3-D Loss Prediction Based on Secondary Flow and Blade Shear Layer Interaction

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ABSTRACT
The 3-D loss distribution along axial turbomachines has been investigated in the present work. For this purpose, our well documented secondary flow calculation method has been interactively coupled with our fast blade-to-blade shear layer calculation code, in order to predict the complete secondary flow and profile loss distribution.

The presence of an even moderate secondary flow field modifies the flow angle distributions, thus making clear the need of a reliable profile loss prediction method for off-design cases, especially in the presence of separation.

Accordingly, the blade-to-blade code predicts the profile loss distribution and subsequently provides the correct total pressure field for the secondary flow calculation. The combination of the above mentioned codes finally gives a realistic picture of the flow quantities at any S3 surface along a machine at a minimal computer cost.

Several test cases, including axial compressor and turbine cascades as well as transonic axial compressor blade rows have been investigated. The results are in good agreement with the experimental data and are favourably compared with various recent correlations.

NOMENCLATURE

Ttr relative total temperature
(u',v',w') components of velocity fluctuation with time
u velocity
W(Wu,Wu,We) relative velocity vector
X added work (per sec)
Y energy form factor =\ln(Re) + 2.1a
Z ratio of specific heats
S shear layer thickness
S1 displacement thickness of the shear layer
S2 momentum thickness of the shear layer
S2(\theta,\mu,\nu) coordinates of an orthogonal curvilinear axisymmetric coordinate system
H constant of Sutherland's equation (=5 + 1.7)
\mu meridional striction parameter
\nu kinematic viscosity coefficient
\rho density
\sigma density ratio
\omega angular velocity vector

Subscripts

e external flow
H,T value at hub and tip
k kinematic shear layer quantities
o,co reference quantity
p,s pressure, suction wall respectively
r either pressure or suction wall
t total value
(x,y,z) components in curvilinear axisymmetric orthogonal coordinate system (peripheral, meridional, normal)
\theta shear layer edge

Superscripts

(-) mean peripheral value
(\ast) initial external flow field
(\ast) time averaged value
one involves accurate but quite expensive 3-D numerical computational fluid dynamics (CFD) and by Leyl and Leoek (1987), Dass, 1986, Leonard and Wisler, 1990, Xu and Denton, 1990) which give an enormous volume and variety of information at a significant computational cost. The above research utilizes experimental data for the analysis of a flow field (Binder and Romey, 1983, Moore and Adhye, 1985, Kaldellis et al., 1990, Tremblay et al., 1990, Mee et al., 1990), but usually give the correct level but the wrong distribution of the blockage. All three of the above mentioned research directions recognize the importance of the 3-D losses and the blockage induced by the shear layers. However, the influence of the spanwise and chordwise distributions of these two factors upon the through-flow approximation has been only recently acknowledged. For example, Dring and Joslyn (1987) reported that the quality of the computed results strongly depends on the spanwise distribution of the aerodynamic blockage. They demonstrated that significant discrepancies are caused by using the correct level but the wrong distribution of the blockage. Besides, they mentioned (Dring and Joslyn, 1989) the impact of the endwall flow on airflow incidence and deviation.

The present work deals with the evolution of the spanwise distribution of the two main loss terms (i.e. profile and secondary flow losses) through subsonic turbine and compressor configurations. Additionally, an attempt is made to present the distribution of the total loss and the effectiveness of passage and selected blade rows along a machine, taking into account the existence of the blade shear layers and the secondary flow field, is given. Finally, the interaction between the secondary flow field and the blade designduced flow pattern is described. It is interesting to note that even for the moderately loaded cases investigated here, considerable alterations of the flow pattern are encountered.

For the above purposes our well documented secondary flow calculation method (Kaldellis et al., 1988a, Kaldellis et al., 1989a) has been recently extended in order to estimate the secondary losses and the energy exchange process for axial and radial compressors (Kaldellis et al., 1990a). In the present work the method is expanded for axial turbine secondary vorticity upon the flow field.

For internal cascade flows, performance is usually measured in terms of flow turning and total pressure loss through the cascade passage. The present method is able to predict the evolution of profile losses through subsonic cascade configurations over the entire incidence range with sufficient accuracy. For this purpose, the total pressure drop and therefore the characteristics of the blade shear layers at every point of the cascade passage are needed.

The momentum conservation equations written in an orthogonal curvilinear coordinate system associated with the external flow streams. These equations are rearranged along a normal (-ne) from the solid wall to the end of the shear layer. For an axial configuration we get:

\[
\begin{align*}
\frac{d(RP_0 u^2 \delta_2)}{dt} &+ \frac{\partial}{\partial s} \left( R \rho u^2 \delta_2 \right) + \frac{\partial}{\partial \eta} \left( R \rho u u_0 \right) = C_f \frac{\partial^2 i}{\partial \eta^2} + \frac{\partial (\rho u^2)}{\partial \eta} \int_{0}^{\eta} \left( \rho \frac{u^2}{2} - p \frac{u^2}{2} \right) \, d\eta
\end{align*}
\]

\[\frac{d}{dt} \left( R \rho u^2 \delta_2 \right) = C_f \frac{\partial^2 i}{\partial \eta^2} + \frac{\partial (\rho u^2)}{\partial \eta} \int_{0}^{\eta} \left( \rho \frac{u^2}{2} - p \frac{u^2}{2} \right) \, d\eta
\]

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\]
Integral total kinetic energy equation
\[
\frac{d}{dt}(R \rho u^2 \delta_1) + d \frac{u}{u} \cdot (\delta_1 - \delta_1) - 2 \frac{d (\omega^2 R^2)}{2 u^2} \cdot (\delta_1 - \delta_1) + \delta_1 \frac{d u}{D} + C_d d S_e
\]
(5)

The above mentioned equations are valid for cases with zero heat exchange. Gravity terms and expressions containing derivatives of the viscous terms along the mass flow direction are neglected. The complete form of these equations is given by Kaldellis (1988b), Kaldellis and Katramatos (1989b) and Renidis (1989).

(c) The total mass conservation equation written as:
\[
\int \rho u \cdot d n_e = m
\]
(6)

This equation is used to establish an approximate viscous-inviscid interaction procedure in order to relate the external flow and the two wall shear layers for confined configuration cases.

Viscous - Inviscid Interaction Procedure

Since our objective is to calculate fully separated internal flows a new viscous-inviscid interaction scheme is established. The scheme dictates strong coupling between the inviscid core and the shear layers, thus overcoming stability problems in the separation region. The key element in the new interaction procedure is the use of the total mass conservation equation (6) as a coupling equation solved simultaneously with the wall shear layer integral equations, and not as an additional equation to be verified via iterations.

During the development of our method several well documented inviscid numerical codes have been tested. It is interesting to mention that the results obtained by our method are not significantly dependent on the inviscid code used for the calculation of the external flow field for most of the cases examined. However, an enhanced form (grid independent) of the Alkalai and Leboeuf (1985) inviscid code, based on a potential method with rotational source terms, is adopted as the standard tool to be used with the present method. This inviscid method gives us the possibility to take into account the shear layer effects, introducing them as rotational source terms either at the wall, as a mass flow injection, or inside the flow field. It can also be extended in the frame of Navier-Stokes calculations by using the real velocity vector and the static pressure distribution as primitive variables.

Taking into account that the viscous-inviscid interaction scheme is approximative, we assume that the direction of the external velocity vector does not change. The corresponding modification of the velocity magnitude due to the shear layer displacement thicknesses can be expressed as:
\[
u_s = \mu \cdot \Psi \cdot \omega_s
\]
(7)

where:
\[
\Psi = \frac{Z - 1}{Z - 2 \cdot u^2 \cdot \omega_s}
\]
(8)

Applying the transformation of Truckenbrodt (1952) we define the velocity potential \( \Phi \) as:
\[
\Phi = \rho \cdot u_s \cdot d S_e
\]
(9)

and the non-dimensional velocity \( q \) as:
\[
q = \ln \left( \frac{u_s}{u_0} \right)
\]
(10)

Using equation (7) and differentiating equations (9) and (10) we get:
\[
\delta \Phi = \mu \cdot \Psi \cdot \delta \Phi
\]
(11)
\[
\delta q = \frac{d \cdot \omega_s}{u_0} \cdot \delta \Phi + \delta \Phi^2
\]
(12)

Introducing equations (11) and (12) in equations (3), (5) and (6) we obtain the final (SAS) system of equations:
\[
f_1(L_k, X_k, \mu) = -A_1 \cdot \rho \cdot u_s \cdot \delta \Phi + A_0 \cdot d L_k
\]
(13)
\[
f_2(L_k, X_k, \mu) = B_1 \cdot \left( \rho \cdot u_s \cdot \delta \Phi + B_0 \cdot u_0 \cdot \delta \Phi^2 \right)
\]
(14)
\[
f_3(L_k, X_k, \mu) = -B_1 \cdot \left( \rho \cdot u_s \cdot \delta \Phi + B_0 \cdot u_0 \cdot \delta \Phi^2 \right)
\]
(15)

The rest coefficients \( A_1, A_2, A_3, A_4, A_5, B_1, B_2, B_3 \) are defined in Appendix one.

Turbulence Modelling

The behaviour of the turbulent terms has been proven experimentally and computationally to be of great importance for the successful prediction of the flow field, especially in the separation region. For the evaluation of the integrals containing the turbulent terms in equations (13) and (14) we used the available experimental data assessed and presented for the first time by Kaldellis et al. (1983).

In order to realistically reconstruct the natural changes expressed by the experimental measurements and keeping in mind the scattering of the experimental data which discourages direct calibrations of the A and B integrals also reported by Kallas and Papailiou (1987), Kaldellis (1988b) and Gerolymos et al. (1989), a successfull effort to calibrate the turbulent components \( u' \) and \( v' \) was made. For this purpose calibrations for the maximum values of the turbulent components \( u'^{-2 \text{max}} \) and \( v'^{-2 \text{max}} \) were used (Kaldellis, 1988b), of the type:
\[
(u'^{-2 \text{max}}) = \frac{6}{\delta_s} \left( \frac{\delta_s}{\delta_{12}} \right)^2
\]
(20)
\[
(u'^{-2 \text{max}}) = \frac{6}{\delta_s} \left( \frac{\delta_s}{\delta_{12}} \right)^2
\]
(21)
\[
(v'^{-2 \text{max}}) = \frac{6}{\delta_s} \left( \frac{\delta_s}{\delta_{12}} \right)^2
\]
(23)

The combination of the above two expressions gives:
\[
(v'^{-2 \text{max}}) = \frac{6}{\delta_s} \left( \frac{\delta_s}{\delta_{12}} \right)^2 \left( \frac{\delta_{12}}{\delta_s} \right)^2
\]
(22)

Next, the turbulence velocity component profiles \( u'^2 \), \( v'^2 \) were calibrated (see Kaldellis (1988b), Kaldellis and Katramatos (1989b)) thus giving:
\[
(u'^2) = (u'^{-2 \text{max}}) \cdot h_1(\alpha, \phi, \phi_{12})
\]
(24)
\[
(v'^2) = (v'^{-2 \text{max}}) \cdot h_2(\alpha, \phi, \phi_{12})
\]
(25)

with \( \alpha \) and \( \phi \) polynomial functions of \( \phi_{12} \) and \( \phi_{12} \).
polynomial function of \( n, S \) and \( J_{12k} \). Combining relations (20) to (25) we obtain (Kaldellis, 1988b) relations of the type:

\[
\begin{align*}
\left( u'^2 \right) &= \left( u'^2 \right)_{\text{max}}^s \\
\left( u'^2 \right) &= \left( u'^2 \right)_{\text{max}}^p \\
\left( \omega^2 \right) &= \left( \omega^2 \right)_{\text{max}}^s \\
\left( \omega^2 \right) &= \left( \omega^2 \right)_{\text{max}}^p
\end{align*}
\]

(26)

(27)

The last two relations constitute parametric distributions of the turbulent components which not only give a qualitative representation of the real flow distributions, but also approximate quite accurately the corresponding experimental data. The requested values of \( Ak \) (or A) and \( Bk \) (or B) integrals are obtained by numerically integrating the distributions of the fluctuation components involved in definitions (18) and (19).

Profile losses algorithm

Adopting for simplicity the incompressible relation for the definition of the total pressure, we may express the total pressure drop at a point \((S, n_e)\) of the cascade passage as:

\[
\Delta P = \frac{1}{\rho u^2} \left( \frac{1}{\rho u^2} - 1 \right) \Delta P_t
\]

(28)

where \( u \) is the velocity profile at a certain position calculated as a function of the shear layer parameters (Kaldellis and Ktenidis, 1990b) and \( \Delta P_t \) is the static pressure difference between the internal and the real flow field. The \( 6p \) term, commonly neglected in usual analyses, is calculated from the integral momentum equation in the \((\gamma e)\) direction (eq. 4) and is related to the normal to the blade surface Reynolds stresses and the longitudinal curvature of the external flow streamlines. Its contribution to the total pressure drop is quite important (up to 12%) especially for fully separated cases (Kaldellis and Ktenidis, 1990b).

The evolution of the profile losses along the blade is subsequently expressed as:

\[
\omega^2 (S_e, i) = \Delta \bar{P} / (\rho u^2)
\]

(29)

where \( \Delta \bar{P} \) is the mass-averaged total pressure drop and is finally given by the following equation:

\[
\Delta \bar{P} = \frac{\left( \rho \int u^2 \bar{P} \left( \text{d}n \right) / m \right)}{\rho u^2}
\]

(30)

Consequently, the evolution of \( \omega^2 \) through the cascade can be computed once the velocity profile and the total pressure drop distribution are known at the desired normal. Up to now the value of \( \omega^2 \) was given only at the cascade’s exit using semi-empirical correlations. Equation (29) gives us the additional possibility to calculate the evolution of the profile losses throughout the cascade and not only at the exit.

4. CALCULATION RESULTS AND DISCUSSION.

Part one: Fully separated internal flows.

The proposed blade shear layer calculation method has been successfully tested by the authors for a variety of test cases including cases with extended flow separation. The need for a strongly coupling viscous-inviscid interaction scheme guided us to base our method on the simultaneous solution of the interaction equation (modification of the external flow) and the equations that describe both shear layers. Subsequently, a Gauss-Newton type algorithm, developed for the minimization of the sum of the squared residuals of the equations under consideration (Kaldellis and Ktenidis, 1990b) provides the method with the ability to confront the complexities related to cases of fully separated internal flows.

Moore large flow.

The first test case selected concerns an internal shear flow (Moore, 1973) of a fully stalled diffuser. In subfigures \([lb]\) and \([lc]\) the velocity distributions for the external and the modified external flow field are presented in comparison with experimental data. The experimental values correspond to the measured edge of the suction and pressure side shear layers. In the same figure the evolution of the blockage factor \( \mu \) induced by the two wall shear layers is also given. Note the large values of \( \mu \) and its significant influence on the external velocity level towards the measured values. In subfigures \([lb]\) and \([lc]\) the corresponding distributions of \( \bar{S} \) and \( \bar{B} \) for the two wall shear layers are favourably compared to experimental data and the results of another theoretical method (Moore, 1973). Finally, in subfigure \([ld]\) the corresponding distribution of the entire profile losses (i.e. total pressure losses) to such distribution is given by Moore (1973). Closing the analysis of the test case it is interesting to present the iso-mach contours (figure \([2]\)) predicted by our viscous-inviscid calculation scheme. The real flow field is quite realistically represented, demonstrating the existence of high loss core near the suction side of the blade. The interaction scheme guided us to base our method on the simultaneous solution of the interaction equation for the cascade flow - Profile losses.

The next test case considers the results of a classical NACA 65-(12)10 compressor cascade which is systematically analyzed for various solidity values and for a wide range of incidence angles. It takes approximately 2 minutes CPU time on a 25Mhz 366 machine to predict the distribution of the profile loss coefficient as a function of the incidence angle and the channel length (i.e. \( \omega^2 (S_e) \)) for a solidity value \( \sigma \) equal to 0.50. Figure \([3]\) gives a qualitative picture of the distribution of the profile losses along the cascade for the operating incidence range. A peak value of \( \omega^2 \) is encountered for highly positive incidence angles near the cascade’s trailing edge due to the appearance of flow separation. The calculated cascade profile loss at the exit of the blading (figure \([4]\)) is in excellent agreement with the experimental data (Herring et al., 1957). Besides, the results obtained by semi-empirical profile loss correlations (Koch and Serizy, 1976) are also shown in the same figure, especially for positive incidence angles due to the prediction of early flow separation. This is a typical problem of methods based only on diffusion factor blade loading criteria.

In order to underline the abilities of the proposed calculation frame, the total pressure drop throughout the cascade passage is quantitatively shown in figures \([5a]\) and \([5b]\) for the highest off-design value of the incidence angles investigated \( (\sigma = 2) \). Note that the total pressure loss increase is related to the thickening of the two shear layers, especially of the one on the suction side of the blade, when separation is encountered. Although the peak value of the total pressure drop appears near the leading edge of the blade, the excessive profile loss values are mainly related to the increase of the shear layer thickness. For the present case the \( 6p \) term has a negative contribution of approximately 8% to the exit loss value.

Part two: Axial turbines.

ECL Turbine.

After presenting some of the main features of our blade shear layer calculation code we proceed now to analyze some 3-D configurations of axial turbines and compressors with industrial interest.

The first turbine test case investigated (figure \([6]\)) has been experimentally studied by Orban (1983) at Ecole Centrale de Lyon. In our previous work (Kaldellis and Ktenidis, 1990b) the secondary flow field quantities at the turbine exit were calculated. The external flow deflection angle is equal to 60° while the inlet endwall shear layer has \( H_{12} = 1.30 \) and \( 6_{12} = 3.63 \) mm. It is important to note that the inlet endwall shear layer is already three-dimensional, leading to a remarkable variation of the inlet flow angle (figure \([7]\)). This results in a corresponding variation of the flow incidence angle. Even this moderate secondary flow field influence the loading of the blades of the cascade. The calculated blade velocity distributions at three different spanwise locations on the endwall (figure \([8]\)) in comparison with experimental data. A decrease of the blade loading is predicted for negative incidence angles. The calculated boundary layer airfoils at the endwall are quite uniformly loaded from the leading to the trailing edge. Similar results have been obtained by Tremblay et al. (1990) during their detailed experimental study of a linear cascade of a turbine, where the incidence varied between -25 and 25 degrees off-design.

The calculated profile losses due to the blade shear layer calculation code is compared to experimental boundary layer airfoils computed by our secondary flow code (Kaldellis and Ktenidis, 1990b) and the total pressure values are estimated as:

\[
P_{\text{f}} = P_{\text{f}0} - \Delta P_{\text{f}0} - \Delta P_{\text{f}r}
\]

(31)

The resulting total pressure contours are given in
figures [9a], [9b] and [9c] in comparison with the experimental measurements (figures [9d], [9e], [9f]) at the turbine's mid-span. The overall agreement between the calculated results and the experimental data is quite good. The secondary loss values are satisfactorily described for engineering purposes. The predicted total pressure loss contours show a similar behaviour with the experimental results given by many other researchers (Zumino et al., 1987; Gregory-Smith et al., 1988). According to our investigation, excessive loss values exist mainly due to the inlet endwall shear layer located near the endwall. As a result, the corresponding calculated secondary losses are satisfactorily compared to the experimental measurements. In figure [10a], the experimental measurements are given. As observed in the experimental data, the inlet total pressure loss is comparable to the corresponding secondary and mixing loss values.

TUS Axial Turbine

In order to support our opinion we subsequently investigate the loss evolution through an industrial turbine cascade (for details see Kaldellis and Ktenidis 1990b). The cascade is a high-turning one with a flow deflection greater than 100° and a quite thick inlet shear layer (H=1.42 and H=5.44 mm). In figure [11a] the spanwise evolution of the turbine profile, secondary loss, wake loss and wash along with the corresponding results of the experimental measurements are given. As observed in the previous case, the loss core is encountered at about 1/3 of the endwall shear layer thickness. Although the profile losses constitute a relatively small part of the maximum loss, the total loss, resulting from the sum of the above mentioned distributions, is favourably compared with experimental data. It is interesting to mention that a) the profile loss is large when secondary shear layers are thin, b) a secondary loss peak exists at a distance of 3 mm from the endwall and c) the secondary losses are more near the trailing edge of the blade shear layers. A better picture of the losses field is obtained from the results concerning the station near the blade's trailing edge (figure [13b]). The profile and net secondary losses spanwise distributions are satisfactorily compared to the experimental measurements.

As expected, the profile losses are higher near the hub than near the midspan since highly positive incidence angles are imposed by the secondary flow field near the endwall. The predicted method agrees very well with the analysis of Roberts et al. (1988). Subsequently, the secondary losses present a distribution similar to that of figure [15a] but the maximum secondary loss takes now a much greater value (w=0.29) and appears at a distance of 41 mm from the hub (near the middle of the hub shear layer). Reasonable agreement is achieved by comparing our results with the ones obtained from the semi-empirical formulae proposed by Roberts et al. (1988) for compressor stator cases. Using the analysis of Roberts et al. (1988) the maximum secondary loss takes a value of 0.176 while the proposed method predicts a maximum value of 0.29 which is accepted as experimental peak value. Additionally, the location of the maximum secondary loss and the extent of the secondary loss region are only fairly predicted by the above mentioned empirical correlation. In that respect, the correlations investigated seem to give acceptable overall results and provide a first approximation of the secondary loss features. Although the secondary losses are clearly visible, the computer effort, they do not take into account the local characteristics of the secondary flow field and therefore cannot describe in detail the large variety of the existing configurations. For the present test case the proposed method requires approximately 17 minutes CPU time on a 32 MHz Sun workstation for each overall iteration (including blade-to-blade and secondary flow calculation), mainly depending on the number of blade-to-blade sections used. Up to six overall iterations are needed to achieve convergence to the machine's single precision.

Another interesting characteristic predicted by the proposed methodology is the distribution of the displacement thicknesses induced by the blade and the endwall shear layers. Figures [16a], [16b], [16c] demonstrate the local value of the displacement thickness along the edge of the cascade and near the midchord and the trailing edge of the blades. The circumferentially averaged values (obtained at station 6) are also provided in figure [16d]. The secondary flow field dominates the effective passage area up to the midchord. However, near the exit of the blade the contribution of the secondary flow core, which is quite important, is also interesting to examine the evolution of the viscous-inviscid code (as defined by the potential theory) through-
out the cascade (figures [17a, 17b, 17c]). As it was also shown by the experiment, the endwall shear layers tend to fill the cascade passage while the blade suction side shear layer is quite thick. Although the corner flow field is not investigated here both the displacement thickness and the shear layer thickness occupied areas as well as the total pressure loss distribution are scale effects of the high degree near the suction side of the blade. The numerous studies made during the last years, mainly experimental, for the investigation of corner flows (e.g. Bharath et al., 1987, Abdulla et al., 1990), provide us with the main physical information in order to include this important phenomenon in our theoretical model in the near future.

Finally it is important to keep in mind, as elsewhere stated (Kaldellis et al., 1988a), that the secondary flow field modifies the outlet spanwise flow angle distribution even in the inviscid flow core. This is the result of the reaction of the secondary vorticity (passage vortex) to the space confinement. In this way, the deviation angle is modified both due to the incidence change imposed by the inlet secondary flow field and due to the deviation change induced by the secondary vorticity at the exit of the cascade. The latter effect is not taken into account by the semi-empirical correlations by Roberts et al. (1988) and Cetin et al. (1989). However, the deviation angles do influence the incidence angle of a blade row possibly following. It is well known that small errors in the incidence angle affect the performance of an axial compressor in a thoroughly and cumulative way especially in cases of multistage machines.

IGV - Transonic Axial Compressor.

The influence of the flow field developing through the incidence of a compressor upon the initial aerodynamic conditions imposed on the rotor of the machine is examined as another example of the interaction between the secondary flow field and the blade loading. Due to space limitations no additional information about the experimental set up and measurements can be given here (see Leboeuf and Naviere, 1983) other than the nominal rotation speed of 60,000 rpm corresponding of the nominal stage pressure ratio $\text{He}=1.38$ and the nominal mass flow rate $m=16.0$ kg/sec, the nominal stage pressure loss $\Delta p=1.38$ and the nominal rotation speed $n=11,500$ rpm. More information about the flow field of the compressor stage under consideration is given by Kaldellis et al. (1989a).

In figure [18] the complete picture of the total pressure field is given at the outlet of the IGV in comparison with experimental measurements by Leboeuf (1984). The differences near the hub are due to the fact that the leakage effects were assumed negligible since they cause only a local disturbance to the flow field. Although these secondary endwall shear layers can be clearly observed in both figures [18] and [19], which describe the total pressure and static pressure distributions at a pressure ratio of the nominal rotation speed, respectively. In this last figure the circumferentially averaged spanwise distributions of total pressure loss for successive sections of the nominal rotation speed is presented in comparison with experimental data. Even the moderate 3-D flow field leads to circumferentially averaged flow angle distributions similar to the ones given in figure [20] in comparison with experimental measurements. The overturning of the flow field inside the two endwalls (approaching the value of $10^\circ$ near the walls) strongly modifies the flow incidence angle and therefore the blade loading of the next blade row.

According to the design procedure, the design incidence angle is linearly starting from 1 at the hub to 2 at the tip of the blade. The rotor of the compressor is composed of 8 DCA type blades. The induced, due to secondary flow effects, change of incidence is given in figure [21].

The complete loss analysis of the rotor must take into account the evolution of the added work (see Kaldellis et al., 1990a) besides the secondary and profile losses in order to predict the energy exchange process. Since our primary target is the evolution of the 3-D loss field through axial configurations and the interaction of the secondary flow field with the blade shear layers, we shall focus our analysis on the off-design operation of the blades due to the interaction of the secondary flow and the blade shear layers is examined and incorporated in the design procedure. Finally, the validity of various recently presented sophisticated correlations, for both axial compressors and turbines, is verified. It is encouraging that the majority of them shows an acceptable behavior for engineering purposes. However, since the local endwall and blade shear layer characteristics are not taken into account, their accuracy, especially near the endwalls and in cases of strong secondary vorticity, is limited.

The decision to use quasi-3D analytical tools is validated not only by the accuracy of the results obtained, but also because such numerical codes show remarkable reliability and robustness with very low computational cost.

Appendix One

Coefficients of the basic integral equations

The coefficients appearing in the basic integral equations (1.1), (1.2), (1.3) which describe the development of attached or detached shear layers, after the application of Truckenbrodt's (1952) transformation are the following:

$$A_1=\frac{1}{4}(H_2-1)^{-1}$$

$$A_2=\frac{1}{4}H_2$$

$$A_3=\frac{1}{4}(H_2-1)$$

The contributions of the static pressure difference between the external inviscid and the real flow fields is taken into account in order to provide a better correlation. For the calculation of the static pressure difference and for the amplification of the profiles, the corresponding governing equations an improved turbulence model is also embodied in the complete computational frame.

The contribution of the static pressure difference between the external inviscid and the real flow fields is taken into account by these semi-em-

REFERENCES


Figure 1: Comparison between calculation results and experimental data for internal fully separated diffuser shear flow, "Moore large flow" test case.

Figure 2: Iso-mach lines computed by the proposed strong coupling viscous-inviscid interaction scheme for the "Moore large flow" fully separated test case.

Figure 3: Schematic representation of the total pressure drop coefficient distribution along a cascade (NACA 65-(12)10 $\sigma=0.50$) for the operating incidence range.

Figure 4: Distribution of the mean profile $\alpha'$ coefficient at design and off-design incidence angles. Comparison between calculated, experimental and empirical correlation results.
velocity distribution

Figure 6: Schematic representation of the ECL turbine cascade indicating the computational stations where comparison between the theoretical and the experimental results is performed.

Figure 7: Pitchwise distribution of the circumferentially averaged flow angle at the inlet of the ECL turbine.

Figure 8: Blade surface velocity distributions at various distances from the hub. Comparison between theory and experiment.

Figure 9: Total pressure contours at selected SI planes along an axial turbine passage. Comparison between theory (a, b, c) and experiments (d, e, f) by Onvani (1983).
Figure 10: Total pressure loss distribution for the ECL axial turbine. Comparison between theory & experiment.

Figure 11a: Net total pressure loss distribution for the TUS axial turbine. Comparison between theory & experiment.

Figure 11b: Total pressure loss distribution for the TUS axial turbine. Comparison between theory & experiment.

Figure 12: Schematic representation of the experimental cascade indicating the computational stations where the comparison of the theoretical and experimental results is performed.

Figure 13: Inlet transverse velocity profile for the compressor cascade of Flot (1975). Comparison between theory and experiment.

Figure 14: Internal shear flow calculation results for "Flot case B" cascade configuration.
Figure 15a: Spanwise total pressure loss distribution near the midchord of Plot B compressor cascade.

(a) eff. area 91.5%
(b) eff. area 72.1%
(c) eff. area 72.8%
- s.l. block. 8.5%
- mech. block. 0%
- s.l. block. 15.5%
- mech. block. 12.4%
- s.l. block. 24.6%
- mech. block. 2.6%

Figure 15b: Spanwise total pressure loss distribution near the outlet of Plot B compressor cascade.

(a) inv. core 45.7%
(b) inv. core 44%
(c) inv. core 36.1%
- visc. zone 54.3%
- visc. zone 43.6%
- visc. zone 61.3%
- blade area 0%
- blade area 12.4%
- blade area 2.6%

Figure 16: Effective passage area for the cascade channel at x/c=0%, 44% and 88%.

(a) x/c=0.00
(b) x/c=0.44
(c) x/c=0.88

Figure 17: Inviscid flow core at x/c=0%, 44% and 88% respectively.
Figure 18: Total pressure contours at the exit of the IGV of an axial transonic compressor working at 95% of its nominal rotation speed. Comparison between (a) present calculation and (b) experiments by Leboeuf (1984).

Figure 19: Spanwise distribution of circumferentially averaged total pressure at the exit of the IGV of a transonic compressor. Comparison between theory & experiment.

Figure 20: Spanwise distribution of circumferentially averaged flow absolute angle at the exit of the IGV of a transonic compressor. Comparison between theory & experiment.

Figure 21: Spanwise distribution of the flow incidence angle for the rotor of an axial transonic compressor. Comparison between design values, calculated results and experimental data.

Figure 22: Spanwise distribution of the diffusion factor for the rotor of an axial transonic compressor. Comparison between design values, calculated results and experimental data.