Computational Study of Asynchronous Vibrations in a 1.5 Stage Transonic Compressor

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The unsteady RANS equations are solved for the internal flow through the first one and a half stages of a transonic compressor operating in an off-design condition. To establish a steady performance prediction, the mixing-plane method is used and reasonable comparison to experimental data is obtained for the design configuration. To predict the fully unsteady flow, the sliding-mesh method is used. Examining the unsteady force on the blades, substantial subharmonic force fluctuations are shown to exist. The unsteady pressure signals from the domain also support the existence of these fluctuations. Views of the time-mean and notch filtered flow help to highlight the major features of the off-design flow field, and identify how these features relate to the subharmonic fluctuations in force and pressure.

I. Introduction

The desire to use turbomachines over a wide range of operating conditions presents challenges for the designer. One problem that arises is that the flow may become unstable at some condition within the design range, potentially leading to asynchronous oscillations, or flow oscillations not linked to shaft speed. If these oscillations are severe, they will contribute to significant unsteady loading of the blades, which may fatigue and/or lead to their catastrophic failure. If this fate is to be avoided, the designer must be aware of the possible sources of these oscillations so that he may account for them in his design. Because turbomachines are highly dynamic, there are many ways in which harmful vibrations may be internally generated. One important source of these vibrations is the unsteady flow field that results from the interaction of consecutive blade rows in multi-stage turbomachines.

The practice of using CFD solvers to obtain “steady” flow solutions for three-dimensional isolated turbomachinery blade rows has been evolving for about twenty years. These methods have matured and gained ground in recent years so that they are now an industry standard for airfoil design. However, CFD methods for considering the effects of multi-stage interactions in turbomachine flows are considerably less mature. Since these effects can have a large impact on the resulting flow, this represents a serious limitation to the current CFD capabilities available to the designer.

Responding to this limitation, several researchers have worked on the Mixing-plane method1–7 to approximate the unsteady flow field that occurs in multistage turbomachines with a steady flow. The idea is that by adding a mixing layer between adjacent blade rows, the flow can essentially “mix” and thus smooth out the unsteady variations that would occur in a real multistage machine. While this approach neglects the real unsteadiness of the flow, it has provided useful insight regarding the general performance of multi-stage machines.

As an attempt to increase the fidelity of these simulations, considerable work has been done on methods which model the effects of synchronous unsteadiness in the flow through multi-stage turbomachines. Both the Average Passage8 and time linearized methods9 improve on the “steady” solution of the mixing-plane approach by accounting for these effects. The Average Passage method accomplishes this by time averaging the flow equations over one shaft rotation and obtaining additional stress correlations in the governing equations. Time linearized methods account for the synchronous unsteadiness by assuming a steady mean flow with periodic perturbations linked to shaft speed. While these methods represent improvements to the
mixing-plane method, they account only for unsteadiness linked to shaft speed and so are not suitable for resolving the effects of asynchronous unsteadiness which may be important to turbomachine performance and durability.

To resolve the fully unsteady flow through a multi-stage turbomachine, a sliding-mesh method may be used. This sliding-mesh method is an interpolation scheme which allows the flow to remain continuous across blade non-matching row interfaces even when they move relative to each other. A major reason that fully unsteady flow computations have been avoided is the large computational costs associated with them. However, in order to resolve the potentially harmful asynchronous unsteadiness that may result in a multi-stage turbomachine, it is a necessity.

A. Current Work

In this work “steady” flow solutions for a multi-stage turbomachine are obtained using the Mixing-plane approach. This solution is used to validate the flow solver for machine performance predictions. Proceeding from these results, the unsteady flow through the multi-stage machine is solved using the sliding-mesh method. The resulting unsteady flow field is analyzed, with particular attention given to potential sources of asynchronous flow oscillations.

Section II describes the numerical methods used for this work. In section III the results of both the steady and unsteady computations are presented, and Section IV gives some conclusions and suggested future work.

II. Computational Methods

A. Governing Equations

A density-based, finite-volume method is used to solve the unsteady Favre-averaged Navier-Stokes equations which may be expressed in differential form as:

\[
\begin{align*}
\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i}(\rho \tilde{u}_i) &= 0 \\
\frac{\partial \rho \tilde{u}_i}{\partial t} + \frac{\partial}{\partial x_i}(\rho \tilde{u}_i \tilde{u}_j) &= \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left( \tau_{ij} + \rho u'_i u'_j \right) \\
\frac{\partial \rho E}{\partial t} + \frac{\partial}{\partial x_i}((\rho \tilde{u}_j) H) &= \frac{\partial}{\partial x_j} \left[ \tilde{u}_i \left( \tau_{ij} - \rho u'_i u'_j \right) \right] \\
&\quad - \frac{\partial}{\partial x_j} \left[ q_j + \rho u'_j h' - \tau_{ij} u'_i + \rho u'_j u'_i \right] \end{align*}
\]

In order to close this set of equations, we also require the equation of state

\[ p = \rho RT, \]

the relationship between energy and temperature for a calorically perfect gas

\[ e = c_v T, \]

and a turbulence model for predictions of the Reynolds stresses and turbulent heat flux. The one-equation Spalart-Almaras turbulence model is solved to account for the turbulent stress and heat flux terms which result from the Favre-averaging.

B. Flow Solver

The second order accurate JST scheme is used to discretize the fluxes, so that the discrete form of the governing equations is

\[
\frac{\partial}{\partial t} (W \Delta V)_{i,j,k} = -Q_{i,j,k} + D_{i,j,k} = R(W)_{i,j,k}
\]

with the flow vector \( W \)

\[
W = \begin{bmatrix}
\rho \\
\rho u \\
\rho E
\end{bmatrix}
\]

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and with $D$ as the artificial dissipation, $R$ the residual, and $Q$, the sum of the fluxes over the surface of the volume, defined as

$$Q_{i,j,k} = \sum_{l=1}^{6} F_l \Delta S_l$$  \hspace{1cm} (2)

To solve the unsteady flow equations, a dual-time stepping procedure is used. A second order backward-difference implicit operator is used to discretize the unsteady term. An efficient explicit multistage Runge-Kutta method is used to march the equations in pseudotime ($t^*$), so that we essentially solve a steady state problem at each real time step. The expression for the pseudotime marching is

$$\frac{\partial}{\partial t^*} W_{n+1} = \frac{1}{\Delta V_{n+1}} R^* (W_{n+1})$$  \hspace{1cm} (3)

where the pseudo residual $R^*$ is defined as

$$R^* (W_{n+1}) = R (W_{n+1}) - \frac{3 (W \Delta V)^{n+1} - 4 (W \Delta V)^n + (W \Delta V)^{n-1}}{2 \Delta t}$$  \hspace{1cm} (4)

\section*{C. Mixing-plane Boundary Condition}

To allow a steady flow solution through multi-stage turbomachines, the mixing-plane boundary condition is used. First, the interface is divided into circumferential sections by the grid lines with approximately constant radius. Next, at each iteration of the flow solver, mass averaging of the flow variables is performed over these sections to obtain radial profiles of the corresponding flow variables. These profiles are then exchanged from one side of the interface to the other and applied as boundary conditions. In this way, when a converged solution is obtained, the average flow variables at the mixing plane will be equivalent and thus mass, momentum and energy will be conserved.

It should be noted that, as pointed out by Denton,\textsuperscript{1} some entropy will be generated by this mixing process. This entropy gain can be thought of as an artificially introduced loss mechanism. It has been demonstrated that this loss will exceed the loss generated in the actual mixing which occurs at multi-stage interfaces. For this reason, we should expect our performance predictions from the mixing-plane method to under predict the efficiency of the real machine.

\section*{D. Sliding-mesh Method}

To obtain a solution for the fully unsteady flow through a multi-stage turbomachine, a sliding-mesh method is used. This method is a means of interpolating the flow variables across a non-matching and moving interface so that flow remains continuous. We use a bi-linear interpolation scheme, which decomposes the flow variables by radius and circumferential angle $\theta$. Using this approach, we must compute linear interpolation coefficients which map each cell center on one side of the interface to the surrounding four cell centers on the opposing side.

To reduce the computational effort, cells at the interface are required to have constant radius at top and bottom surfaces. With this restriction, the radial interpolation coefficients at the interface will not change when the grids move relative to each other. This allows us to compute the radial interpolation coefficients before starting the computation, and only update the $\theta$ coefficients at each time step.

\section*{E. Numerical Considerations}

The tip gap was modeled with a tip grid consisting of 8 cells in the span-wise direction. When gridding the domain, the mixing-plane (and sliding-mesh) interfaces had to be inclined with respect to the rotational axis in order to leave sufficient space between the blade rows. This required some modifications to the traditional mixing-plane/sliding-mesh methods, and prevented using the same grids for the mixing-plane and sliding-mesh computations. The grid used for the mixing-plane computations required modeling one passage for each blade row, and consisted of approximately 500,000 nodes. The sliding-mesh computations required modeling five passages, one rotor and two each of the IGV and stator blade rows, and had a total of 700,000 nodes.

Mixing-plane solutions were used as the initial condition for the unsteady sliding mesh computations. The mixing-plane solutions were run until mass flow rates across the interfaces converged to within 0.5 %.
Figure 1. Operating Configurations for the 1.5 Stage Compressor

Figure 2. Side View of the 1.5 Stage Compressor

III. Results

A. Siemens 1.5 Stage Compressor (S15)

The results presented here are for the first one and a half stages of a multi-stage compressor design by Siemens. The domain forms the inlet to the compressor, and is composed of an inlet guide vane (IGV) a rotor and a stator. Figure 2 shows the domain from the side view, Here we consider two geometrically distinct configurations, referred to as the design configuration and off-design configuration. Figure 1 shows one passage from each blade row as a 2D top view for both configurations. Notice that in the off-design configuration, the IGV is closed substantially. This enables the machine to operate in a part-load state and avoid flow separation through the rotor stage. Since the sliding mesh computations require that the computational domain for each blade row have the same circumferential pitch, the grids for the stator blade row were scaled down by 5% to achieve a blade count ratio of 1:2. Along with scaling the domain, the stator blades were also scaled in pitch and chord length, to keep approximately the same solidity.

B. Steady Performance

In order to ensure that the flow through the S15 case is accurately modeled, the steady flow through the machine was computed using the aforementioned mixing-plane method. Figure 3(a) shows the performance for the design configuration as compared with experimental data provided by Siemens. Relatively good
agreement between the computed and experimental pressure ratio data is achieved. Discrepancies in the predicted adiabatic efficiency of the machine are suspected to be, at least in part, due to the excess loss generated by the mixing-plane model at the blade row interfaces. Figure 3(b) shows the pressure ratio versus corrected mass flow rate for the off-design configuration. Relatively little experimental data is available for this case, but the general trend of the prediction is as expected. Note that the mass flow of the machine is approximately half of that for the design configuration.

For both the Design and Off-design cases, one or more data points are provided for comparison with the unsteady results obtained using the sliding-mesh method. For the Design case, the difference is significant, and the time-averaged pressure ratio from the unsteady result appears to agree much better with the experimental data than does the steady (mixing-plane) result. There is some improvement in the agreement of the adiabatic efficient as well. However, a significant discrepancy still exists between the experimental data and the time-averaged unsteady result. One contributor to this discrepancy is the increase in blade surface area due to the geometry scaling used for the unsteady computations.

Interestingly, for the Off-design case, very little difference is observed between the steady and the time-averaged unsteady results. This may be due to the large separation which occurs in the stator section for both the steady and unsteady results. This is in contrast to the Design case, where the steady result shows some separation in the stator blade row, while the unsteady result does not.

C. Unsteady Aerodynamics for the Off-design Configuration

Attempting to identify and characterize the asynchronous unsteadiness observed in the off-design configuration, fully unsteady flow computations were performed for both the off-design and design configurations. The results obtained from these computations will be presented here. First, we will examine the unsteady force acting on the rotor blade to determine whether any asynchronous unsteadiness may be observed. Second, pressure signals from various locations in the flow domain will be examined. Third, contours of flow variables from the time-averaged flow field will be presented. Fourth, notch-filtered contours of flow variables from the unsteady flow field are shown to help isolate the location, magnitude and dynamics of the asynchronous unsteadiness in the flow field. Next, a few comments are made about the effects of domain choice on the character of the unsteady results. Finally, a summary description of the observed features from the unsteady flow through the S15 compressor in the off-design configuration is presented.
1. Modal Projection of Rotor Blade Force

At each time step, the force acting on the blade, including components from both pressure and viscous stresses, was computed for the rotor. To examine the potential of these forces to excite structural vibration in the rotor, this force was projected on to the first natural mode shape of the rotor, seen in Figure 4. The undeformed blade is shown in black. This mode shape was obtained from a structural analysis of the rotor at design conditions. The significance of this projection can be seen by examining the structural equations of motion for the rotor blade under unsteady aerodynamic loading.

The structural equations of motion for a system with a finite number of degrees of freedom are:

\[
M\ddot{q} + C\dot{q} + Kq = F
\]  \tag{5}

where \(q\) is the displacement vector, \(M\) is the mass matrix, \(K\) is the stiffness matrix and \(F\) is the loading vector for the system. For small displacements \(q\), we may assume a modal solution of the form

\[
q = \sum_{i=1}^{N} \eta_i \Phi_i
\]  \tag{6}

where \(\Phi_i\) are the eigenmodes and \(\eta_i\) are the generalized displacements for each mode. If the system of equations is diagonalized using the matrix of eigenvectors \(\Phi\), we obtain an expression for the generalized motion of the structure as

\[
\ddot{\eta}_i + 2\zeta_i \omega_i \dot{\eta}_i + \omega_i^2 \eta_i = Q_i
\]  \tag{7}

where the aerodynamic forces \(F\) are projected onto the mode shapes to obtain the generalized forces \(Q_i\), for each mode.

\[
Q_i = \Phi_i^T F
\]

Also, the eigenmodes are related to the eigenfrequencies and mass matrix through the following expressions

\[
\omega_i^2 = \Phi_i^T [K] \Phi_i, \quad \Phi_i^T [M] \Phi_i = 1
\]

Examining Equation 7, it is apparent that projection of the aerodynamic loads on to the mode shape \((\Phi_i^T F)\) is the “forcing function” for the generalized equation of motion for a given mode. As such, it is expected that if this unsteady aerodynamic forcing has a significant periodic component with a frequency near the natural frequency of the first eigenmode blade, vibration may be excited in that mode.

Figure 5 shows the Fourier transform of the fluctuations in the modal projection of the aerodynamic force for both the Design and Off-design configurations. The frequencies in these plots are normalized by the
passing frequency of the rotor blade row. Figure 5(a) shows the transform up to the first three harmonics of the passing frequency. Here we see that the dominant component of the unsteady forcing on the blade is due to the passing itself. Also, the response from the first harmonic is of the same order of magnitude as the response from the passing frequency. This is due to the 2:1:2 blade count ratio used to perform the sliding-mesh computations for this geometry.

Figure 5(b) shows a close-up of the subharmonic components of the Fourier transform. Even though the magnitude of the response is several orders of magnitude smaller, there is a noticeable subharmonic component of the modal blade force for the Off-design configuration. This component peaks at about one quarter of the passing frequency. Indicated with the dotted line is the natural frequency of the first eigenmode of the rotor blade. This natural frequency is about one eighth of the passing frequency, or about one half of the peak subharmonic force response. Even though the peak of the response does not coincide with the natural frequency of the rotor blade, the fact that it is “in the ballpark” gives reason to suspect some excitation may occur.

It should also be noted that there are several reasons to suspect the frequency of the peak subharmonic response predicted by the current simulations may vary from the value observed in the actual machine. First, this simulation has been conducted with a slight scaling of the blade count ratio. When using the 2:1:2 blade count ratio, and circumferentially periodic boundary conditions we are enforcing a zero nodal diameter periodicity to the flow solution which may not exist in actuality. This constraint may alter the resulting frequency of any unsteady flow oscillations. Secondly, in the real machine, when the blade is forced at a frequency near one of its natural frequency, it will start to vibrate. Once vibration has been established, the forcing frequency itself may change (as in the case of “lock-in”) due to the nonlinear interaction between the fluid and structure.

2. Pressure Probe Data

Figures 6(a), 7(a) and 8(a) are diagrams depicting the locations at which the pressure was monitored during the unsteady flow solutions. The adjacent subfigures are plots of the Fourier transform of the normalized pressure fluctuations obtained at these locations. For the IGV and Stator, pressure was monitored at three different span-wise planes. In the Rotor blade row, the pressure was monitored in a “halo” about the rotor tip gap region, as well as in the tip gap itself.

Like with the rotor blade force, all of the Fourier transforms of the pressure fluctuations are dominated by the response at the passing frequency and the first harmonic. Here we show only the subharmonic portion of the pressure fluctuation Fourier transforms. For the IGV (Figure 6), there is a significant component of the response at about one quarter of the passing frequency for both the bottom plane and the top plane,
while the middle plane shows significantly less response.

For the rotor blade row, the magnitude of the response is somewhat larger, but the peak frequency is also at one quarter of the passing frequency for all pressure probe locations around and within the tip gap. In the rotor blade row, there are also several probes which show significant response at one half of the blade passing frequency, namely those probes which are approximately near the trailing edge on the suction side of the rotor.

There are a few striking features of the subharmonic components of the pressure fluctuations in the stator blade row. First, the subharmonic response in the stator blade row is of higher magnitude than in either the IGV or the rotor blade row. Second, the magnitude is largest in the bottom plane pressure probes. Third, the frequency of the response is lower than that in the other blade rows, and also much closer to the natural frequency of the rotor blade row.

For reference, Figure 16 shows the Fourier transform of the unsteady pressure fluctuations on the bottom plane from the design configuration sliding-mesh computation. Though not shown in the figure, the response at the rotor passing frequency and first harmonic are of equal magnitude to those for the off-design case. However, as the figures shows, there is almost no subharmonic response in this case. This supports the proposal that the subharmonic pressure fluctuations discussed above are a result of the geometry and flow conditions of the Off-design configuration, not the machine in general.

D. Time-Mean Flow Field

The time-mean flow field was obtained by averaging over the final eight passing periods of the unsteady computations. Figures 9, 10 and 11 are contours of flow variables taken from this time-mean flow field. All three figures contain slices through the flow field taken at values of 20, 50 and 80 percent span. You will note that when viewed in this manner, the flow field is not continuous across the sliding-mesh interface. This is due to the relative motion of the rotor with respect to the IGV and the stator. Spatial fluctuations of flow variables in a given passage may be viewed as temporal variations from the neighboring, moving, blade row. When the flow field is time averaged, these temporal fluctuations are “averaged” out. The resulting time-averaged flow field then shows variations within each blade row which are due primarily to the local blade.

Figure 9 contains Mach contours which are helpful for highlighting some of the major features of the flow. Figure 9(a) shows that at 20% span, the wake from the IGV is relatively severe. Notice the appearance of high Mach regions at the leading edge of the rotor on the pressure side, and at about three quarters chord on the suction side. Focusing on the rotor leading edge, it appears that the incidence angle of the oncoming flow may be too high to avoid this high Mach region and the small separation region that results. There is also a high Mach region in the stator blade row near the leading edge on the suction side. This results in a severe separation region which extends to the exit of the domain.

Moving on to the higher span slices, we note some significant differences between the 20% and 50% and 80% span slices. At 50%, the severity of the IGV wake is decreased, and the high mach region which appeared at the rotor leading edge in the 20% span splice does not appear. The separation region in the stator has increased in size, and the incident mach number at the stator leading edge has decreased. At 80% span, we see that the wake from the IGV is not apparent, and the separation in the stator has decreased significantly in width.

Figure 10 shows contours of nondimensionalized pressure from the time-averaged flow field. At 20% span, we see low pressure regions due to the separation over the suction side of the IGV and at the leading edge of the rotor. The leading edge of the stator also shows a low pressure region which will undoubtedly have an impact on the flow near the trailing edge of the rotor as two blades have a relatively small axial gap at this span-wise location. At 50% span we see, that the flow field appears to be fairly contained within each blade row, with the exception of the high pressure region which appears at the leading edge of the stator on the pressure side. At 80% span the most apparent flow feature is the large gradient in pressure across the stator passage. This large pressure change is accompanied by the compressed separation region and high Mach region at the stator leading edge.

Figure 11 shows contours of the nondimensionalized radial component of vorticity from the time-averaged flow field. These plots help to make clear the areas where separation, recirculation and large amounts of shear exist in the flow. In the 20% span slice, we see a negative vorticity (clockwise) wake extending down stream from the IGV trailing edge. We also see a region of strong positive vorticity (counter-clockwise) which starts at the rotor leading edge and extends along the rotor pressure side. Another strong region of
positive vorticity shows up at the stator leading edge, and the stator trailing edge shows a strong negative vorticity. At 50% span, the most noticeable difference is the absence of the leading edge vorticity at the rotor leading edge. Also apparent is the widening of the high vorticity region in the stator passage. In the 80% span slice, we see that the vorticity is confined to a much smaller region surrounding the rotor blade. We also see, as noted before, that the separation region in stator appears to be swept back, or compressed at this high span location.

E. Notch Filtered Unsteady Results

The time-mean flow field is useful for identifying flow features which may play an important role in the overall dynamics of the compressor. However, this time-averaged view into the flow field cannot give a clear perspective into the unsteadiness present in the compressor. Examining snapshots of the instantaneous unsteady flow field will surely be more illustrative of the dominant unsteady features in the flow. However this will be a poor means of ascertaining the extent of the asynchronous unsteadiness in the flow field. As observed in the pressure probe data, the magnitude of the asynchronous fluctuations in the flow field are much smaller than the magnitude of the shaft-linked fluctuations. Thus, when the unsteady flow field is examined directly, these fluctuations mask the underlying asynchronicity.

In order to examine the asynchronous fluctuations in the flow field more directly, a technique called notch filtering is used. We start with the collection of instantaneous flow fields from the final eight periods of oscillation. At every spatial point in the domain, for each time instant, we subtract the local time-mean flow field. With this fluctuation quantity, we take the Fourier Transform over the final eight periods to obtain the frequency response of the flow field. From the frequency response, we apply the notch filter by setting to zero those components of the response which fall outside of the frequency “notch” of interest. Now, performing the inverse Fourier transform, we obtain the notch filtered flow field. For the case of this analysis, we have filtered out all frequencies except for those within the range $[0.1, 0.4] \omega_p$, where $\omega_p$ is the rotor passing frequency.

Figure 12 shows 20% span slices of the notch filtered pressure field at fractional multiples of the passing period $T_p$. Notice that the polarity of the large fluctuation at the stator trailing edge seems to oscillate with a period of approximately $2T_p$. This large pressure fluctuation sends pressure waves upstream into the rotor passage. These can be seen clearly traveling upstream along the suction side of the rotor blade.

In Figure 12 we see 50% span slices at the same time intervals. Here we have even more violent fluctuations in the stator passage but these fluctuations do not seem to make it all the way upstream to the rotor. Perhaps this is due to the larger axial separation of the rotor from the stator at this span-wise location.

As mentioned earlier, by using only one rotor passage in the unsteady simulation, we effectively limit the spatial wavelength of allowable disturbances to be one passage width in the circumferential direction. To evaluate the effect of this on the resulting solution, a simulation was run using double the original domain. This case consisted of four IGV passages, two rotor passages and four stator passages. An example of an instantaneous slice through the flow domain is shown in Figure 14. In the figure, we clearly see that among the coherent spatial fluctuations, those whose wavelength spans the entire domain make up a significant portion of the total visible fluctuations. Also, for this simulation, the most dominant temporal disturbance is observed at one half of the rotor passing frequency as compared to the passing frequency itself. These signs indicate that the choice of the domain may have a significant impact on the resulting observed unsteadiness using the current method. Further computations with still larger domains are planned to continue examining this effect.

However, apart from these two noted effects, the remaining unsteadiness appears to be of a similar character in both the original domain and the double domain cases. For instance, the subharmonic components of the unsteady pressure fluctuations show the same behavior for all non-integer multiples of half the passing frequency. Also, the Fourier Transform of the modal blade force shows peaks at the same subharmonic frequencies as for the original domain case. These two facts lead us to believe that while there are some noticeable effects due to the choice of the flow domain, the method used here may still capture the asynchronous unsteadiness of the flow field.

F. Flow Field Summary Description

The diagram in Figure 15 is an attempt to highlight the major features of the off-design flow through the S15 compressor. In the IGV blade row we have seen that below mid-span, there exists a relatively severe
wake which extends downstream and may impinge on the rotor leading edge. Near the rotor leading edge, a separation region exists below 50% span which appears to fluctuate with at least some asynchronous component. In the off-design configuration, the flow is separated from the pressure side over much of the span of the stator passage. This separation region is widest near mid-span, and the resulting fluctuations in this region send coherent pressure waves back up stage into the rotor blade row, most noticeably near the hub and the tip.

IV. Conclusions

In this paper the flow through a multi-stage transonic turbomachine is computed with both steady (mixing-plane) and fully unsteady (sliding-mesh) approaches. The test case considered is the first one and a half stages of a compressor design by Siemens. Simulations were conducted as an attempt to identify and characterize potential asynchronous flow oscillations which may occur when operating at the Off-design condition, but not at the design condition.

The steady simulations indicate that the methodology used will predict the performance of the ma-
Figure 7. Pressure Probe Diagram and Fourier Transform of Unsteady Pressure Signal in the Rotor passage (Off-design Case)

By examining the time-mean flow field, major features of the off-design flow have been identified. Examination of the notch-filtered unsteady flow field helped give insight into the asynchronous component of the unsteadiness in the flow field. This pointed to the importance of pressure waves being sent upstream from the stator passage as a contribution to the asynchronous fluctuations in the flow field.
Figure 8. Pressure Probe Diagram and Fourier Transform of Unsteady Pressure Signal in the Stator passage (Off-design Case)
References


Figure 9. Mach Contours of the Time-Averaged Flow Field
Figure 10. Pressure Contours of the Time-Averaged Flow Field

(a) 20% Span

(b) 50% Span

(c) 80% Span
Figure 11. Radial Vorticity Contours of the Time-Averaged Flow Field
Figure 12. Time Series of Notch Filtered Pressure Fluctuation Contours displayed on 20% Span Slices
Figure 13. Time Series of Notch Filtered Pressure Fluctuation Contours displayed on 50% Span Slices
Figure 14. Snapshot of Notch Filtered Pressure Fluctuation Contours displayed on 20% Span Slice through the Flow Field for the Double Domain Case.
Figure 15. Summary Diagram for the Major Flow Features in the Off-design Configuration

Figure 16. Fourier Transform of Unsteady Pressure Signal in the Stator passage (Design Case)