AIAA 2001–3472
Blade-Forced Vibration Effects on Turbomachinery Rotor-Stator Interaction
Joseph E. Collard III and Paul G. A. Cizmas
Texas A&M University, College Station, TX 77843-3141

37th AIAA/ASME/SAE/ASEE Joint Propulsion Conference
July 8–11, 2001/Salt Lake City, UT
Blade-Forced Vibration Effects on Turbomachinery Rotor-Stator Interaction

Joseph E. Collard III* and Paul G. A. Cizmas†
Texas A&M University, College Station, TX 77843-3141

Understanding rotor-stator interaction is essential for the design of turbomachines with increased performance and improved reliability. The sources of unsteadiness currently modeled in the rotor-stator interaction studies include potential flow interaction, wake interaction, hot streak interaction, vortex shedding and shock/boundary layer interaction. The objective of this paper is to investigate the effects of blade-forced vibration on the rotor-stator interaction in turbomachinery. An existing parallelized solver of the Euler/Navier-Stokes equations for multi-stage compressors and turbines has been extended to model vibrating rotor blades. This numerical implementation can model rotor blades that vibrate with a plunging and/or pitching motion. The rotor blade vibration is superimposed on the blade rotation. The numerical algorithm has been used to simulate the flow in a two-stage turbine. Assuming that the first-stage rotor blades were plunging with an amplitude equal to 10% of the chord and the second-stage rotor blades were pitching with an amplitude of 5 deg, both rows vibrating with a frequency of 1230 Hz, the total-to-total efficiency decreased approximately 2 points compared to the rigid blade turbine. Consequently, blade-forced vibration can significantly affect the rotor-stator interaction in turbomachinery.

Nomenclature

\[ c \] Local speed of sound or chord of the airfoil
\[ C_p \] Coefficient of pressure,
\[ C_p = \left( p - p_\infty^* \right) / \left( \frac{1}{2} \rho \left( \omega r \right)^2 \right) \]
\[ f \] Frequency
\[ n_{cycle} \] Number of time steps per cycle
\[ p \] Pressure
\[ r \] Radius
\[ s \] Entropy
\[ t \] Time
\[ T \] Temperature or period
\[ u \] Component of velocity in \( x \)-direction
\[ U \] Freestream velocity
\[ y \] Plunging motion displacement
\[ y_0 \] Amplitude of plunging motion
\[ \alpha \] Pitching motion rotation angle
\[ \alpha_0 \] Amplitude of pitching motion
\[ \gamma \] Ratio of specific heats of a gas
\[ \eta \] Total-to-total efficiency
\[ \rho \] Density
\[ \mu \] Viscosity
\[ \tau_n \] Non-dimensional skin friction,
\[ \tau_n = \mu \frac{n_{pp}}{(\mu - \infty U_\infty / c)} \]
\[ \phi \] Flow coefficient or phase lag
\[ \omega \] Angular velocity
\[ \tilde{\omega} \] Reduced frequency, \( \tilde{\omega} = \omega_{pp}/U_\infty \)

Subscripts

\[ n \] Non-dimensional quantity
\[ pp \] Plunging and pitching motion quantity
\[ rot \] Rotational motion quantity
\[ 0 \] Reference quantity or maximum value
\[ 1 \] Plunging motion
\[ 2 \] Pitching motion
\[ -\infty \] Upstream infinity

Superscripts

\( \sim \) Non-dimensional quantity
\( * \) Total (or stagnation)

Introduction

Due to the complex nature of fluid flow in a turbomachine, early studies have focused on the analysis of steady flow.\(^{1-4}\) These studies used both Euler and Navier-Stokes equations to calculate the two- and three-dimensional models of steady flow through turbomachinery. Although these studies were able to predict some of the flow features using a time-marching method, they did not include the effects of unsteadiness inherent to turbomachinery flow.

More recently, investigations have included the effects of unsteadiness associated with rotor-stator interaction. Gibeling et al.\(^5\) presented a distorting-grid technique in which a single grid is wrapped around both airfoils and is distorted during computation so that it contains both the stator and the rotor at all times. This method works well for geometries with large gaps between airfoils. For closely spaced airfoils,
the grid becomes very distorted and may produce inaccurate solutions. To resolve this, a system of patched and overlaid grids, shown in Fig. 1, was developed by Rai to discretize the flow field surrounding the rotating and stationary airfoils. This method works for closely spaced airfoils since it allows the grids surrounding the airfoils to slip past each other.

Huff used a deforming grid technique to predict the oscillatory motion of the airfoils. Huff generated a time-marching, finite difference code to model the inviscid flow through oscillating cascades. Using this method, however, the quantitative agreement with experimental results is not always good.

An improvement of the solution accuracy and a reduction of the computational effort has been obtained by Hall, who used time-linearization, as opposed to time-marching, to solve the unsteady flow equations. A finite element method based on a variational principle was developed to model the small disturbance behavior of the full potential equation. In addition, the method used a deforming computational grid that conforms to the vibrating airfoils to predict flutter and forced response in turbomachinery.

Swafford et al. developed a code that is capable of simulating both the viscous flow, using the Navier-Stokes equations, and the inviscid flow, using the Euler equations. This code used block-structured H-grid to predict the flow.

To simulate the rotor-stator interaction effects, Giles developed a method that uses a set of quadrilateral elements to span the small axial gap between airfoils. As proven by comparison with experimental results, this method accurately simulates the aerodynamics of rotor-stator interaction for small gaps in turbomachinery.

The sources of unsteadiness in rotor-stator interaction include potential flow interaction, wake interaction, hot streak interaction, vortex shedding, shock/boundary layer interaction, and blade-forced vibration. All of the sources of unsteadiness except for blade-forced vibration were modeled in previous studies of rotor-stator interaction. The present paper investigates the effects of blade-forced vibration in the model of rotor-stator interaction.

**Numerical Model**

The numerical method is an extension of the scheme developed by Rai and Gundy-Burlet et al. that has been parallelized by Cizmas and Subramanya. The extension presented herein allows the rotor blades to vibrate with a plunging and pitching motion. The rotor blade vibration is superimposed on the blade rotation motion.

The quasi-three-dimensional, unsteady, compressible flow through a multi-stage turbomachine is modeled by using the Navier-Stokes/Euler equations. The computational domain for each airfoil is broken up into an inner and an outer region. The inner region, close to the airfoil where the viscous effects are strong, is simulated using the thin-layer Navier-Stokes equations. The Euler equations are solved in the regions away from the airfoil, where the viscous effects are weak.

The integration method of the Navier-Stokes/Euler equations is a third-order-accurate, upwind-based Osher scheme. The non-linear finite-difference approximation is solved iteratively at each time level using an approximate factorization method. Two to four Newton iterations are used at each time to step in order to reduce the factorization and linearization errors.

The flow is assumed to be fully turbulent. The kinematic viscosity is computed using Sutherland’s law. The Baldwin-Lomax turbulence model is used to simulate the turbulent eddy viscosity. The Baldwin-Lomax turbulence model is written in a stationary reference frame for the stators and in a rotating reference frame for the rotors.

**Grid Generation**

A combination of patched and overlaid grids is used to discretize the flow field, as shown in Fig. 1. Body-fitted O-grids, generated using an elliptic grid generator, are used to capture the viscous effects near the airfoil. The O-grid is used to resolve the Navier-Stokes equations. The H-grid, which discretizes the computational domain away from the airfoil, is generated using an algebraic grid generator. The H-grid is used to resolve the Euler equations which model the flow in the outer region. The O-grids are overlaid on the H-grids, which slip past each other to simulate the rotor-stator interaction.

**Boundary Conditions**

Since multiple grids are used to discretize the Navier-Stokes and Euler equations, two classes of boundary conditions must be enforced on the grid boundaries: natural boundary conditions and zonal boundary conditions. The natural boundaries include inlet, outlet, periodic and the airfoil surfaces. The zonal boundaries include the patched and overlaid boundaries.

The inlet boundary conditions include the specification of flow angle, total pressure and either down-
stream propagating Riemann invariant or the total temperature. The upstream propagating Riemann invariant is extrapolated from the interior of the domain. At the outlet, the average static pressure is specified, while the downstream propagating Riemann invariant, circumferential velocity, and entropy are extrapolated from the interior of the domain. Periodicity is enforced by matching flow conditions between the lower surface of the lowest H-grid of a row and the upper surface of the topmost H-grid of the same row. At the airfoil surface, no-slip, adiabatic wall and zero pressure gradient condition are enforced.

For the zonal boundary conditions of the overlaid boundaries, data are transferred from the H-grid to the O-grid along the O-grid’s outermost grid line. Data are then transferred back to the H-grid along its inner boundary. At the end of each iteration, an explicit, corrective, interpolation procedure is performed. The patch boundaries are treated similarly, using linear interpolation to update data between adjoining grids.

Blade Vibration

In order to model blade-forced vibration, the grids must follow the pitching and/or plunging motion of the airfoil. Both the inner and outer boundaries of the elliptically generated O-grid follow the prescribed airfoil motion. The O-grid lines connecting the inner and outer boundaries are allowed to deform. For the algebraically generated H-grid, the inner boundary and outer boundary are fixed. The grids are then generated for each plunging and pitching location of the airfoil. Once the grids are generated for each time step of pitching and/or plunging vibration, the flow solver loads them as it marches in time.

There are two time scales that govern the unsteady flow: the period of rotation of the shaft and the period of pitching and/or plunging vibration.

Parallel Computation

The current work is an extension of the PARSI2 (Parallel Rotor-Stator Interaction) code development. The parallel code uses message-passing interface (MPI) libraries and runs on symmetric multiprocessors (Silicon Graphics Challenge) and massively parallel processors (Cray T3E). One processor is allocated for each airfoil in the two-dimensional simulation. The processors allocation is presented in Fig. 2. One processor is allocated for each inlet and outlet H-grid. One processor is allocated for the O- and H-grids corresponding to each airfoil. Interprocessor communication is used to match boundary conditions between grids. Periodic boundary conditions are imposed by cyclic communication patterns within rows. Slip boundary conditions are imposed by gather-send-receive-broadcast communication routines between adjacent rows. Load imbalance issues need to be considered at grid generation time to reduce synchronization overhead.

Results

This section presents the numerical prediction of the rotor-stator interaction in a turbine that has either rigid or vibrating blades. The rotor-stator interaction is computed in a two-stage steam turbine. First, the geometry and flow conditions are presented. Next, the blade motion for blade-forced vibration simulation is discussed. Then the accuracy of the numerical results and grid independence are outlined. Lastly, the results for pressure, velocity, and entropy are presented for both the blade-forced vibration and rigid blade cases.

Geometry and Flow Conditions

The two-stage steam turbine used in this study has 68 first-stage stators, 55 first-stage rotors, 70 second-stage stators, and 60 second-stage rotors. A dimensionally accurate simulation of this geometry would require modeling the flow in all 253 (68+55+70+60) inter-blade passages. To reduce the computational effort, it was assumed that there were an equal number of blades (68) in each turbine row. To maintain the same flow area, the airfoils were re-scaled by the factors shown in Table 1.

Table 1  Airfoil rescaling factors.

<table>
<thead>
<tr>
<th>Airfoil</th>
<th>Rescaling Factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>first-stage stator</td>
<td>1</td>
</tr>
<tr>
<td>first-stage rotor</td>
<td>55/68</td>
</tr>
<tr>
<td>second-stage stator</td>
<td>70/68</td>
</tr>
<tr>
<td>second-stage rotor</td>
<td>60/68</td>
</tr>
</tbody>
</table>

The inlet temperature in the turbine is 1469.9 degrees Rankine, the inlet Mach number is 0.079 and the pressure is 3.285 MPa. The ratio of the exit static pressure and inlet stagnation pressure is 0.7469. The ratio of specific heats at the given temperature and pressure is $\gamma = 1.273$. The inlet flow angle is 13.4 degrees and the inlet Reynolds number is 322,630 per inch, based on the axial chord of the first-stage stator. The Prandtl number is 0.922 and the turbulent Prandtl number is 1.153.

Results are presented at a mid-span radius of 18.115 inches for a rotational speed of 3,600 RPM. The value of the flow coefficient, $\phi$, is 0.2998. The results presented herein have been computed using two Newton
sub-iterations per time-step and at least \( n_{cycle} = 3,000 \) time-steps per cycle. Here, a cycle is defined as the time required for a rotor to travel a distance equal to the pitch length at midspan. Each simulation was run in excess of 50 cycles to ensure time-periodicity.

The computations were performed on the Silicon Graphics Origin 2000 computer at Texas A&M University Supercomputing Facility. The computation can, however, be done on any parallel computer, including an inexpensive Beowulf type PC cluster. Six processors were used for this analysis, as shown in Fig. 2. For the case with rigid blades the computation time was \( 1.4 \times 10^{-5} \) secs/grid point/iteration on a Silicon Graphics Origin2000 computer. The computation time for the simulation with blade-forced vibration was \( 1.5 \times 10^{-5} \) secs/grid point/iteration. The computation of the blade-forced vibration takes an extra \( 1 \times 10^{-6} \) secs/grid point/iteration due to the additional input/output access time necessary to update the position of the vibrating rotor blade.

**Blade-Forced Vibration Case**

The blade-forced vibration phenomenon is simulated by imposing a motion of the blade, which consists of harmonic body translation in the y-direction (plunging) and rotation (pitching). In the example presented here, the first rotor both rotates and vibrates with a pitching motion and the second rotor both rotates and vibrates with a plunging motion. The code is able to simulate both pitching and plunging of one blade, but for this analysis each blade only pitches or plunges. The harmonic plunging motion is given by

\[
y(t) = y_0 \sin \omega_1 t
\]

and the pitching motion is defined as

\[
\alpha(t) = \alpha_0 \sin (\omega_2 t + \phi)
\]

where \( \omega_1 \) and \( \omega_2 \) are the angular velocities for plunging and pitching, respectively, \( y_0 \) and \( \alpha_0 \) are the amplitudes of plunging and pitching motion, respectively, and \( \phi \) is the phase lag between plunging and pitching. In this analysis it is assumed that \( y_0 = 10\% \) chord, \( \alpha_0 = 5\text{ deg} \), \( \phi = 0 \) and \( \omega_1 = \omega_2 \). One considers a blade vibration case with a reduced frequency, \( \bar{\omega} = 5.6 \), where \( \bar{\omega} = \omega_{pp}/U_{\infty} \). The chord \( c \) is 0.0312 m, given by the first-stage stator airfoil, and the freestream velocity, \( U_{\infty} \), is 43.1 m/s. Using the values defined above, the angular velocity, \( \omega_{pp} \), is 7,733 rad/s. Consequently, the period of the pitching and plunging motion is \( T_{pp} = 2\pi/\omega_{pp} = 8.13 \times 10^{-4} \) s and the frequency for pitching and plunging is \( f_{pp} = 1/T_{pp} = 1230.73 \) Hz.

The rotation speed of the turbine is 3,600 RPM. For 68 blades per row, the time is taken to rotate the distance between two consecutive stator blades is \( T_{rot} = 2.45 \times 10^{-4} \) s. The frequency a rotor blade excites a stator blade (and vice-versa) is \( f_{rot} = 4080 \) Hz. Therefore, if it takes \( n_{cycle} = 3,000 \) time-steps per cycle for rotational motion, then each time step is given by

\[
\Delta t_{rot} = \frac{T_{rot}}{n_{cycle}} = 8.16 \times 10^{-8} \text{s.}
\]

The period of pitching and plunging motion is split into 50 equal intervals. For the pitching and plunging motion, the time-step is given by

\[
\Delta t_{pp} = \frac{T_{pp}}{50} = 1.626 \times 10^{-5} \text{s.}
\]

The number of pitching and plunging time-steps per rotational time-step is given by \( \Delta t_{pp}/\Delta t_{rot} = 200 \). Comparing the period due to rotation, \( T_{rot} \), and the period due to pitching and plunging, \( T_{pp} \), it is found that \( T_{pp}/T_{rot} = 3.3 \). As a result, it takes 3.3 rotational cycles to complete one pitching/plunging period.

**Accuracy of Numerical Results**

To prove the accuracy of numerical results, one has to show that the solution is periodic and the results are grid independent. Three grids were used to assess the grid independence of the solution. The number of grid points of the coarse, medium, and fine grids are presented in Table 2.

**Table 2 Grid points of the coarse, medium and fine meshes at mid-radius.**

<table>
<thead>
<tr>
<th>Grid Points</th>
<th>Coarse</th>
<th>Medium</th>
<th>Fine</th>
</tr>
</thead>
<tbody>
<tr>
<td>H-grid inlet</td>
<td>75x60</td>
<td>100x60</td>
<td>120x96</td>
</tr>
<tr>
<td>H-grid airfoil</td>
<td>67x60</td>
<td>90x60</td>
<td>108x96</td>
</tr>
<tr>
<td>O-grid airfoil</td>
<td>112x37</td>
<td>150x50</td>
<td>180x60</td>
</tr>
<tr>
<td>H-grid outlet</td>
<td>75x60</td>
<td>100x60</td>
<td>120x96</td>
</tr>
<tr>
<td>Total grid points</td>
<td>41,656</td>
<td>63,600</td>
<td>107,712</td>
</tr>
</tbody>
</table>

To validate the grid independence, three values of the pressure coefficient and skin friction have been compared: the averaged, minimum and maximum over one period. The conclusion was that the medium grid is the best compromise between accuracy and computational cost.\(^{15}\) To reduce the computational cost, herein only the coarse grid results are presented.

The flow periodicity is monitored on the last row of the turbine since it is expected that the flow on this row is the last to become periodic. The periodicity of the unsteady flow corresponding to the blade-forced vibration and to the rigid blade motion are investigated. The periodicity of the unsteady solution is verified by comparing the pressure coefficient and skin friction on the second-stage rotor at three consecutive cycles. The periodicity of the unsteady solution is also verified by investigating the variation of the efficiency as a function of the number of cycles.

First, the periodicity of the unsteady solution is verified by comparing the pressure coefficient, \( C_p \), results

4 OF 10

**AMERICAN INSTITUTE OF AERONAUTICS AND ASTRONAUTICS PAPER 2001-3472**
of three consecutive cycles for the blade-forced vibration case, as shown in Figs. 3 to 5. Since the values of the pressure coefficient for the blade-forced vibration case are nearly identical over three consecutive cycles, the solution is considered periodic.

The periodicity of the unsteady solution for the rigid blade case is verified by comparing the pressure coefficient, $C_p$, for three consecutive cycles, as shown in Figs. 6 to 8. The values of the pressure coefficient for the rigid blade case are nearly identical over three consecutive cycles. Consequently, one can conclude that the solution is periodic. The comparison of the maximum, averaged and minimum pressure coefficient for the blade-forced vibration case and the rigid blade case is presented in Figs. 9 to 11. The minimum and maximum values of the pressure coefficient for the blade-forced vibration case and the rigid blade case differ greatly. The averaged values of the pressure coefficient for the blade-force vibration and rigid blade are quite similar. The averaged pressure on the rigid blade, however, is higher than on the vibrating blade. The lift of the rigid blade and the vibrating blade are approximately the same.

The periodicity of the unsteady solution for the blade-forced vibration case is also verified by comparing the non-dimensional skin friction, $\tau_n$, results of three consecutive cycles, as shown in Figs. 13 to 15. Since the values of the non-dimensional skin friction for the blade-forced vibration case are nearly identical over three consecutive cycles, one concludes that the solution is periodic.

The periodicity of the unsteady solution for the rigid blade case is verified by comparing the non-dimensional skin friction, $\tau_n$, for three consecutive cycles, as shown in Figs. 16 to 18. As expected, the solution is periodic for this case also, since the values of the non-dimensional skin friction are nearly identical over three consecutive cycles.
Fig. 7  Averaged pressure coefficient for the rigid blade case at three consecutive cycles.

Fig. 8  Minimum pressure coefficient for the rigid blade case at three consecutive cycles.

Fig. 9  Maximum pressure coefficient for blade-forced vibration versus the rigid blade case.

Fig. 10  Averaged pressure coefficient for blade-forced vibration versus the rigid blade case.

Fig. 11  Minimum pressure coefficient for blade-forced vibration versus the rigid blade case.

Fig. 12  Efficiency convergence for the case with pitching and plunging and the rigid blade case.
Fig. 13  Maximum non-dimensional skin friction for blade-forced vibration at three consecutive cycles.

Fig. 14  Averaged non-dimensional skin friction for blade-forced vibration at three consecutive cycles.

Fig. 15  Minimum non-dimensional skin friction for blade-forced vibration at three consecutive cycles.

Fig. 16  Maximum non-dimensional skin friction for the rigid blade case at three consecutive cycles.

Fig. 17  Averaged non-dimensional skin friction for the rigid blade case at three consecutive cycles.

Fig. 18  Minimum non-dimensional skin friction for the rigid blade case at three consecutive cycles.
Fig. 19 Maximum non-dimensional skin friction for blade-forced vibration versus the rigid blade case.

Fig. 20 Averaged non-dimensional skin friction for blade-forced vibration versus the rigid blade case.

Fig. 21 Minimum non-dimensional skin friction for blade-forced vibration versus the rigid blade case.

Fig. 22 Time-average non-dimensional pressure for blade-forced vibration.

Figures 19 to 21 show a comparison of the maximum, averaged, and minimum skin friction for the blade-forced vibration case and the rigid blade case. The values of the skin friction differ significantly between the vibrating and rigid blade cases. Figure 20 shows that for the rigid blade case, the averaged value of the skin friction is positive for approximately 90% of the airfoil surface. A separation region exists near the leading edge. For the blade-forced vibration case, the averaged value of the skin friction is much smaller than for the rigid blade case. Since the averaged value of the skin friction is very close to zero, during part of the cycle the flow is separated. For this reason the efficiency of the turbine in the blade-forced vibration case is smaller than in the rigid blade case, as shown in Fig. 12.

Unsteady solution periodicity can also be proven by showing that the total-to-total efficiency averaged over consecutive cycles converges, as shown in Fig. 12 for blade-forced vibration and the rigid blade case. Here the total-to-total efficiency is defined as:

\[
\eta = \left( 1 - \frac{T_{exit,ca,ta}}{T_{inlet,ca,ta}} \right) / \left[ 1 - \left( \frac{p_{exit,ca,ta}}{p_{inlet,ca,ta}} \right)^{\frac{\gamma - 1}{\gamma}} \right]
\]

where subscript “ca” denotes circumferential-averaged, “ta” denotes time-averaged.

The flow is considered to be periodic if the variation of the averaged total-to-total efficiency from one cycle to the next is less than 0.05 points. Figure 12 also shows that the efficiency for blade-forced vibration is approximately 94.7% and the efficiency for non-blade forced vibration case is approximately 96.5%. The 1.8 point efficiency difference between the two cases is significant and shows the negative impact of blade-force vibration on turbine efficiency.

Variation of Flowfield Variables

This section presents the variation of the following time-averaged flow field variables: pressure, velocity and entropy. These flow variables are shown at mid-span for both the rigid and vibrating blades.

The time-averaged non-dimensional pressure variation is shown in Figs. 22 and 23. The non-dimensional
Fig. 23 Time-average non-dimensional pressure for the rigid blade case.

Fig. 24 Time-average non-dimensional velocity for blade-forced vibration.

Fig. 25 Time-average non-dimensional velocity for the rigid blade case.

Fig. 26 Time-average non-dimensional entropy for blade-forced vibration.

Fig. 27 Time-average non-dimensional entropy for the case without pitching and plunging.

The non-dimensional pressure, $\tilde{p}$, is defined as $\tilde{p} = \frac{p}{p_{-\infty}}$, where $p_{-\infty}$ is the pressure at upstream infinity. The differences between the pressure variation corresponding to the blade-forced vibration case and the rigid blade case is relatively small.

Time-averaged non-dimensional velocity variation is shown in Figs. 24 and 25. The non-dimensional velocity, $\tilde{u}$, is defined as $\tilde{u} = \frac{u}{\sqrt{\rho_{-\infty}/p_{-\infty}}}$, where $\rho_{-\infty}$ is the density at upstream infinity. The velocity variation between the blade-forced vibration and the case without pitching and plunging is relatively small. For both cases, the maximum velocity occurs at the leading edge of the second stage rotor.

Time-averaged non-dimensional entropy variation is shown in Figs. 26 and 27. The non-dimensional entropy, $\tilde{s}$, is defined as $\tilde{s} = \frac{s}{s_{-\infty}}$ where $s_{-\infty}$ is the entropy variation at upstream infinity. The entropy variation between the blade-forced vibration and the case without pitching and plunging is the largest of the non-dimensional flow variables presented in this paper. The entropy is higher for the blade force vibration case, indicating that losses are higher for this case. This result is in agreement with the efficiency values shown in Fig. 12.

**Conclusions**

An existing parallelized solver of the Euler/Navier-Stokes equations for multi-stage compressors and turbines has been extended to model vibrating rotor blades. This numerical implementation, which can model rotor blades that vibrate with a plunging and/or pitching motion, has been used to evaluate the effects of blade-forced vibration on rotor-stator interaction. To the best knowledge of the authors, this is the first time when the effects of blade-forced vibration on rotor-stator interaction are presented in the literature.

The numerical algorithm has been used to simulate the flow in a two-stage turbine. For the type of vibration assumed for the rotor blades, the numerical prediction showed a reduction of the total-to-total efficiency of approximately 2 points, compared to the turbine with rigid blades. The efficiency deterioration
was mainly due to the flow separation on the second stage rotor blades. Consequently, blade-forced vibration can significantly affect the rotor-stator interaction and, as a result, the overall efficiency in turbomachinery.

Acknowledgments
The authors wish to thank Texas Engineering Experiment Station for supporting part of this work. The authors are thankful to Texas A&M University Supercomputing Facility for making the computing resources available.

References


