ON THE PREDICTION OF SWIRLING INDUCED RECIRCULATION

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ABSTRACT

The present paper is devoted to the assessment of several turbulence models for the prediction of swirling induced recirculation. In addition to the Launder-Sharma model which is used as reference model, several turbulence models have been assessed. The first model investigated is the Launder-Sharma model with a rotation correction. The performance of the cubic Craft-Launder-Suga model as well as the Reynolds stress modelings are also evaluated. For Reynolds stress model, both the IP and SSG model have been assessed. Computation has been performed on three confined swirling flow configurations involving a central toroidal recirculation zone. Numerical results reveal that the prediction of swirling induced recirculation remains a challenging task for turbulence modelization. None of the tested turbulence models is capable to give satisfactory prediction for the swirling flows investigated in the present study. Among all turbulence models tested, the best compromise is obtained by the linear eddy-viscosity model with rotation correction rather than by more complex turbulence models such as non-linear cubic or Reynolds stress models.

INTRODUCTION

Swirling jets are often employed in combustor to enhance air-fuel mixing and flame stabilization. Usually, a central toroidal recirculation zone is created due to the discharge of the swirling jets into the combustor. Since such a recirculation zone plays an important role in the mixing and stabilizing process, its accurate prediction is of crucial importance in a CFD computation. A swirling fluid has mainly two effects on the flow field. The first is a kinematic effect. It exists even in a laminar flow and can be easily understood by examining the Navier-Stokes equations written in the cylindrical coordinate system. The w^2/r term appearing in the v momentum equation is a dominant term which is mainly balanced by the pressure gradient. Consequently, the decay of the swirling velocity component in the axial direction due to the discharge of the swirling jet into the combustor will create an adverse pressure gradient, resulting in the formation of the central recirculation zone when the swirling intensity is strong enough. This effect can be called pressure-induced swirling effect. In a turbulent flow, a swirling fluid influences substantially the turbulent structure of the flow. This is the second effect that can be called turbulent transport induced swirling effect. While the turbulence modelization is not so critical for the pressure effect, the modelization of the turbulence transport effect for a swirling flow is a challenging task for all turbulence models. It is well known that the conventional linear eddy-viscosity models are unable to predict correctly a swirling flow. Considerable efforts have been devoted during the past three decades to develop more advanced models. Reynolds stress models, cubic nonlinear models and rotation sensitive linear eddy-viscosity closures are among the most representative. Frequently, those advanced models are presented separately only for the test case for which they are found to work well. Few systematic comparisons between them have been performed. The main purpose of this paper is to contribute to the assessment of different types of turbulence models for the prediction of swirling induced recirculation. In addition to the Launder-Sharma model chosen as reference linear eddy-viscosity model, the models selected for the present study are Reynolds stress models with the IP and SSG pressure-strain rate correlation models, Craft-Launder-Suga's cubic nonlinear model, and a swirl sensitive linear eddy-viscosity model originally proposed by A. Hellsten that has been re-tuned for swirling flow in the present study. Three test cases have been chosen for the assessment, the coaxial jet configuration of Roback and Johnson (1983), the central jet configuration of Ahmed and Nejad (1992), and the annular jet configuration of Holzäpfel et al. (1999). All of them are confined swirling flow configurations with moderate swirling number (0.4-0.5) involving a central toroidal recirculation zone.

TURBULENCE MODEL

The earliest, simplest but still among the most efficient approaches to take into account the swirling effect is to introduce empirical correction into the length scale equation in an eddy-viscosity model. Such an approach often relies on a parameter representing the swirling intensity called the Richardson number. In early implementation, the Richardson number was defined specifically for a certain type of flow and coordinate system. A more general definition based on the invariants of the vorticity and the strain rate tensor is proposed by Hellsten (1998). Application of Hellsten's model by the authors reveals that it is unsuitable to swirling flow computation. This is why the rotation correction proposed by Hellsten is re-tuned in the present study for swirling flow. It is based on the Richardson number defined by :

$$R_i = max\left[max\left(2, \frac{|\Omega_{ij}|}{|S_{ij}|}\right)\left(max(2, \frac{|\Omega_{ij}|}{|S_{ij}|}) - 1\right), -\frac{0.8}{C_{rc}}\right]$$

Here, the limiter is introduced in the definition of the Richardson number mainly to enhance numerical robustness. Hellsten applied the rotation correction to the Menter's SST model. Our computation reveals that for the prediction of the swirling induced recirculation, k-ϵ model gives better result that k-ω model. In the present study, the rotation correction is applied to the Launder-Sharma model by replacing the $C_{\epsilon 2}$ coefficient in the ϵ transport equation by $1 + (C_{\epsilon 2} - 1)F_{rc}$. The rotation correction function F_{rc} is computed by

$$F_{rc} = \frac{1}{1 + C_{rc}R_i}$$

Where $C_{rc} = 2.75$, $|S_{ij}| = \sqrt{2S_{ij}S_{ij}}$, and $|\Omega_{ij}| = \sqrt{2\Omega_{ij}\Omega_{ij}}$, S_{ij} and Ω_{ij} being the strain-rate and the vorticity tensors.

With RANS computation, the Reynolds stress model is the only existing model that has the potential to compute correctly a swirling flow. The success of the Reynolds stress model depends largely on the modelization of the pressure-strain rate correlation and the dissipation tensor. It has been shown by Younis et al. (1996) and also by Chen and Lin (1999) that the quadratic SSG pressure-strain model (Spezial et al., 1991) performs better than the linear IP or LRR pressure-strain model for the prediction of swirling flow. Computation with the SSG and IP models is performed in the present study to see if the improvement of the SSG model is also confirmed for confined swirling flows involving a central toroidal recirculation zone. The computation is performed with a low Reynolds number model proposed by the authors (Deng and Visonneau, 1999) to avoid the use of the wall function approach which may introduce uncertainty.

Theoretically, swirling effect can also be taken into account by a cubic nonlinear closure. In the present study, we have tested the Craft-Launder-Suga cubic nonlinear model (Craft et al., 1996) in order to assess the predictive capability of this class of models.

RESULTS AND DISCUSSIONS

The computations are performed on three different configurations as mentioned above. Although they all involved confined swirling flow with central recirculation, the flow characteristics are quite different. Numerical result is obtained with a finite volume code by using second order discretization scheme. All computations have been done in 3D domain in Cartesian coordinate system using a single control volume in the circumferential direction where rotationally symmetric condition is applied. All computations have been done with a 121x91x2 grid. Attentions have been made for all test cases to ensure that grid effect is negligible compared with model effect. Computational results for the three configurations are detailed in the following subsections.

Roback and Johnson coaxial jet test case

The first configuration investigated is the Roback and Johnson coaxial jet test case. This configuration has been chosen as a test case for the 9th ERCOFTAC/IAHR Workshop on Refined Turbulence Modeling (http://www.sla.maschinenbau.tu-darmstadt.de/workshop01.html). It consists of central non-swirling jet with an outer diameter of 29mm which is also the inner diameter of the outer swirling jet. The outer diameter of the swirling jet is 59mm. The computational domain begins at the exit of both jets (x=0mm) and covers the entire combustor up to the exit located at x=1016mm. The inlet condition is imposed by extrapolating the measurement data taken at x=5.1mm to the inlet plane, satisfying the global mass balance. Dissipation rate of the kinetic energy of turbulence at the inlet is obtained by assuming a constant dissipation length scale that takes the value of 0.008m and 0.0045m respectively for the inner jet and the outer swirling jet.

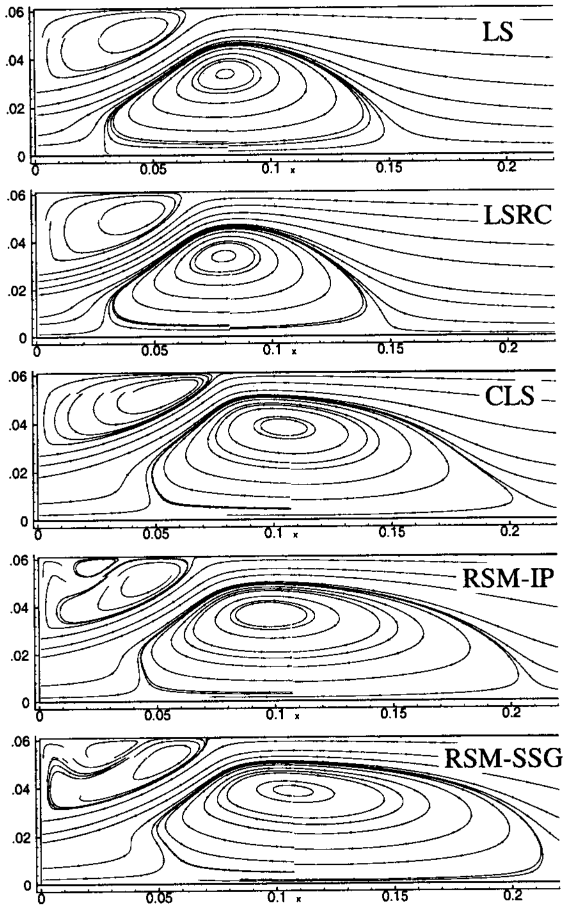

Figure 1: Comparison of the streamlines.

For this test case, the outer swirling jet is pushed away from the axis of symmetry by the central non-swirling jet. Consequently, the pressure effect is much more important than the turbulent transport effect near the entrance region. The formation of the central toroidal recirculation is mainly determined by the pressure field. All tested turbulence models are able to predict the recirculation (figure 1). But none of the tested models is capable to give a satisfactory solution. The recirculation length predicted by linear eddy-viscosity model with or without rotation correction is too short, while the cubic model and the Reynolds stress models give too long a recirculation length. Both Reynolds stress models predict a counter-rotating recirculation region in the upper-left corner which is not present in the measure-

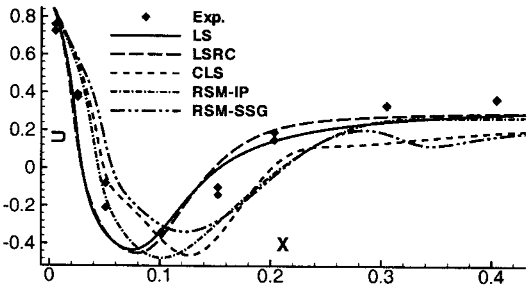

Figure 2: Centerline U velocity profiles.

ment. No significant improvement is obtained with the SSG model compared with the IP model except the S shape central toroidal recirculation zone which is also observed in the measurement (Roback and Johnson 1983). Figure 2 compares the centerline U velocity profiles. It confirms that the recirculation length is poorly predicted by the computation what ever the turbulence model used. We can also observe that the Launder-Sharma model with and without rotation correction provides a similar prediction, which suggests that the swirling effect is small because of the presence of the central non-swirling jet.

The above observations are confirmed by the comparison of the U velocity profile obtained near the reattachment point at x=0.1524m shown at figure 3. Comparison of the W velocity component at the same position shown at figure 4 indicates that the prediction obtained with the SSG model is quite different from that obtained with the IP model. The former model predicts a much higher value of W near the axis of symmetry than the later model, which is in better agreement with the measurement.

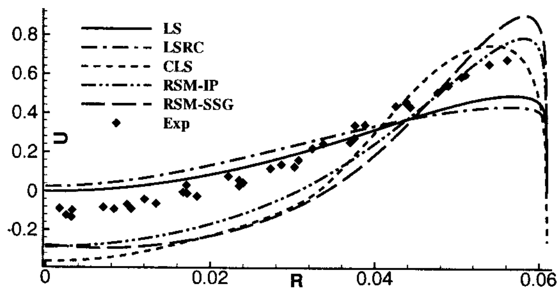

Figure 3: U velocity profiles at x=0.1524m.

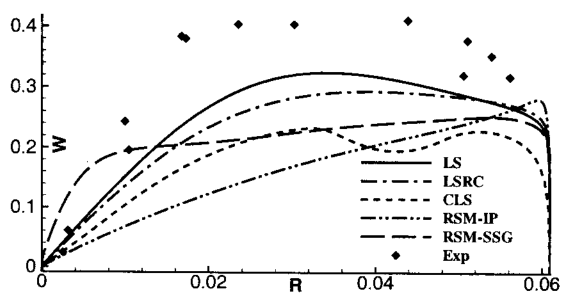

Figure 4: W velocity profiles at x=0.1524m.

Another important phenomenon to study in a confined swirling configuration is the decay of rotating fluid. It is well known that a rotating fluid exhibits a non solid body rotation when it decays inside a pipe. This phenomenon

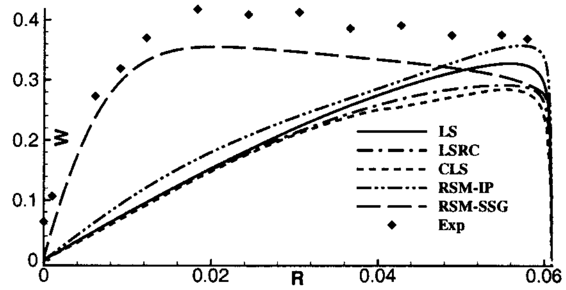

Figure 5: W velocity profiles at x=0.4064m.

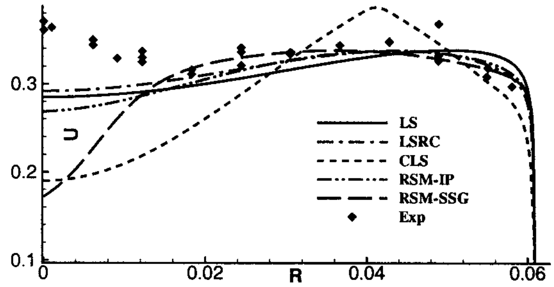

Figure 6: U velocity profiles at x=0.4064m.

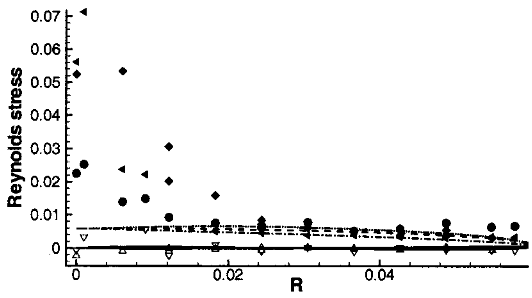

Figure 7: Reynolds stress profiles obtained with the RSM-SSG model at x=0.4064m. Symbols: measurement, Lines: computation.

can not be predicted by any linear eddy-viscosity model. Reynolds stress model is believed to be capable to predict the non solid body rotation decay of a rotating fluid. The present computation indicates however that the success of the Reynolds stress model in predicting this phenomenon depends on the pressure-strain model used. Figure 5 shows the prediction obtained with different turbulence models for the swirling velocity component at the position x=0.4064m. One can observe that only the RSM-SSG model is capable to predict correctly the non solid body rotation. The prediction obtained by the RSM-IP model is almost as poor as that obtained by the linear eddy-viscosity model. It is not surprising to observe that the rotation correction on the Launder-Sharma model brings little improvement, since it remains a linear eddy-viscosity model. The performance of the cubic model is very disappointed. The cubic term is designed and calibrated with the rotating pipe flow to predict the non solid body rotation behavior of the fluid. Although it is capable to predict the non solid body rotation of a fluid inside a rotating pipe, it fails completely to predict the non solid body rotation decay of a rotating fluid inside a stationary pipe. As the linear eddy-viscosity model, the CLS cubic model predicts nearly a solid body rotation for the rotating fluid. In addition, the prediction of the streamwise component shown in figure 6 is completely wrong.

Even if the SSG model is capable to predict correctly the swirling velocity component, its performance cannot be considered as satisfactory. Inspection of the U velocity component shown in figure 6 reveals that the behavior of the U velocity component observed in the measurement is quite different from that predicted by the SSG model. The measurement indicates an increase of the U velocity component approaching the centerline, while the SSG model predicts an opposite tendency. The turbulence characteristics of the rotating fluid in this region are quite special. In the region near the centerline, it exhibits a very high level of normal stresses (figure 7, closed symbols), while the shear stresses are fairly small (figure 7, opened symbols). Such a special feature is totally missing in turbulence modeling. The SSG model for example predicts a low level of normal stress everywhere (figure 7) although the swirling mean velocity component is correctly predicted. Since the streamwise variation of the mean flow is very small, such a high level of turbulent kinetic energy cannot be generated locally. It can only be generated in the upstream region and convected without losing too much its intensity downstream. Here, two distinct turbulence characteristics are observed, i.e. the coexistence of a high level of normal stress with a low level of shear stress, and the permanence of a high level of turbulent kinetic energy without substantial local generation. Unfortunately, none of them can be predicted correctly with any existing statistical turbulence model. ¿From the statistical point of view, the anisotropy of the high level normal stress is certainly responsible for the non solid body rotation of the rotating fluid. Without being able to predict such a high level of normal stress, the SSG Reynolds stress model is unable to give a good prediction simultaneously for the streamwise velocity component and the azimuthal velocity component.

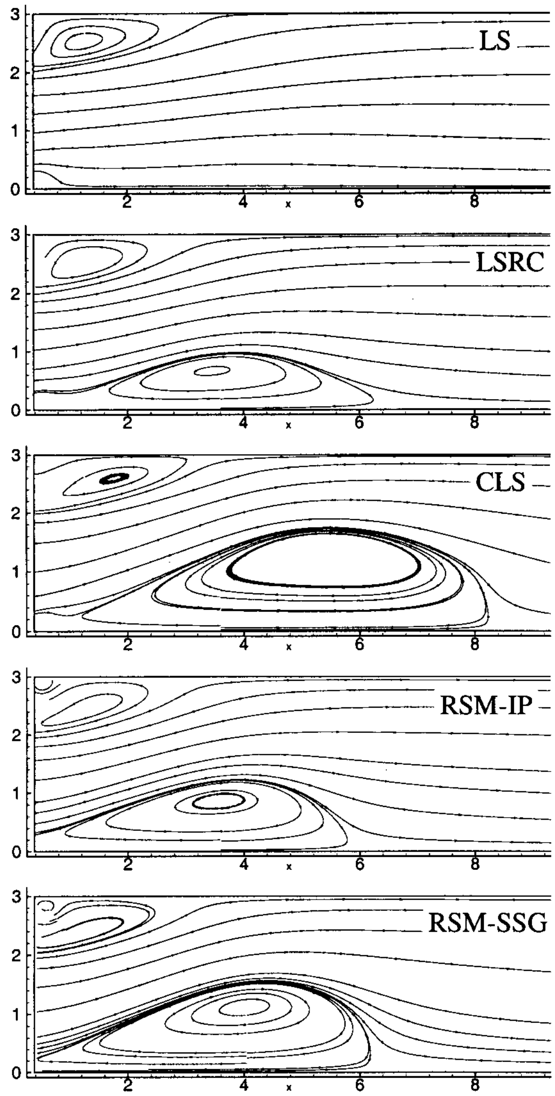

Figure 8: Comparison of streamlines.

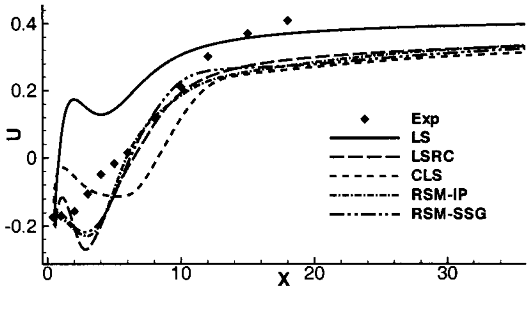

Figure 9: Centerline U velocity profiles.

Ahmed and Nejed central jet test case

Ahmed and Nejed (1992) have studied a central jet configuration. It consists of an inlet pipe with a diameter of 101.6mm and a combustor chamber with a diameter of 152.4mm and a total length of 1850mm. Ahmed and Nejed have performed several studies with different inflow conditions. The test case chosen in the present study is the one with a swirling number of 0.5 for which the experimental data is available in the ERCOFTAC database (http://ercoftac.mech.surrey.ac.uk/). Based on the inlet pipe centerline velocity and the step height (h=25.4mm), the Reynolds number is 31250. Measurement data available at x=0.38h is used as inlet condition. The turbulent dissipation is estimated by k**1.5/L with a length scale L prescribed as 1.08h.

The Ahmed and Nejed central jet configuration is quite different from the previous test case. Since the swirling jet is present in the center of the pipe and the swirling effect is inversely proportional to the radius, its effect becomes very important. The successful prediction of the flow recirculation depends on an accurate description of the pressure effect as well as the turbulent transport effect. The measurement done by Ahmed and Nejed (1992) shows existence of a central recirculation zone extended to x=5.5h. The Launder-Sharma model is unable to predict the flow recirculation in the central region. This failure was attributed to the instinctive shortcoming of the linear eddy-viscosity model by Lai (1995). With the rotation correction however, the central toroidal recirculation zone is well captured with the corrected Launder-Sharma model. On the other hand, the Craft-Launder-Suga cubic model predicts a too long recirculation length. Lai (1995) have performed a computation for the same flow with Reynolds stress model using IP pressure-strain model, giving a recirculation length of 5.96h. The present result using RSM-IP model is in good agreement with Lai's result. The RSM-SSG model provides similar solution as the RSM-IP model with a slightly larger recirculation region. The comparison of the centerline U velocity profiles shown in figure 9 confirms the poor performance of the Launder-Sharma model and the Craft-Launder-Suga model. The non linear SSG pressure-strain model does not provide any improvement for this configuration compared with the IP pressure-strain model. It is unexpected to observed that the Launder-Sharma model with rotation correction gives almost the same prediction as more complex Reynolds stress model. However, none of the tested turbulence models is capable to give a good prediction inside the recirculation zone.

Figures 10 and 11 compares the U and W velocity profiles in different sections in more detail. Only three turbulence models are now compared. The results obtained by the cubic model is not shown because of its poor performance. The RSM-SSG model is not included in the comparison because it gives no better prediction than the RSM-IP model. The rotation correction is found to improve the performance of the Launder-Sharma model on the prediction of the streamwise velocity component, especially near the entrance region. But it deteriorates the prediction of the azimuthal velocity component. The RSM-IP model gives no better prediction than the Launder-Sharma model with rotation correction, both on the U velocity component and on the W velocity component. It is interesting to note in figure 11 that better results are obtained on the azimuthal velocity component by the Launder-Sharma model when the rotation correction is not applied. Such behavior is also observed in other test cases such as in figure 4. The fact that the rotation correction improves the prediction of the streamwise velocity component, while at the same time deteriorates the prediction of the swirling velocity component illustrates the limitation of linear eddy-viscosity model in the prediction of swirling flow.

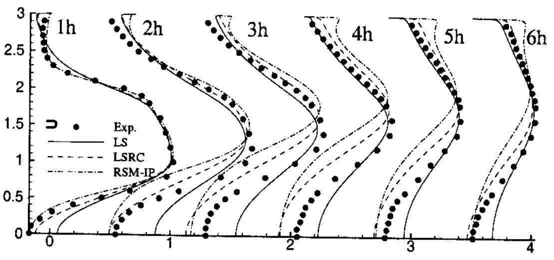

Figure 10: U velocity profiles.

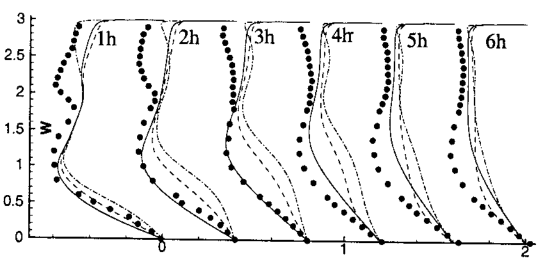

Figure 11: W velocity profiles (same legend as figure 10).

Holzäpfel et al. annular jet test case

Holzäpfel et al. (1999) have conducted an experimental study on a annular jet configuration with two different swirling numbers. The present computations are performed only for the smaller swirling number test case (0.4). The initialization approach is similar to the previous test case. A turbulent dissipation length scale of 0.008m is used to determine the turbulent dissipation at the inlet.

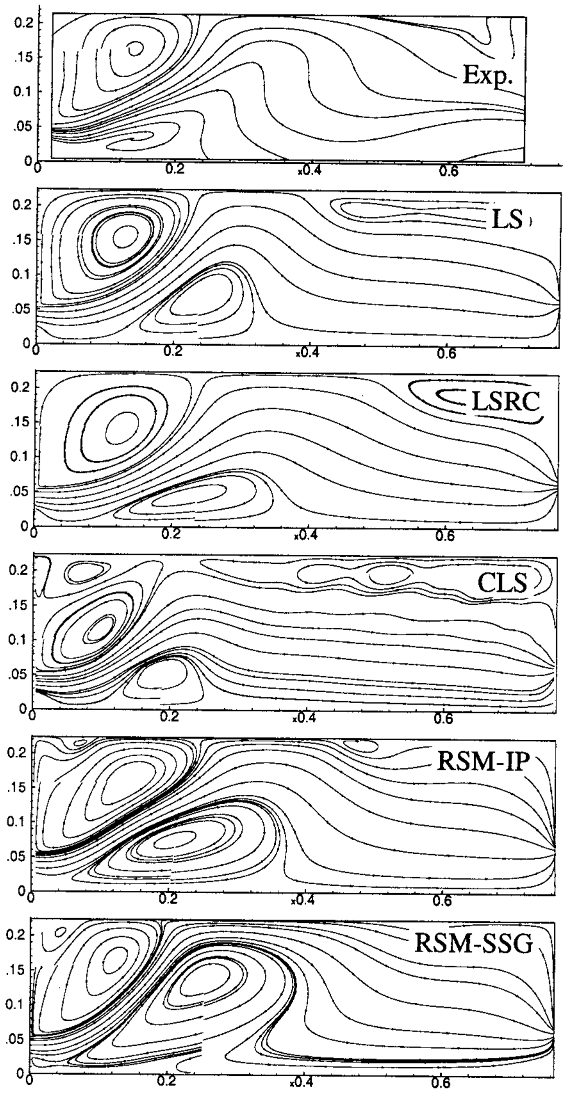

Figure 12: Comparison of streamlines.

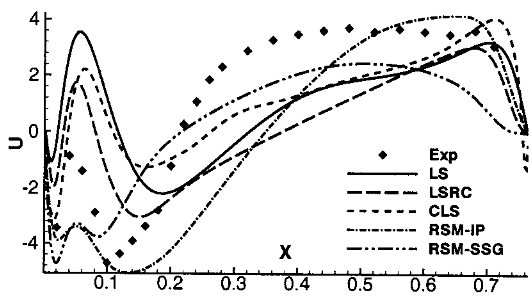

Figure 13: Centerline U velocity profiles.

This test case is even more challenging than the two previous ones. The swirling jet is located quite near the axis, introducing therefore a strong swirling effect. Compared with the two previous test cases where the flow is strongly confined by the outer wall, the dimension of the combustor is quite big compared with the width of the swirling jet (about 8:1), leaving thus enough space for the flow to develop freely inside the combustor. Streamlines predicted with different models are compared in figure 12. The measurements indicate an ear-like central recirculation extended to about x=0.2 meter. With the Launder-Sharma and the Craft-Launder-Suga cubic model, the central recirculation zone is splitted into two parts. The LSRC model improves the prediction on the shape of the recirculation zone, although the recirculation length is longer than measured. With Reynolds stress models, the recirculation zone is too large compared with the measurements. Results obtained with the SSG model are equally bad as that obtained with the IP model, although the shape of the recirculation zone looks more like an ear as observed in the measurement. Figure 13 which compares the central line axial velocity profiles, confirms the improvement of the rotation correction on the result obtained with the Launder-Sharma model. However, the discrepancy compared with the measurement data is still quite important. Due to the ear shape central recirculation zone, the RSM-SSG model gives a centerline U velocity profile closer to the measurement data when compared with other turbulence models.

More detailed comparison of U and W velocity profiles at different sections are shown in figures 14 and 15. For the reason as mentioned above for the previous test cases, results obtained by CLS model and RSM-SSG model are not included for comparison. Similar conclusions can be drawn as for the previous test case. One can observe an improvement on the U velocity component, but a deterioration on the W velocity component with the rotation correction. The RSM-IP model gives a very poor result both on the U velocity component and on the W velocity component.

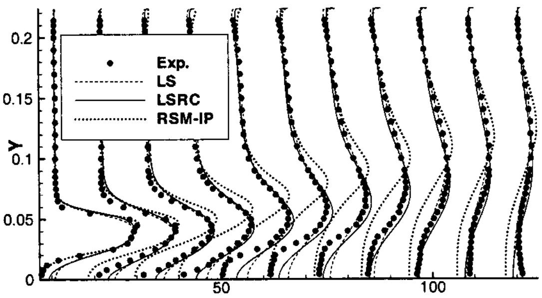

Figure 14: Comparison of U velocity profiles.

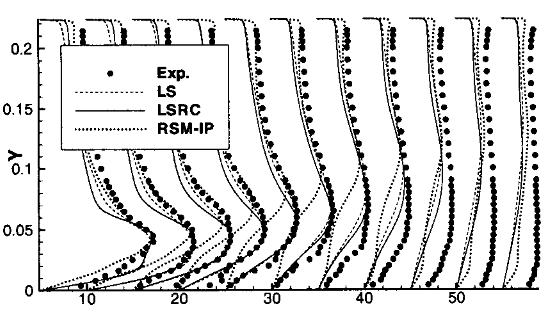

Figure 15: Comparison of W velocity profiles.

CONCLUSIONS

Turbulence modelization of swirling flow remains a challenging task for all turbulence models. Computations of several swirling induced recirculation flows indicate that the cubic model tends to deteriorate the result of linear model rather than to lead to improvement. The Reynolds stress model tends to exaggerate the swirling effect. No significant improvement is obtained with the quadratical SSG model compared with the linear IP model. Except for a few specific case, Reynolds stress model can not be considered as a good alternative for swirling flow computation. With the rotation correction proposed in the present paper, systematic improvement can be obtained with the Launder-Sharma linear model. However, it inherits all drawbacks of the linear eddy-viscosity model. In spite of that, linear eddy-viscosity model with rotation correction such as the LSRC model presented in this paper remains the best compromise for the computation of swirling induced recirculation flow.

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