) of 10.

The effect of the initial magnitudes of the turbulence intensity k and its rate of decay ϵ on TDC turbulence was investigated by starting the computations at different crankangles θ_i with the same initial values. The effect of initial nonuniformities of k and ϵ on TDC turbulence was studied starting from two coaxial cylindrical regions, which differ in their turbulent kinetic energy and dissipation rate but share the same diffusivity D. This test is identified by 2-4, where 2 refers to the turbulence kinetic energy of the inner region (which extends to 25% of the radius) being two times that of the outer region and 4 refers to the dissipation rate being four times. The initial turbulent diffusivity is still spatially uniform since it goes as k^2/ϵ. The effect of initial nonuniformities in k, ϵ, and D was then investigated in the 3-4 test where the 3 and 4 have the same meaning as 2 and 4 of the previous test. In the 3-4 test the turbulent diffusivity of the inner region is 9/4 times that of the outer region. In all cases k and ϵ were initialized[7,8] by $k = \alpha\eta_v^2$ rpm^2, $\epsilon = \beta\eta_v^3$ rpm^3, $D = 0.09\ \eta_v$ rpm α^2/β, $\alpha = 0.45$, and $\beta = 0.305$. All reported turbulent intensities $[u' = (2k/3)^{1/2}]$ and diffusivities are nondimensionalized by $u'_R = 612$ cm/s and $D_R = 367$ cm^2/s and the radial r and axial z coordinates are in terms of cell numbers (cylinder radius $= R = 6$ cm, cup radius $= R_c = 2.43$ cm).

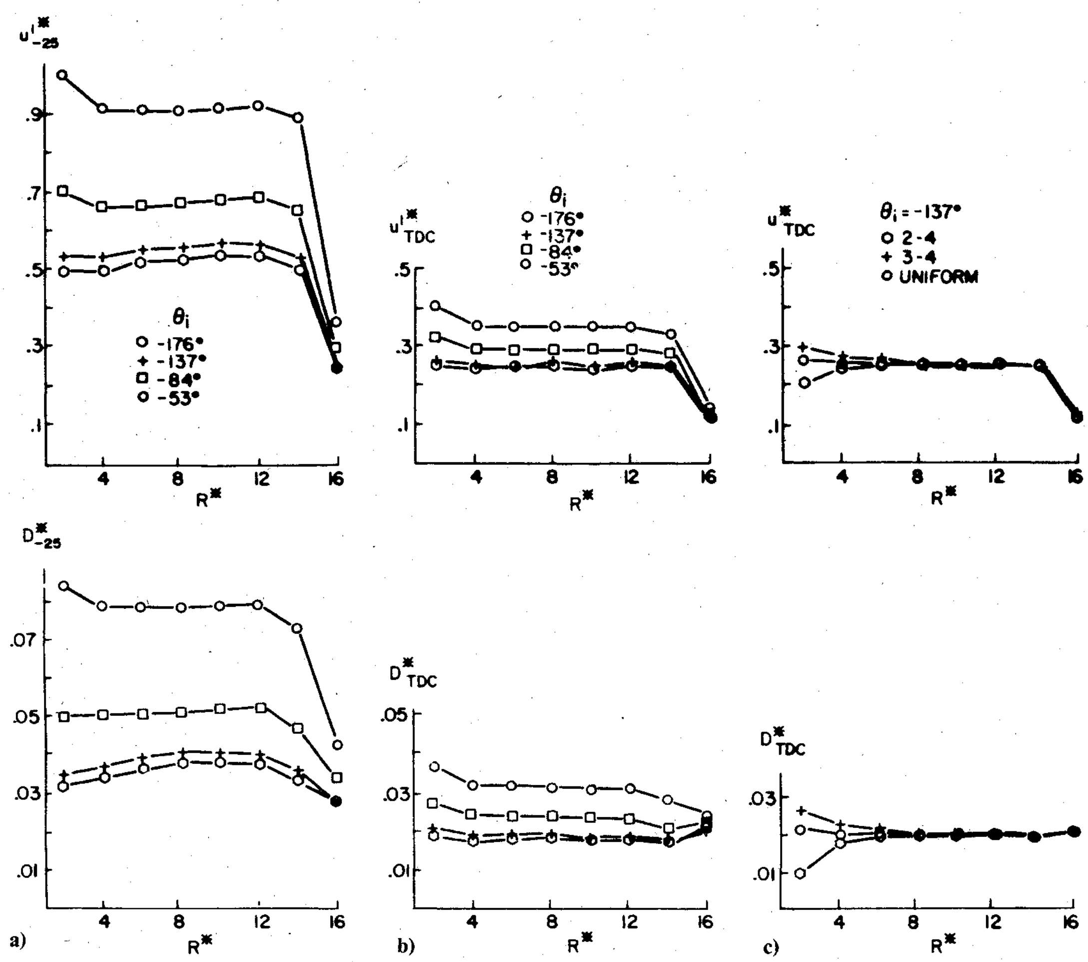

Fig. 1 Pancake chamber, without swirl: effect of initial magnitudes and uniformities of k, ϵ, and D on u' and D at −25 deg and TDC, 3.6 mm below the engine head (SQ = 0, CR = 21, SR = 0, $\eta_v = 100\%$, rpm = 1700).

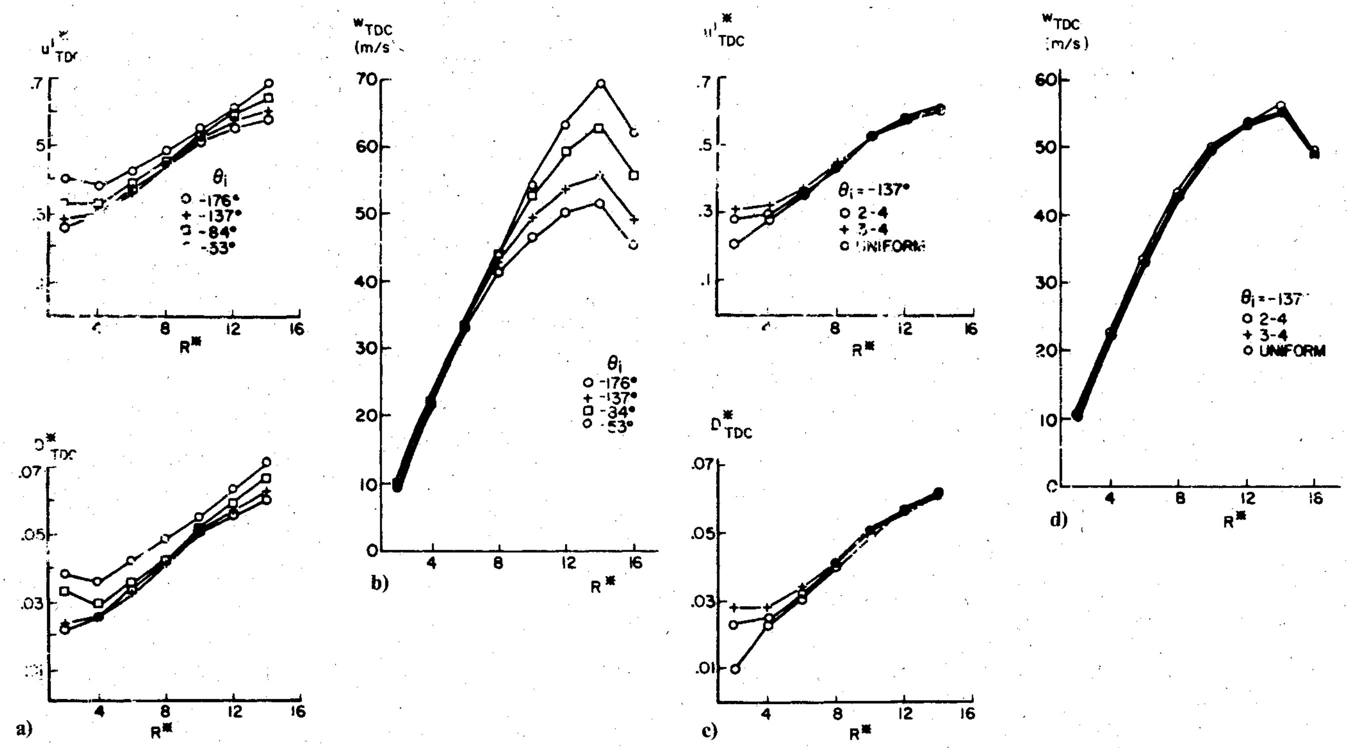

Fig. 2 Pancake chamber with swirl: effect of initial magnitudes and uniformities of k, ϵ, and D on u', and w at TDC, 3.6 mm below the engine head (SQ = 0, CR = 21, SR = 10, η_v = 100%, rpm = 1700).

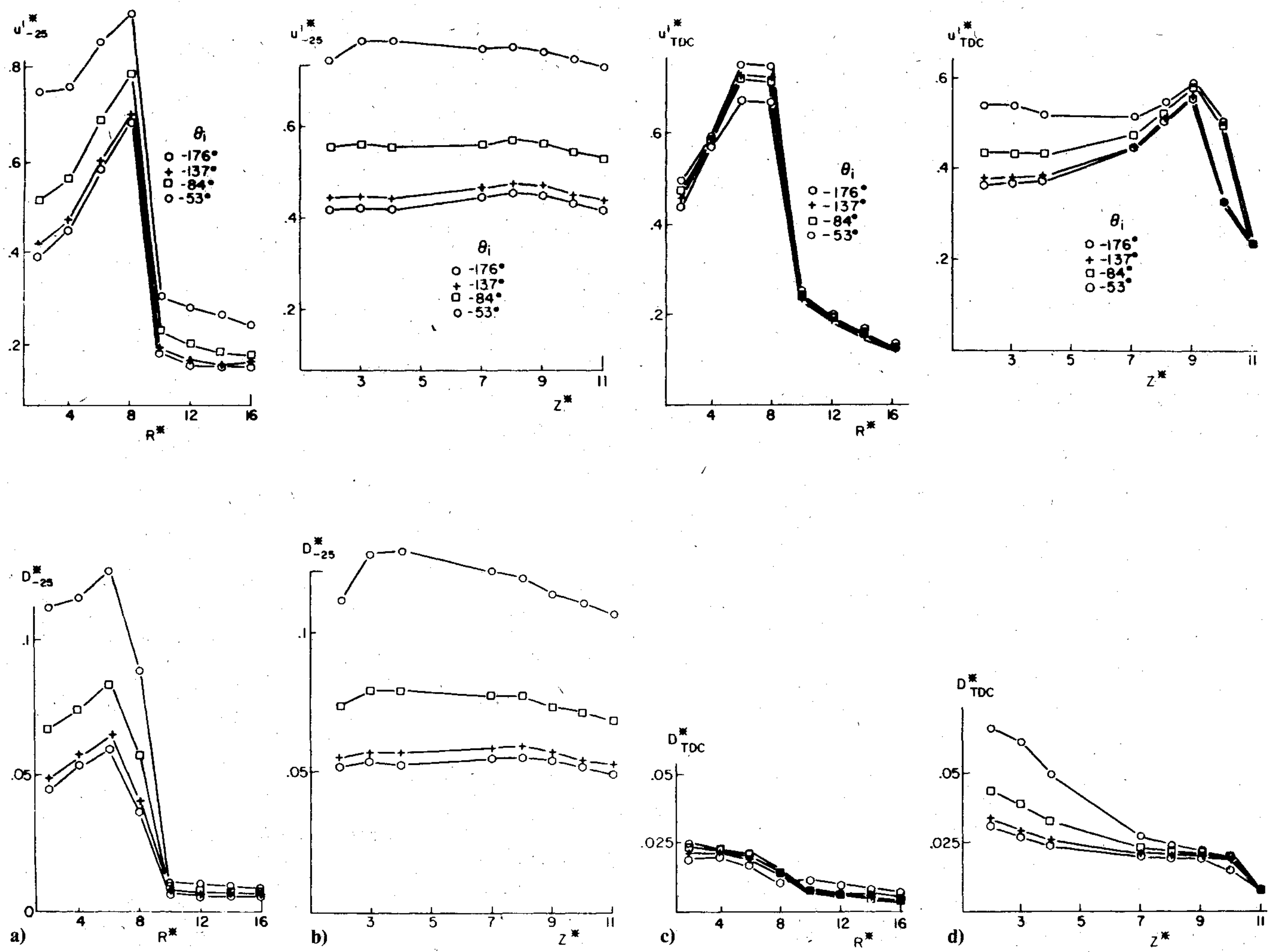

Fig. 3 Cup-in-piston chamber without swirl: effect of initial magnitudes of k and ϵ on u' and D at −25 deg and TDC: a) and c) 1.2 mm below the head; b) and d) at half the radius of the cup (SQ = 84%, CR = 21, SR = 0, η_v = 100%, rpm = 1700).

Pancake-Type Chamber without Swirl

As far as the effect of the initial magnitudes of k and ϵ is concerned, comparing Figs. 1a with Figs. 1b a significant decrease of turbulence intensity and diffusivity is noticed in the last 25 deg before TDC but the values predicted at both -25 deg and TDC are rather insensitive to the crankangle at which the computation is started. We conclude the relaxation crankangle is about 100 deg and is independent of speed, in agreement with experimental observations of Lancaster[9] and Witze[10] and the results of Tabaczinsky,[11] and also of load and compression ratio. In Figs. 1c it is seen that by TDC the magnitude of various initial nonuniformities has decreased significantly. TDC uniformity is greater in test 3-4 than in test 2-4 on account of the greater initial diffusivity of the former in the inner region and in spite of its larger initial nonuniformity. That is, the tendency to uniformization is strong but the details of remnant nonuniformities depend on the details of the initial field.

Pancake-Type Chamber with Swirl

With a strong initial solid-body rotation of the charge, during compression turbulence is produced at the walls and diffuses to the core while the bulk flow slows down. Figures 2a and 2c show that near the axis, where the mean tangential motion tends to remain solid-body rotation throughout compression, sensitivity to initial magnitude and nonuniformities is similar to that of the case without swirl. Conversely, near the walls, TDC turbulence has increased and is least sensitive to its initial values but is sensitive to the strength and distribution of the initial bulk flow. The bulk flow itself decays most where it generates turbulence (see Figs. 2b and 2d in which w is the mean tangential velocity).

Cup-in-Piston Chamber without Swirl

The results of Fig. 3 are for a flat-head engine with a cup with straight walls in the piston and high squish as in Fig. 1b of Ref. 8 ($SQ = 1 - R_c^2/R^2 = 84\%$). The bottom of the cup is at $z^* = 1$. On most of the piston shoulder, the higher local surface-to-volume ratio contributes first to a faster decay of the intake turbulence and then to generation of it (Figs. 3a vs Figs. 1a). At TDC the intensity is again about equal to the initial one but the diffusivity is lower (Figs. 3c vs Figs. 1b). Around the lip, the shear induced by the squish increases the intensity of turbulence, particularly around TDC. However the TDC diffusivity is not markedly greater than without squish, due to the simultaneous reduction of the length scale (Figs. 3c vs Figs. 1b). Expectedly in regions that are most influenced by the squish, e.g., the shoulder and the lip, TDC turbulence is least sensitive to its initial magnitude (Figs. 3a and 3c) and uniformity,[7] whereas along the axis and bottom of the cup (for this particular cup geometry) the decay and development of uniformity of the initial turbulence are more similar to those of the case without squish (Figs. 3 vs Figs. 1).

Cup-in-Piston Chamber with Swirl[7]

With both strong squish and strong swirl, turbulence intensity and diffusivity are everywhere higher than in the pancake-type chamber (at the same volumetric efficiency) at both -25 deg and TDC due to the turbulence generated within the chamber. Correspondingly, in the shoulder region the swirl velocity decays more rapidly with squish than without it, but in the cup the opposite is true. Since the swirl generates a significant fraction of the chamber turbulence near TDC, the effect of the initial magnitude of the bulk flow is substantial, particularly in regions dominated by the swirl, e.g., in the piston shoulder area. In regions where squish dominates, e.g., in the lip area, the initial magnitude of both turbulence and swirl are less important. And where both swirl and squish are least influential, e.g., near the axis and bottom of the cup, the initial turbulence is remembered most in accordance with experimental trends.[12] Initial turbulence nonuniformities are also overshadowed by the nonuniform turbulence generated within the chamber and at TDC remain detectable only in the most protected of the cup regions, e.g., near its axis and bottom.

Conclusions

In the absence of squish and of strong initial bulk flows, intake turbulence decreases during compression and becomes spatially uniform. The relaxation crankangle is about 100 deg and is independent of engine speed, load, and compression ratio. Thus, in the absence of strong initial bulk flows TDC turbulence is insensitive to the details of the intake process. Squish does not influence the entire chamber uniformly. For a simple coaxial straight cup, the shoulder of the piston and the lip of the cup are the regions most affected by it. There, near TDC, turbulence is dominated by the squish itself. Toward the center and bottom of the cup TDC turbulence behave similarly to the case of no squish. Swirl, with or without squish, generates its own turbulence that in turn modifies the swirl. The initial strength and distribution of the swirl is remembered at TDC both in the mean flow and in the turbulence, but initial nonuniformities in turbulence still tend to decrease rapidly during compression.

These trends and conclusions pertain to the limit cases of no swirl and no squish and of very strong swirl and squish. The nonlinear nature of the equations suggests that specific computations should be made when specific chamber geometries and operating conditions are of interest.

Acknowledgments

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