model on two different mesh levels, with the experimental data are presented. It is seen that the code predicts well the very complex three-dimensional flow in a linear compressor cascade with tip clearance. The tip separation vortex inside the gap deduced from ink-trace visualization on the tip surface in the previous experimental study is reproduced. Increasing grid points greatly improves the predicted results. However, the tip leakage flow is still underestimated, which may be attributed to insufficient mesh resolution and/or to the simple turbulence model. Significant influence of relative motion on the tip leakage vortex is confirmed through a simulation under the same flow condition.

The present validations show that:

- the overall three-dimensional flow structure can be globally reproduced with a coarse mesh of close to 50,000 points, including the tip clearance gap.
- in order to capture the tip separation vortex, a minimum of 13 points over the clearance height appears necessary; earlier computation with 7 points did not allow us to capture this vortex.
- more detailed comparisons with the experimental data, especially the viscous and turbulent dominated effects, require finer meshes.
- the present fine mesh calculations, with close to 200,000 points, may still not be fine enough to ensure mesh independence and to reproduce quantitatively all the details of the flow.
- in addition to the above, the influence of turbulence modeling and/or transition effects should also be assessed. However, this can only be done, without ambiguity, on fine enough meshes.

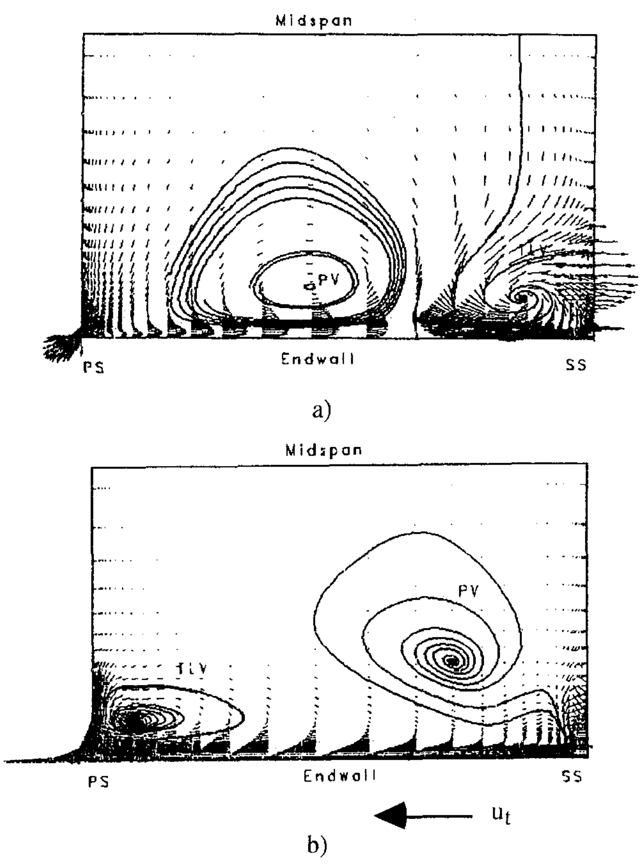

Fig. 23 Secondary flows near trailing edge at design condition: (*a*) stationary endwall, and (*b*) moving wall at a flow coefficient of 0.5

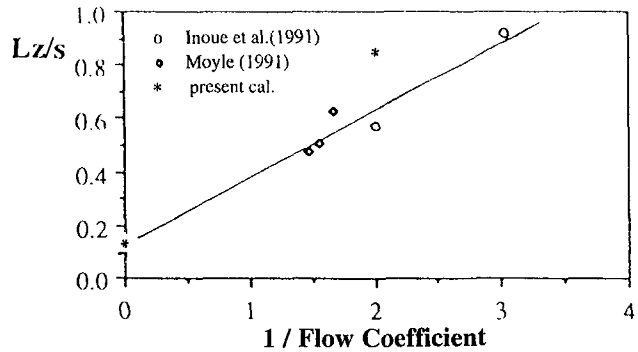

Fig. 24 Variation of tip vortex loci with flow coefficient, from Kang and Hirsch (1993b)

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DISCUSSION

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The authors continue to provide detailed information about the flow in the clearance region of compressors in cascade. We

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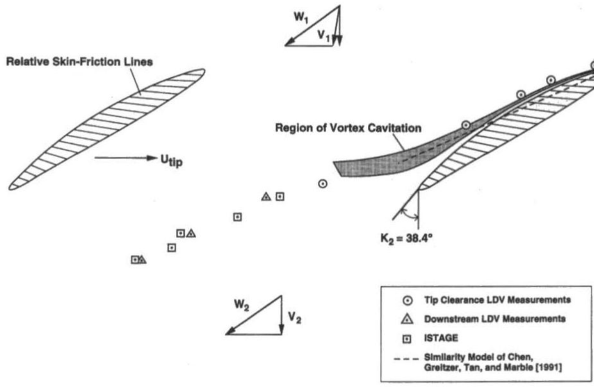

Fig. 25 Trajectory of rotor tip clearance vortex core

have been concerned for some time about a difference that appears in the authors' estimates for a parameter A in a simple model for predicting the locus of the clearance vortex by Chen et al. (1991). Chen et al. suggested a value of 0.46, whereas Kang and Hirsch (1993b) suggested 0.19; the two values imply a difference of more than a factor of two in the circumferential displacement of the vortex from the suction side of the blade. In our discussion we emphasized the importance of endwall motion and a non-collateral inlet boundary layer, as would occur in a rotor.

In the authors' response (Kang and Hirsch, 1993b) they introduced speed of the blade tip (relative to the wall) as an independent variable. We have to admit to being unable to follow the process leading from Eq. (4) to Eq. (6) in the paper and by the subsequent treatment of Inoue's data. As regards the latter, reducing the flow coefficient from 0.5 to 0.33 does more than just alter the speed of the wall (assuming the axial velocity is held constant)—it alters the pressure distribution around the whole blade.

In the final part of section 5 of the present paper the authors return to our discussion of the earlier paper (Kang and Hirsch, 1993b), and the effect of a moving endwall. This time they are presenting numerical results with a moving endwall. As far as we can understand the calculations with a moving endwall refer to the condition of flow coefficient equal to 0.5. With an axial inlet flow to the rotor a flow coefficient of 0.5 means the axial velocity outside the boundary layer is equal to half the blade speed, so the inlet flow direction is $\cot^{-1} 0.5 = 63.4$ deg. The blade inlet direction is, however, 32.5 deg, so the specified flow coefficient would correspond to an incidence of almost 30 deg. It may be that we have misunderstood what was done, but it is clearly important that the conditions chosen be physically plausible if the effect of relative motion of the endwall is to be assessed.

At the conference at which this paper was presented there was a two-part paper by Zierke et al. [later consolidated and published as Zierke et al. (1985)], who examined clearance flows in a high-Reynolds-number axial pump. With Dr Zierke's permission, Fig. 4 is reproduced here as Fig. 25. This shows how well the model of Chen et al. fits the observations of the vortex position; it also shows how well it fits the predictions of Navier–Stokes solver, ISTAGE.

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Authors' Closure

The authors would like to thank N. Cumpsty and E. Greitzer for their discussion and for raising the question of the range of validity of the simplified model, developed by Chen et al. (1991), for the prediction of the tip vortex trajectory, which is represented as follows:

$$\frac{L_z/s}{x/c} = A \cdot \sqrt{\frac{c(\tan \beta_1 - \tan \beta_2)}{s \cos \beta_m}} \tag{1}$$

This discussion provides an opportunity for a more critical analysis of this model, based on inviscid considerations and on a number of assumptions, some of which deserve particular attention:

(*i*) It is fair to assume, following Chen et al., that the basic mechanism for the creation of the tip vortex is of inviscid nature, since it is mainly pressure driven;

(*ii*) It is also acceptable to consider that the influence of the tip clearance height is not a first-order effect, although a certain dependence of the tip vortex trajectory on clearance height, especially for small values, is clearly observed by Inoue et al. (1986), Fig. 9 in Chen et al. (1991);

(*iii*) The influence of the entrainment effect of the end wall is totally ignored, which is clearly the main weakness of the model of Chen et al.

Due to the small values of tip clearance gap heights, the viscous effects from the end wall tangential motion, leading in addition to the end wall boundary layer skewness upstream of the blade, reinforces the tip vortex and drags it away from the suction surface. A rigorous analysis of the data of Fig. 15 of Chen et al. clearly indicates the nonnegligible effect of the flow coefficient. With higher flow coefficients, that is for smaller end wall entrainment at fixed axial velocity, the experimental position of the tip vortex is shifted toward the suction surface and the correspondence with the simplified model seen on this figure is, at best, qualitative. This appears, both from numerical simulations and from experimental data of Inoue et al. (1991) and Moyle et al. (1992) to be an important effect. In high-speed axial compressors, Adamczyk et al. (1993) and Jennions and Turner (1993) observed from their numerical results on NASA Rotor67 that with reducing mass flow the tip vortex inclined more toward the tangential direction.

On the other hand, the calibration of the constant in the model of Chen et al. to the value of 0.46 is based on data corresponding mostly to flow coefficients around 0.5, as seen from Table 1 in their paper. Only one set of data, from Smith, refers to a lower flow coefficient of 0.29. As seen from Fig. 5 in the paper of Chen et al., reproduced here as Fig. 26, the corresponding points (open diamonds) are off the main line of slope 0.46 and fit quite well to a slope of 0.55, indicated with a dashed line.

The assumptions made in our response to a previous discussion by the same authors on our paper (Kang and Hirsch, 1993b), are based on the following considerations:

(*i*) The quasi-linear variation of tip vortex position with the square root of main loading is well confirmed by experimental and numerical data;

(*ii*) The nearly linear variation of tip vortex position L_z with axial distance is also well validated, at least over the range about 70 to 80 percent chord from the leading edge. The position L_z is measured normal to the camber line as defined by Chen et al. (1991), even though a measurement from the chord is more convenient and may be even closer to a linear relation at certain conditions;

(*iii*) A linear variation of the entrainment effect due to the end wall relative motion, scaled by the axial velocity, is the simplest assumption. Based on our experimental and numerical data for a fixed end wall and the data of Inoue et al. (1991) and Moyle et al. (1992) for axial compressor rotors, measured at the trailing edge section, we tentatively calibrated the constant A in the model of Chen et al. as