Rotordynamic Effects of Compressor Rotor-Tip Clearance Asymmetry

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Rotordynamic effects caused by nonaxisymmetric rotor-tip clearance in an axial compressor stage have been analyzed. Two coupled actuator disc analyses are carried out at blade (meridional plane) and radius (radial plane) scales. Continuity, axial, and tangential momentum equations are used to connect upstream and downstream flows across the compressor stage, and a small perturbation analysis is used to obtain flowfield nonuniformities. Predictions indicate the following. Relative to the passage flow, the flow associated with the rotor-tip clearance is axially retarded and tangentially underturned. Furthermore, the compressor blades near the smaller tip gap are more highly loaded. Thus, the tangential force asymmetry promotes a backward whirl. However, the pressure asymmetry caused by azimuthal flow redistribution induces a forward whirl. The net result is a backward-whirling force whose magnitude is much smaller than that of the forward-whirling force found in turbines.

Nomenclature

\( B \) = Bernoulli constant
\( C \) = absolute flow velocity
\( c \) = axial blade chord
\( c_l \) = lift coefficient
\( e \) = magnitude of rotor offset
\( F_x \) = lateral force in the direction of the offset
\( F_y \) = lateral force perpendicular to the direction of the offset
\( f_y \) = tangential force per azimuthal length
\( H \) = annulus height
\( H_b \) = rotor blade span
\( h \) = enthalpy
\( L \) = rotor thickness
\( \dot{m} \) = mass flux
\( P \) = pressure
\( Q \) = strength of the shear layer
\( q \) = nondimensional vorticity strength; local mass flux
\( R \) = mean compressor radius; turbine reaction
\( s \) = blade pitch
\( t \) = radial tip clearance
\( U \) = turbine rotational speed at the mean radius, \( \omega R \)
\( W \) = relative velocity
\( X \) = direction along the rotor offset
\( x \) = axial direction
\( Y \) = direction perpendicular to rotor offset
\( y \) = tangential direction
\( z \) = radial direction
\( \alpha \) = absolute flow angle, eigenvalue
\( \alpha_x \) = direct excitation force coefficient
\( \alpha_y \) = cross excitation force coefficient
\( \beta_y \) = relative flow angle; angle between tip jet flow and passage flow
\( \Delta \) = thickness of underturned layer downstream of actuator disc
\( \Phi \) = turbine flow coefficient, \( C_t/U \)

\( \theta \) = azimuthal angle measured in the direction of rotation from the minimum gap location; angle between the underturned stream and the bladed stream
\( \lambda \) = nondimensional mass fraction of underturned flow
\( \rho \) = density
\( \Psi \) = meridional stream function; work coefficient
\( \omega \) = angular velocity of rotor shaft rotation; vorticity

\( D \) = design value
\( m \) = mean
\( t \) = stagnation condition
\( 0 \) = far upstream of actuator disc in the blade scale analysis
\( 0- \) = near upstream of the actuator disc in the blade scale analysis
\( 0+ \) = near downstream of the actuator disc in the radius scale analysis
\( 1 \) = rotor inlet in the blade scale analysis
\( 2 \) = stator outlet in the blade scale analysis
\( 3 \) = far downstream of stator in the blade scale analysis
\( \perp \) = meridional component

Superscripts

\( ^{-} \) = the part of flow downstream that has crossed the bladed part of turbine flow
\( ^{+} \) = the part of flow downstream that was underturned due to the rotor tip gap
\( ^{'} \) = nonaxisymmetric perturbation
\( ^{\#} \) = azimuthal mean, or axisymmetric value
\( ^{\circ} \) = complex amplitude

I. Introduction

The effects of nonaxisymmetric tip clearance on the performance and rotordynamics of axial turbomachinery have been investigated by various authors. In turbines the rotordynamic forces were first suggested independently by Thomas\(^1\) and Alford.\(^2\) They both assumed that the blades in the smaller gap region are more...
highly loaded. The net effect is a cross-coupled force acting perpendicularly to the rotor offset in the direction of rotation, causing a forward whirl. Experimentally, Ulrichs and Martinez-Sanchez et al. obtained rotordynamic force and aerodynamic data for unshrouded turbines. Analytically, Qiu formulated an actuator disk turbine model based largely on the compressor model of Horlock and Greitzer. However, the assumption of perfect flow guidance in the tip clearance region led to poor predictions. Song and Martinez-Sanchez presented an actuator disk analysis capable of accurately predicting forces measured in turbines.

On the other hand, research on the effects of nonaxisymmetric rotor-tip clearance in axial compressors has been relatively scarce. Experimentally, Vance and Laudadio measured forces in an axial fan. Analytically, Alford assumed that, in compressors, blades near the smaller gap are more lightly loaded, and, thus, predicted a backward whirl inducing force. Horlock and Greitzer examined the effect of clearance asymmetry on the velocity field. Colding-Jorgensen adapted Qiu’s analysis for a compressor and predicted forward whirling forces in the “normal operating range.” Ehrich used the parallel compressor model and data from the General Electric Low Speed Research Compressor to predict mostly backward whirling forces. Recently, Graf et al. developed a model to predict the effects of clearance asymmetry on compressor stability.

Despite such efforts, flowfields and rotordynamic forces induced by rotor-tip clearance asymmetry in an axial compressor are not well understood. If one assumes, as Alford did, that the blades near the smaller tip gap are more lightly loaded, and, thus, predicted a forward whirl inducing force. However, if one assumes, as Ehrich did, that the blades near the smaller tip gap are more highly loaded, and, thus, predicted a backward whirl inducing force. Furthermore, the effects of nonaxisymmetric flow redistribution have not yet been addressed by anyone. Thus, neither direction nor magnitude of such forces can be predicted with confidence (Fig. 1).

Therefore, the objective of this investigation is to gain physical understanding of flowfields and rotordynamic effects induced by static, nonaxisymmetric rotor-tip clearance in a single-stage axial compressor. An analytical compressor model based on actuator disk theory has been newly formulated. From the model’s flowfield predictions the rotordynamic forces can be inferred.

II. Analytical Model

The approach taken in this investigation is conceptually similar to that adopted in Song and Martinez-Sanchez. The blade scale analysis solves for the radial flow redistribution caused by axisymmetric rotor-tip clearance. The radius scale analysis examines the azimuthal flow redistribution caused by nonaxisymmetric rotor-tip clearance.

The actuator disk consists of a compressor stage with a full-span inlet guide vane (IGV) row, a partial-span rotor row, and a full-span stator row, collapsed into a single plane at $x = 0$. The various axial locations at the blade scale are as follows. Upstream of the IGV is referred to as station 0. Inlet to the rotor is referred to as station 1, and the rotor exit is called station 2. Downstream of the stator row is called station 3. Far downstream is referred to as station 4. On the radius scale $x = 0^\circ$ is equivalent to the blade scale station 0, and $x = 0^\circ$ is equivalent to the blade scale station 4. Figure 2 shows the blade scale and the radius scale views of the actuator disk. Figure 3 shows compressor blade geometry with the velocity triangles from Cohen et al.

The assumptions used in the analysis are listed here: 1) incompressible, inviscid flow; 2) compressor stage collapsed in the axial direction to $x = 0$; 3) the blade geometry is radially uniform, except for the rotor-tip clearance; and 4) in the bladed region away from tip clearance, the blades guide the flow perfectly, or the relative flow angles are same as the blade angles. Because of assumptions 1 and 4, the effects such as deviation and blockage are not accounted for in this model.

The current approach is different from the preceding nonaxisymmetric compressor models in the following ways. First, empirical inputs are not needed. Approaches of Horlock and Greitzer and Colding-Jorgensen both require empirical performance (e.g.,

![Fig. 1 Lateral force on an offset compressor rotor.](image1)

![Fig. 2 Blade and radius scale views of a compressor stage.](image2)

![Fig. 3 Compressor velocity triangles.](image3)
pressure rise coefficient, efficiency) variation with clearance. Ehrlch's method requires detailed aerodynamic data, and Graf's approach requires compressor characteristics. However, the models of Horlock and Greitzer and Graf consider more complex physics such as deviation in the passage and compressor stability, which are not modeled in the current model. Second, the current model explicitly accentuates the effects of rotor-tip clearance by transforming to a streamline coordinate from the radial $z$ coordinate. Song and Martinez-Sanchez showed that it was important to explicitly model the flow associated with the tip region. Qiu uses experimentally determined dependence of global parameters (e.g., pressure rise coefficient or efficiency) on tip clearance as inputs in his model. Such smearing of the tip clearance leads to poor predictions. Colding-Jorgensen's is similar to that of Qiu.

A. Blade Scale Analysis

The compressor blade scale analysis presented here is based on the work of Roh. The additional assumptions for the blade scale analysis are as follows:

1) The flow is axisymmetric $\partial/\partial y = 0$.
2) Flow conditions are radially uniform at the rotor inlet (station 1).

Because there are no variations with $y$, the azimuthal momentum equation for the flow is

$$C_\perp \cdot \nabla C_y = 0$$

(1)

where $C_\perp$ is the meridional velocity defined as

$$C_\perp = iC_x + kC_y$$

In this case the vorticity equation can be written as

$$C_\perp \cdot \nabla \omega_y = 0$$

(2)

and the Bernoulli equation becomes

$$C_\perp \cdot \nabla B_{x,1} = 0$$

(3)

The stream function $\Psi$ can be introduced for the meridional flow to obtain

$$\omega_y(\Psi) = \nabla^2 \Psi \left( \nabla^2 \Psi = \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial z^2} \right)$$

(4)

Upstream of the stage ($x < 0$) the flow is assumed to be irrotational ($\omega_y = 0$). Therefore, $\Psi$ obeys Laplace's equation. Downstream ($x > 0$), according to Gauthier, the axial velocity at the rotor exit can be assumed to be piecewise constant, with a discontinuity at the blade tip. Therefore, the $\omega_y$ vorticity is concentrated at their interface. Quantities in the undisturbed layer are denoted with a $(+)$ superscript, and those in the rotor blade region by a $(-)$ superscript.

Because $\omega_y = d B_{x,1} / d \Psi$

(5)

vorticity $\omega_y$ is created and then convected along each streamline. From the definition of $B_{x,1}$ with the continuity constraint and the assumption of spanwise uniform blade dis,

$$B_{x,1} \cdot B_{x,1} = (P_3 - P_1)/\rho$$

(6)

Upstream of the stage, the irrotationality means $d B_{x,1} / d \Psi = 0$. Therefore, Eq. (5) can be rewritten as

$$\omega_y = \frac{d B_{x,1}}{d \Psi} = \frac{d (B_{x,1} - B_{x,1})}{d \Psi} = \frac{d}{d \Psi} \left( \frac{P_3 - P_1}{\rho} \right)$$

(7)

Thus, $\omega_y$ can be determined from the static enthalpy $(P_3 - P_1)/\rho$ addition by the compressor.

The static pressure rise is equivalent to the stagnation enthalpy increase, given by Euler's equation, minus the kinetic energy gain.

$$\Delta P_{13}/\rho = \Delta h_e - \Delta K.E.$$

(8)

where $\Delta P_{13} = P_3 - P_1$, $\Delta h_e = U(C_{x,2} - C_{x,1})$ and $\Delta K.E. = \frac{1}{2} \left( C_{x,2}^2 - C_{x,1}^2 \right)$. The assumptions of perfect guidance by the blades and radial uniformity lead to

$$C_{x,1} = C_{x,1} \tan(\alpha_1)$$

(9)

At the rotor exit (station 2) the flow has split into two streams. For the bladed stream

$$C_{x,2}^- = U - C_{x,2}^- \tan(\beta_2)$$

(10)

Therefore,

$$[(P_3 - P_1)/\rho]^+ = U \left( C_{x,2}^- - C_{x,1}^- \right) - \frac{1}{2} \left[ C_{x,2}^2 + C_{x,1}^2 - (C_{x,1}^2 + C_{x,2}^1) \right]$$

(11)

The tip clearance flow model (Fig. 4) used in this analysis is described in detail in Roh. Basically, the tip leakage flow and an equal amount of passage flow form a tip vortex. Trajectory, strength, and mass fraction of the vortex can be determined as a function of the compressor blade geometry and tip clearance. Thus, for the underturned stream the azimuthal velocity is given as

$$C_{x,2}^+ = U - C_{x,2}^+ \frac{\cos(\theta)}{\sin(\beta_m + \theta)}$$

(12)

where $\beta_m$ is the mean flow angle through the rotor and $\theta$ is the angle relative to $\beta_m$ at which the undisturbed stream is assumed to roll up into vortices (Fig. 4). $\beta_m$ is given by

$$\tan(\beta_m) = \frac{1}{2} \left[ \tan(\beta_1) + \tan(\beta_2) \right]$$

(13)

and $\theta$ is a function of the lift coefficient per unit blade span $c_s$, which can determined as follows:

$$c_s = 2.0 (s/c) \cos(\beta_m) \left[ \tan(\beta_1) - \tan(\beta_2) \right]$$

(14)

where $s/c$ is the reciprocal of solidity. Thus,

$$[(P_3 - P_1)/\rho]^+ = U \left( C_{x,2}^+ - C_{x,1}^+ \right) - \frac{1}{2} \left[ C_{x,2}^2 + C_{x,1}^2 \right]$$

(15)

In Eqs. (14) and (15) the axial velocity $C_{x,1}$ at the disk is a function of $z$. Thus, $d/d\Psi$ in Eq. (7) can be expressed as

$$\frac{d}{d \Psi} = \frac{\partial}{\partial C_x} \frac{\partial C_x}{\partial z} \frac{\partial z}{\partial \Psi} = \left( 1 - \frac{C_x}{C_0} \frac{\partial C_x}{\partial z} \right) \frac{\partial C_x}{\partial \Psi}$$

(16)

Then, from Eqs. (7), (11), (14), and (15), the equation for $\Psi$ becomes the following.
The boundary conditions are

\[ \nabla^2 \Psi = 0 \]  
\[ \nabla^2 \Psi = Q \delta(\Psi - \Psi_{\eta_0}) \]

where

\[ Q = \int_{-\infty}^{+\infty} \alpha \, d\Psi = B_2^{+3} - B_2^{-3} \]

\( \delta \) is Dirac's delta function, and \( Q \) is the strength of the \( y \) component of the vorticity between the underturned and bladed streams. From Eqs. (6), (11), and (14)

\[ Q = U \left( C_{12}^{+} - C_{12}^{-} \right) - U \left( C_{12}^{+} - C_{12}^{-} \right) \]

\[ - \frac{1}{2} \left[ C_{31}^{+2} + \left[ C_{31}^{+} \left( \frac{\sin(\theta)}{\cos(\beta_m)} \right) \right]^2 - C_{31}^{-2} \right] \]  

The boundary conditions are

\[ \Psi(x, 0) = 0, \quad \Psi(x, H) = C_{10} H \]

\[ \Psi(x, 0, z) = C_{10} \left[ 1 + \left( \frac{q}{2} \right)(1 - \lambda) \right] \]

\[ \Psi_{1}(z) = \Psi_{3}(z), \]

\[ \frac{\partial \Psi_{1}(z)}{\partial z} = \frac{\partial \Psi_{3}(z)}{\partial z} \]

The axial velocities for each stream can be expressed as

\[ C_{12}^{+} = C_{12} \left[ 1 + \left( \frac{q}{2} \right)(1 - \lambda) \right] \]

\[ C_{12}^{-} = C_{12} \left[ 1 - \lambda \left( \frac{q}{2} \right) \right] \]

where \( q = Q / C_{12}^2 \). Substituting Eqs. (21) and (22) into the definition of \( Q \) yields a quadratic equation for \( Q \) as a function of \( \lambda \):

\[ G(1 - 2\lambda) - \left[ \frac{\sin(\theta)}{\cos(\beta_m)} \right]^2 \left( \frac{q}{2} \right)^2 \]

\[ + 2 \left[ \frac{q}{2} \right] \left( 2 + \frac{1}{\Phi} \right)(1 - \lambda)T + \frac{2}{\Phi} \tan(\beta_2) + G - \lambda \left( \frac{\sin(\theta)}{\cos(\beta_m)} \right)^2 \right]^2 \]

\[ \times \left( \frac{q}{2} \right) + \left[ \frac{2}{T} - \frac{2}{T} \tan(\beta_2) + \left( \frac{\sin(\theta)}{\cos(\beta_m)} \right)^2 \right]^2 = 0 \]

where \( G = \tan^2(\alpha_s) + \left( \frac{\sin(\theta)}{\cos(\beta_m)} \right)^2 \), \( T = \cos(\theta) \sin(\beta_m) \). The amount of flow leaked through the tip gap is \( \lambda \), and \( \alpha_s \) is a function of \( T \) and \( \lambda \):

\[ t/H = (\lambda/2)(1 - (1 - \lambda/2)q/2) \]

Thus, once the values of \( \lambda \) and \( q \) that satisfy the system of Eqs. (23) and (24) have been found for a given \( t/H \), the velocities can be evaluated.

B. Radius Scale Analysis

To model a compressor offset at some amplitude \( \hat{\delta} \) much smaller than the rotor blade span \( H_0 \), a linear perturbation approximation is used. For small tip gaps \( \hat{t} / R \ll 1 \) with an offset rotor, the azimuthal distribution of the rotor-tip gap is given as

\[ t = \hat{t} + \text{Re} \left[ \hat{c} e^{i(y/R)} \right] \]

where \( \hat{c} \) is the mean rotor tip gap and \( y \) is the distance from maximum tip gap in the azimuthal direction (Fig. 5). This analysis is similar to the turbine radius model described in Song and Martinez-Sanchez. Basically, continuity, axial, and tangential momentum equations are used to connect upstream and downstream flowfields on the radius scale. For a prescribed eccentric perturbations in velocity and pressure are obtained via a small perturbation about the mean solution provided by the blade scale analysis. Further details of the compressor radius scale analysis can be found in Park.

C. Calculation of Rotordynamic Coefficients

The rotordynamic excitation force can be obtained from the perturbation quantities. The tangential force exerted on the compressor per azimuthal length by the fluid can be defined as

\[ f_t = \lambda q \left( C_{12}^{+} - C_{12}^{-} \right) + (1 - \lambda)q \left( C_{12}^{+} - C_{12}^{-} \right) \]

The mean and the perturbation of \( f_t \) are, respectively,

\[ \bar{f}_t = \lambda q \left( C_{12}^{+} - C_{12}^{-} \right) + (1 - \lambda)q \left( C_{12}^{+} - C_{12}^{-} \right) \]

\[ f_t = \lambda q \left( C_{12}^{+} - C_{12}^{-} \right) \left[ \frac{\lambda - q}{1 - \lambda} + \frac{C_{12}^{+} - C_{12}^{-}}{C_{12}^{+} - C_{12}^{-}} \right] + (1 - \lambda) \]

Thus, once the values of \( \lambda \) and \( q \) that satisfy the system of Eqs. (23) and (24) have been found for a given \( t/H \), the velocities can be evaluated.

III. Model Predictions

This part presents the model predictions for a test compressor. The input flow parameters include the design flow coefficient \( \Phi_D \), the design reaction \( R_D \), and the design work coefficient \( \Psi_D \). The input geometry parameters are the mean rotor-tip gap \( \hat{t} / H \) and the pitch-chord ratio \( s / c \). Howell's \( s / c \) compressor was chosen for this study because \( \Phi_D = 0.5 \) and \( R_D = 0.5 \) are common values for compressors. The compressor's parameters are listed in Table 1. \( t / H \) values of 0.01, 0.02, and 0.04 have been used.
<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow coefficient</td>
<td>0.50</td>
</tr>
<tr>
<td>Reaction</td>
<td>0.50</td>
</tr>
<tr>
<td>Rotor inlet angle</td>
<td>52.5</td>
</tr>
<tr>
<td>Rotor exit angle</td>
<td>35.0</td>
</tr>
<tr>
<td>Pitch/chord</td>
<td>1.0</td>
</tr>
</tbody>
</table>

**Fig. 6** Radial distribution of relative tangential velocity at rotor exit.

**Fig. 7** Radial distribution of axial velocity at rotor exit.

### A. Blade Scale Predictions

Figure 6 shows the radial location plotted vs the relative tangential velocity at the rotor exit for three different mean tip clearances. \( z/H = 0 \) is the hub, \( z/H = 1.0 \) is the casing wall, and the blade tips are located at \( z/H = 0.99, 0.98, \) and 0.96. Tangential velocities are considered to be positive in the direction of rotation. In Figure 6 the relative tangential velocity of the clearance stream (\( + \)) has a higher magnitude in the direction opposite to the direction of rotation than that of the passage stream (\( - \)). Therefore, relative to the passage stream, the clearance stream is underturned, and, thus, has less swirl. In other words, relative to the passage stream, less work is imparted by the compressor onto the stream associated with tip clearance. Figure 7 shows the radial location plotted vs the axial velocity at the rotor exit for three different mean tip clearances. Axial velocity is positive in the downstream direction, and the underturned stream (\( + \)) is retarded relative to the passage flow (\( - \)) in the axial direction. The figures also show that the defects in swirl and axial momentum of the clearance stream (\( + \)) increase with increasing tip clearance. These trends agree with the experimental findings of Hunter and Cumpsty, and the comparison is presented in Roh and Song.

### B. Radius Scale Predictions

Nondimensional flow perturbations, normalized by nondimensional eccentricity \( e/H \), are plotted vs azimuthal location \( \theta \) in Figs. 8 and 9. The tip clearance is the smallest at \( \theta = 0 \) deg and the largest at \( \theta = 180 \) deg. Figure 8 shows the perturbation in axial velocity upstream of the stator (station 0) plotted vs \( \theta \). As already discussed, a bigger tip clearance means a higher mass fraction and axial momentum defect of the underturned stream. Therefore, flow migrates azimuthally away from the larger gap toward the smaller gap. Thus, the highest axial velocity occurs near \( \theta = 15 \) deg. From the Bernoulli relation between \( x = -\infty \) and station 0, pressure should be the highest where velocity reaches its minimum. Therefore, the maximum pressure near \( \theta = 200 \) deg can be seen in Fig. 9.

### C. Rotordynamic Coefficient Predictions

For the test compressor the tangential force perturbation is plotted vs azimuthal angle in Fig. 10. The mean value of the nondimensional tangential force \( f_{\theta}/mU \) is \(-0.29\). The negative sign signifies that the fluid receives work from the compressor. Also, the negative perturbation near the minimum gap (\( \theta = 0 \) deg) means that compressor does more work on the fluid in that region. Consequently, the compressor is loaded more highly in the smaller gap region.
Table 2 Predicted excitation force coefficients for the test compressor ($\Phi_D = 0.50$, $R_D = 0.50$)

<table>
<thead>
<tr>
<th>Direction</th>
<th>(\alpha_p)</th>
<th>(\alpha_{wd})</th>
<th>(\alpha_{total})</th>
</tr>
</thead>
<tbody>
<tr>
<td>(X)</td>
<td>+1.3</td>
<td>-0.2</td>
<td>+1.1</td>
</tr>
<tr>
<td>(Y)</td>
<td>+0.5</td>
<td>-0.6</td>
<td>-0.1</td>
</tr>
</tbody>
</table>

This result is mainly caused by the fact that the mass fraction of underturned stream is the largest near the maximum gap. However, the tangential force in a compressor acts in a direction opposite to that of rotation; therefore, the tangential force asymmetry promotes a backward whirl in compressors (Fig. 11). The forces as a result of the pressure nonuniformity result from the azimuthal flow redistribution caused by the rotor offset. \((P')/\rho U^2(e/H)\) is plotted vs azimuthal angle in Fig. 12. The maximum pressure is at approximately \(\theta = 200\) deg, and, thus, the pressure asymmetry promotes a forward whirl (Fig. 13).

The predicted excitation force coefficients as a result of the tangential force and pressure asymmetries for the test compressor are listed in Table 2. For the direct force the destabilizing pressure effect is approximately twice as large as the stabilizing tangential force effect. Therefore, the total direct force is positive and destabilizing. For the cross-coupled force the magnitude of the forward-whirl-inducing pressure effect is comparable to that of the backward-whirl-inducing tangential force effect. The result is a small backward-whirl-inducing force. These results are for design point operation of the test compressor, and the values would be different for off-design operation and other compressor designs.

For comparison Table 3 lists the coefficient found for a turbine. In magnitude and direction the forces in compressors are different from those in turbines. First, the magnitudes of compressor excitation force coefficients are smaller than those of turbines. This result is caused by the relatively low loading of the test compressor stage, with a loading coefficient value of 0.30, compared to the turbine, with a loading coefficient value of 1.5. Second, the rotodynamic effects of flowfield asymmetries on compressor and turbine are different. In the compressor’s case the forces caused by pressure...
asymmetry counteract the forces caused by tangential force asymmetry in both $X$ and $Y$ directions. In turbines the situation is different. In the $X$ direction the force caused by pressure asymmetry acts in the same direction as that caused by tangential force asymmetry. However, in the $Y$ direction the forces caused by pressure asymmetry and tangential force asymmetry both act in the same direction.

IV. Conclusions

The main conclusions from this investigation on flow redistribution as a result of rotor-tip clearance in compressors are listed here:

1) An actuator disk model has been newly developed to examine the flow redistribution and rotordynamic effects induced by axisymmetric and nonaxisymmetric rotor-tip clearance in a single-stage axial compressor.

2) In the axisymmetric case, relative to the passage flow, the stream associated with tip clearance is retarded in the axial direction and underturned in the tangential direction. In the nonaxisymmetric case the following apply:

3) The flow preferentially migrates toward the smaller gap region.

4) Near the larger gap the compressor is loaded more lightly.

5) The direct force caused by pressure asymmetry is positive, whereas that caused by torque asymmetry is negative. Thus, the net direct force is positive.

6) The cross-coupled force caused by pressure asymmetry is positive (inducing a forward whirl), whereas that caused by torque asymmetry is negative (inducing a backward whirl). Thus, the net cross force has a negative sign and a small magnitude.

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References


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