Effects of Low Reynolds Number on Loss Characteristics in Transonic Axial Compressor

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A three-dimensional computation was conducted to understand effects of low Reynolds numbers on loss characteristics in a transonic axial compressor, Rotor 67. As a gas turbine becomes smaller and it operates at high altitude, the engine frequently operates under low Reynolds number conditions. This study found that large viscosity significantly affects the location and intensity of the passage shock, which moves toward the leading edge and has decreased intensity at low Reynolds number. This change greatly affects performance as well as internal flows, such as pressure distribution on the blade surface, tip leakage flow and separation. The total pressure ratio and adiabatic efficiency both decreased by about 3% with decreasing Reynolds number. At detailed analysis, the total pressure loss was subdivided into four loss categories such as profile loss, tip leakage loss, endwall loss and shock loss.

Key Words: Low Reynolds Number, Loss, Transonic Axial Compressor, Rotor 67

Nomenclature

\( A_w \): endwall area
\( C \): chord length
\( C_c \): contraction coefficient
\( C_d \): dissipation coefficient
\( C_v \): specific heat capacity
\( C_x \): axial chord length
\( h \): blade height
\( M \): Mach number
\( m \): mass flow rate
\( m_{j} \): mass flow rate across tip clearance
\( \rho \): air density
\( \delta^* \): displacement thickness
\( \zeta_b \): boundary layer loss coefficient
\( \zeta_{ew} \): endwall loss coefficient
\( \zeta_{tl} \): tip leakage loss coefficient
\( \zeta_{te} \): trailing edge loss coefficient
\( s \): pitch
\( S \): entropy generation
\( S_{ew} \): entropy generated on endwall
\( t \): blade thickness
\( T \): static temperature
\( V \): absolute velocity on inertial coordinate
\( V_s \): surface velocity relative to endwall
\( W \): relative velocity
\( W_s \): surface velocity on blade surface
\( W_{sp} \): surface velocity on pressure surface
\( W_{ss} \): surface velocity on suction surface

Greek
\( \alpha \): angle between tip leakage flow and main flow
\( \beta \): flow angle
\( \gamma \): specific heat ratio
\( \Delta \): increment or decrement

Subscripts
1: inlet in computational domain
2: outlet in computational domain

1. Introduction

The Reynolds number based on operating conditions of gas turbines has been decreasing recently, because aero-engine designers are trying to reduce gas turbine size and the need to operate gas turbines at high altitudes is increasing for high-altitude UAVs (Unmanned Air Vehicles), mainly used in military surveillance and reconnaissance. It is known that wall boundary layer thickens and separation occurs frequently on blade surfaces with decreasing Reynolds number. Because these degraded internal flows have a negative effect on axial turbomachinery performance, many researchers are investigating the effects of low Reynolds number.

From the design viewpoint, a low Reynolds number condition is often attributed to low air densities at high altitudes. Weinberg and Wyzkowski\(^{(1)}\) investigated the effects of low Reynolds number on a gas turbine with NASA. The PW545 jet engine, designed to operate at 13,700 m altitude, was tested above 18,300 m. They reported that the engine efficiency and performance were reduced by the low air density.
at high altitude. Castner et al.\textsuperscript{2}) compared their numerical results with the experimental data of Weinberg and Wyzykowski.\textsuperscript{1}) There was a good agreement between the test data and the calculated efficiency and performance within Reynolds numbers from 30,000 to 295,000. Schreiber et al.\textsuperscript{3}) researched transition phenomena on the blade surface in a compressor cascade by changing the Reynolds number and turbulence intensity, finding a laminar separation bubble with reattachment on the suction surface at relatively low Reynolds number. Van Treuren et al.\textsuperscript{4}) ran wind tunnel tests at Reynolds numbers from 25,000 to 50,000 and found that a large separation near the trailing edge of a turbine cascade degraded total pressure loss. After wind tunnel tests on a turbine cascade with tip clearance, Matsunuma\textsuperscript{5}) analyzed the relationship between tip leakage flow, Reynolds number and free-stream turbulence intensity don’t affect tip clearance loss although they change the distribution of the internal flow. Using experiments on total pressure loss change caused by low Reynolds numbers from 32,000 to 128,000 in a small one-stage axial turbine, Matsunuma and Tsutsui\textsuperscript{6}) showed that the region with high turbulence intensity due to the wake and the flow fluctuation due to the stator-rotor interaction increase with decreasing Reynolds number.

Although there are some studies on the effect of low Reynolds numbers on the internal flow in turbomachinery, most of them have focused on flow phenomena but not on loss characteristics. After extensive numerical simulations, Choi et al.\textsuperscript{7}) analyzed the effects of Reynolds number on profile loss, tip leakage loss and endwall loss in a subsonic axial compressor, reporting that the tip leakage loss drops with low Reynolds number due to weaker mixing of the tip leakage and mainstream flows. However, the internal flow in a transonic compressor is more complex due to the presence of shocks.\textsuperscript{8–13}) This study investigated the effects of low Reynolds numbers on loss characteristics in a transonic compressor using numerical simulations. A comprehensive analysis of the loss mechanism was attempted by applying Denton’s loss model\textsuperscript{14}) to the computational result.

2. Test Configuration

2.1. Geometric specifications

We conducted a numerical study on the effect of low Reynolds number on internal flows and loss characteristics using a transonic axial compressor, Rotor 67, which has been tested previously by Strazisar et al.\textsuperscript{15}) Figure 1 shows the schematic diagram of Rotor 67 and measurement positions. This rotor rotates about its axis at 16,043 rpm with a relative tip Mach number of 1.38. The mass flow rate and pressure ratio between the inlet and outlet are 33.25 kg/s and 1.63 at the design condition respectively. The Reynolds number is about 1.17 × 10\textsuperscript{6} based on the inlet velocity at the design condition and meridional blade chord length on the hub. This rotor has 22 blades with an aspect ratio of 1.56.

Detailed design parameters are summarized in Table 1. The spanwise distribution of the static and total pressures, the total temperature and the flow angle were measured by Strazisar et al.\textsuperscript{15}) at aero-station 1 and 2 located upstream and downstream of the rotor.

2.2. Computational grid

Using the blade profile given by Strazisar et al.,\textsuperscript{15}) the computational domain was fixed in the region including aero-station 1 and 2, and a multi-block hexahedral mesh was generated as shown in Fig. 2. Each passage consists of 117 nodes in the streamwise direction, 57 nodes in the pitchwise direction, and 53 nodes in the spanwise direction. To capture the tip leakage flow accurately, the region of the tip clearance was filled with an embedded H-type grid, which has 52 nodes from the leading edge to the trailing edge, 9 nodes across the blade thickness and 9 nodes from the blade tip to the casing. Therefore, the computational domain has about 350,000 nodes.

2.3. Numerical method

Simulations of the three-dimensional flow were conducted using the in-house flow solver called TFlow, which has been improved to calculate internal flow in turbomachinery since its development in the mid-1990s.\textsuperscript{16}) This flow
solver has been pre-validated by a series of calculations in a transonic axial compressor, a subsonic axial compressor and a subsonic axial turbine, and has also been used to simulate rotating stall in a subsonic rotor.\textsuperscript{13,17–19} TFlow uses compressible RANS (Reynolds Averaged Navier-Stokes) equations as the governing equations, discretized in space by the finite volume method. An upwind TVD (Total Variational Diminishing) scheme based on Roe’s flux difference splitting method was used to discretize inviscid flux terms and the MUSCL (Monotone Upstream Centered Scheme for Conservation Law) technique was used to discretize viscous flux terms. The equation was solved using the Euler implicit time marching scheme with first-order accuracy to obtain a steady solution. The laminar viscosity was calculated by Sutherland’s law and the turbulent viscosity was obtained by the algebraic Baldwin-Lomax model.\textsuperscript{20}

2.4. Boundary conditions and Reynolds number

At internal flow simulations of turbomachinery, there are four types of boundaries such as inlet, outlet, wall, and periodic conditions. The total pressure, total temperature, whirl, and meridional flow angle were fixed at the inlet condition and the outgoing Riemann invariant was taken from the interior domain. The inlet total pressure was specified as a constant in the core flow and was reduced in the endwall regions with the $1/7$ power law velocity profile. The inlet boundary layer thickness on the hub and the casing was set to be 12 mm, which was estimated from the experimental data. The inlet total temperature was also specified as a constant. The whirl velocity was set as zero, and the radial velocity was chosen to make the flow tangential to the meridional projection of the inlet grid lines. For the outlet condition, the static pressure on the hub was specified and the local static pressures along the span were given by using the SRE (Simplified Radial Equilibrium) equation. Other flow variables such as density and velocity were extrapolated from the interior. To calculate velocity components on the wall, the relative velocity was set to be zero on the blade surface and rotating part of the hub, and the absolute velocity on the casing and stationary part of the hub was set to be zero as the no-slip condition. The surface pressure and density were obtained using the normal momentum equation and the adiabatic wall condition respectively. Since only one blade passage was used here, the periodic condition was implemented using the ghost cell next to the boundary cell and enabled flow variables to be continuous across the boundary.

To study the effect of Reynolds number on loss characteristics in a transonic axial compressor, the boundary conditions at each operating point were fixed and only the Reynolds number was changed by varying the kinematic viscosity. Referring to the U.S. standard atmosphere,\textsuperscript{21} the density and viscosity at 20,000 m altitude are reduced by the ratio of 1 to 0.07 and 1 to 0.77 respectively in comparison to the values at sea level. The Reynolds number at 20,000 m is 10\% of that at sea level, if other operating conditions except for the density and viscosity are the same. Therefore, simulations were conducted at five different Reynolds numbers between $1.17 \times 10^5$ and $1.17 \times 10^6$, based on the inlet velocity and meridional blade chord length on the hub. We described the lowest Reynolds number ($1.17 \times 10^5$) as Low Reynolds number or Low $Re$, and the highest Reynolds number ($1.17 \times 10^6$) as Reference Reynolds number or Ref. $Re$.

3. Computational Results

3.1. Performance curves

The total pressure ratio and adiabatic efficiency were calculated at the design speed to estimate the performance of Rotor 67. The results are shown in Fig. 3, where the horizontal axis is represented by the normalized mass flow rate, defined as the ratio of the mass flow rate to the choked mass flow rate. It can remove some uncertainties in the mass flow rate measurement in an experiment. The choked mass flow rate was $34.96 \text{ kg/s}$ in the experiment and $34.57 \text{ kg/s}$ in the simulation at Ref. $Re$, where the difference was within 1\%. The total pressure ratios from the computation agreed well with the experimental values at Ref. $Re$. The values at Low $Re$ have a decrement of about 3\% at the same flow condition, compared to Ref. $Re$. The numerical surge at Low $Re$ occurs at higher normalized mass flow rate than Ref. $Re$, resulting in the severe decrease in the compressor operating range. The computed adiabatic efficiency agrees well with the experimental value, although the peak position slightly deviates from the experimental one. There is also a 3\% efficiency drop with decreasing Reynolds number. This decrement is the same as the total pressure ratio drop. Under the same simulation conditions except for Reynolds number, the operating point moves along lines A and B on the performance curves. The Reynolds number affects the mass flow rate as well as the total pressure ratio and adiabatic efficiency. It should be noted that the decrement in the mass flow rate is also the result of the change in Reynolds number.

3.2. Internal flows

From here on, all flow data and loss characteristics are calculated at the peak efficiency with Reynolds number change. Figure 4 shows the spanwise distribution of the...
mass-averaged static pressure, total pressure, total temperature and exit flow angle with experimental data. The computed values at Ref. $Re$ are in good agreement with the experimental values, although the static and total pressures are slightly under-predicted near the casing. At Ref. $Re$, the static pressure increases along the span from the hub to the casing. The flow angle decreases along the span up to the 90% span, but it increases rapidly due to blockage of the tip leakage flow above the 90% span. The total pressure and temperature are almost constant in the core flow region, but the former decreases and the latter increases near the casing, meaning that the tip leakage flow generates large loss in this region. At Low $Re$, the static pressure distribution is similar to its counterpart, but it becomes smaller above the mid-span. The flow is over-turning from the hub to the 90% span but under-turning near the casing. The total pressure and temperature become small in the core flow region but large near the casing unlike the reference case, meaning that the total pressure loss increases from the hub to the 90% span but decreases near the casing with decreasing Reynolds number.

Figure 5 compares the experimental and computed relative Mach number contours at the 30%, 70% and 90% spans from the hub. The calculated values at Ref. $Re$ agree well with the experimental values and a shock structure is clearly observed. There are two shocks above the 70% span, such as a bow shock at the leading edge and a passage shock across the flow passage. The boundary layers on the blade surface grow rapidly downstream of the passage shock. The flow at the 30% span from the hub is well-behaved without a passage shock although the flow field has a small supersonic region at the leading edge. At Low $Re$, relatively large viscosity adds dissipative effects throughout the flow field but it does not affect the bow shock significantly because the flow field upstream of it is nearly inviscid. However, there are considerable changes in the location and intensity of the passage shock. The passage shock at the 70% and 90% spans moves toward the leading edge, enabling the boundary layer to develop more upstream than for Ref. $Re$ and a separation to occur at the trailing edge on the suction surface. The Mach number contours also show that the intensity of the passage shock becomes weak. The pressure jump across this weak shock decreases and makes the static pressure above the 60% span become smaller than the reference case (Fig. 4(a)). The decrease of the normalized mass flow rate caused by low Reynolds number leads to these different shock structures, because the shock location in a transonic compressor depends on the upstream Mach number, which is equivalent to mass flow rate in this case. The overall flow field at the 30% span has a similar feature regardless of Reynolds number, but the flow is more diffused through the passage due to the large viscosity.

Figure 6 shows the static pressure distribution on the blade surface at several spanwise positions, where it was non-dimensionalized by the inlet total pressure. At Ref.

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**Fig. 3. Performance curves.**

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**Fig. 4. Exit flow data along span at aero-station 2.**
The distribution clearly shows the passage shock position and the load variation in the streamwise direction at each span. The pressure jump occurs at the pressure side leg of the passage shock near the 30% chord and at the suction side leg near the 80% chord above the 70% span, while there is no shock below the 50% span. The static pressure distribution near the casing is important because pressure difference in this region drives tip leakage flow. At the 99% span from the hub, the static pressure on the pressure surface increases rapidly due to the passage shock and it is nearly constant behind the pressure side leg of the shock. The static pressure on the suction surface decreases continuously until it increases behind the suction side leg of the shock. The pressure difference starts increasing just behind the pressure side leg and it has the maximum value just before the suction side leg of the passage shock. Therefore, the strong roll-up of the tip leakage flow starts near the 30% chord at Ref. Re. At Low Re, each leg moves upstream so the pressure side leg is located at the 20% chord and the suction side leg at the 70% chord from the leading edge. The pressure jump becomes smaller because the passage shock is weaker than that at Ref. Re. At the 70% span, the effect of the passage shock is so small that it is not easy to recognize its location. Near the casing, the pressure difference has large values between the pressure and suction legs but its value is much smaller than that at Ref. Re. This small pressure difference weakens the tip leakage flow and retards the vortex formation.

Tip leakage flow is generated by the pressure difference between the pressure and suction surfaces near the casing.
Figure 7 shows the static pressure distribution near the casing and the particle traces placed just before and after the pressure side leg of the passage shock. Clearly, the passage shock moves upstream toward the leading edge with decreasing Reynolds number and the particle traces coincide with the static pressure trough in both cases. The pressure trough at Ref. $Re$ is deeper than that at Low $Re$, meaning that the tip leakage flow is stronger in the former case. According to particle motion, the tip leakage flow at Ref. $Re$ is more vortical than that at Low $Re$ due to strong roll-up caused by the large pressure difference across the tip. As a result of different tip leakage flow intensities, the passage shock is severely disturbed by the strong interaction with the tip leakage flow at Ref. $Re$ while it is hardly disturbed at Low $Re$.

To inspect the structure of separations on the blade surfaces, limiting streamlines are shown in Fig. 8. Similar streamlines are found on the pressure surface regardless of Reynolds number. The limiting streamlines near the tip are more disturbed by tip leakage flow at Ref. $Re$, because it has strong roll-up and larger spiral motion than its counterpart. However, the limiting streamlines on the suction surface are significantly affected by Reynolds number. At Ref. $Re$, three separations are found at the leading edge near the hub (A), at the trailing edge near the hub (B) and near the tip (C). The first is caused by the large incidence angle near the hub. The second is generally called hub-corner-separation and is frequently developed by large deflection of the blade in a subsonic flow regime. The last is caused by the passage shock and located just behind the suction side leg of the shock. At Low $Re$, some different features in each separation may be observed, although the three separations are still found on the blade surface. The separation at the leading edge (A') is larger than its counterpart (A). The hub-corner-separation expands to become the full-span separation (B') at the trailing edge. In the results of Van Treuren et al.,$^4$ and Choi et al.,$^7$ similar separations originated on the suction surface of the turbine cascade and the subsonic compressor rotor, resulting in large total pressure loss. The separation line (C') is also caused by
the passage shock and is located further upstream because the shock moves upstream with decreasing Reynolds number.

### 3.3. Loss characteristics

Figure 9 shows the mass-averaged total pressure loss of the transonic rotor with Reynolds number. The calculated loss is almost constant from Ref. Re to 350,000, but it increases rapidly near 100,000 as described by Fielding. For reference, Choi et al. found that the total pressure loss in a subsonic single rotor is distributed between the -0.35 and -0.2 powers of Reynolds numbers below 250,000. These two results suggest that a compressor might have a critical Reynolds number near 200,000 and 300,000 like a turbine, but we need more test cases to confirm this. In addition, the total pressure loss change with Reynolds number is less significant in a transonic compressor than in a subsonic compressor, meaning there are some differences in the loss mechanism between transonic and subsonic compressors. To analyze the effects of low Reynolds number on total pressure loss, we divide total pressure loss into four loss categories using Denton’s loss model: profile loss, tip leakage loss, endwall loss and shock loss. The surface velocity, which is frequently used to calculate each loss, means the velocity at the outer edge of the boundary layer and the method for obtaining this from the numerical result has been explained in the previous study.

The loss generated by the blade itself is called profile loss and it is composed of two parts: loss generated in boundary layers on the blade surface and loss caused by wake mixing. The former is proportional to the cube of the surface velocity and is calculated using Eq. (1).

\[
\zeta_{bl} = \frac{1}{0.5s \cos \beta_1} \int_0^c C_d \left( \frac{W_{bl}}{W_1} \right)^3 dx
\]  

(1)

where the dissipation coefficient, \( C_d \), is equal to 0.002 as suggested by Denton and Cumpsty. The trailing edge loss can be calculated using the following equation when blockage is dominant due to separation on the blade surface.

\[
\zeta_{tr} \equiv \left( \frac{\delta^* + t}{s \cos \beta_2} \right)^2
\]  

(2)

The boundary layer loss and trailing edge loss calculated using Eqs. (1) and (2) are shown in Fig. 10. The boundary layer loss is closely related to inlet velocity profile because this loss is proportional to the cube of the surface velocity on the blade as mentioned above. The overall distributions along the span are similar because an identical inlet velocity profile was imposed, but the magnitude of the loss decreases due to reduction of mass flow rate with decreasing Reynolds number. The trailing edge loss at Ref. Re has a large value from the hub to the 20% span due to the hub-corner-separation, but it diminishes steadily above the 40% span because there is no separation. This loss has a non-physical value above the 90% span because this region is included in the region where tip leakage flow has a major effect. At Low Re, the large trailing edge loss is generated by the full-span separation at the trailing edge from the hub to the 80% span, and it is much larger than that at Ref. Re over the full span.

The tip leakage flow mixes into the main flow and complicates the internal flow, resulting in large loss. The loss caused by tip leakage flow is calculated using the following Eq. (3).

\[
\zeta_{tl} = \frac{2C_c}{s \cos \beta_1} \int_0^c \frac{\tau}{h} \left( \frac{W_{bl}}{W_1} \right)^2 \left( 1 - \frac{W_{tl}}{W_{bl}} \right) \left( 1 - \frac{W_{tl}}{W_{bl}} \right)^2 dx
\]  

(3)

where the contraction coefficient, \( C_c \), is set to be 0.3. The calculated tip leakage loss is shown in Fig. 11 and decreases with decreasing Reynolds number. This trend is the same as the subsonic rotor in the previous study while Matsumuma has shown that tip leakage loss in a turbine cascade is almost constant regardless of Reynolds number. This tendency might be explained by the original form used in deriving Eq. (3).

\[
\frac{\Delta p_{tl}}{0.5 \rho W_{bl}^2} = \frac{m}{m_1} \left( 2 - \frac{W_{tl}}{W_{bl}} \right)
\]  

(4)

In this equation, the total pressure defect caused by tip leakage flow is proportional to \( m_1/m_1 \) and \( 2 - W_{tl}/W_{bl} \), where \( W_{tl}/W_{bl} \) indicates flow angle between the tip leakage flow.
and main flow as \( \cos \alpha = W_{lp}/W_{ls} \). The mass flow ratio at the tip clearance and inlet, \( m_j/m_1 \), and the surface velocity ratio on the pressure and suction surfaces, \( W_{dp}/W_{ds} \), are calculated and the results are shown in Fig. 12. The value of \( W_{dp}/W_{ds} \) increases as Reynolds number decreases, while the \( m_j/m_1 \) is almost constant regardless of Reynolds number. This means that the tip leakage loss drops with decreasing Reynolds number because mixing between the tip leakage flow and main flow weakens.

The loss generated in the boundary layer on the hub and casing is called endwall loss. The endwall loss coefficient is calculated using a similar method to the boundary layer loss with the surface velocity relative to the endwall, and non-dimensionalized by the inlet dynamic head.

\[
\zeta_{el} = \begin{cases} 
\frac{T_{1}\Delta S_{ew}}{0.5mV_1^2} & \text{(for stationary wall)} \\
\frac{T_{1}\Delta S_{ew}}{0.5mW_{1}^2} & \text{(for moving wall)} 
\end{cases} 
\]

\[
\Delta S_{ew} = \int_0^{\alpha_{ew}} C_d \rho V_1^3 \frac{1}{T} \, dA
\]

Because the loss on the casing is severely influenced by tip leakage flow, it is difficult to classify into tip leakage loss and endwall loss. Therefore, only endwall loss on the hub is shown in Fig. 13. The endwall loss is almost constant from Ref. \( Re \) to about 350,000 but drop slightly around a Reynolds number of 100,000. Like the boundary layer loss on the blade surface, it is probably due to the relatively small surface velocity near the endwall at Low \( Re \).

Assuming that the passage shock is a single shock wave and it is sufficiently weak, the entropy generation across the shock is represented by a function of the static pressure as Eq. (6).

\[
\Delta S \approx C_s \frac{\gamma^2}{12} \left( \frac{\Delta p}{p_1} \right)^3 + O \left( \frac{\Delta p}{p_1} \right)^4
\]

Because the passage shock exists above about the 60% span from the hub, the entropy generation is calculated in this region and the result is shown in Fig. 14. The entropy generation is almost constant over 7/10 Reynolds number but decreases significantly with further decreases in Reynolds number, because the intensity of the passage shock becomes weak as shown in Fig. 5. The weak shock diminishes the pressure increment across the shock, causing the reduced entropy generation.

4. Conclusion

Three-dimensional numerical simulations were conducted at five different Reynolds numbers between \( 1.17 \times 10^5 \) and \( 1.17 \times 10^8 \) to study the effects of low Reynolds numbers on loss characteristics in a transonic axial compressor. Compared with Ref. \( Re \), the performance at Low \( Re \) dropped by about 3% and mass flow rate also diminished significantly. The passage shock at Low \( Re \) moved toward the leading edge and its intensity also decreased.
compared to Ref. Re. This change had large effects on the pressure distribution, separation on the blade surface and tip leakage flow. The pressure difference across the blade tip decreased with decreasing Reynolds number and it weakened tip leakage flow. The passage shock movement and large viscosity at Low Re caused a full-span separation on the suction surface at the trailing edge. These changes in internal flows also affected the loss characteristics. As the Reynolds number decreased, the trailing edge loss increased while the other losses decreased. The overall loss was almost constant from Ref. Re to 350,000, but increased rapidly around 100,000. However, the increased loss in a transonic compressor with decreasing Reynolds number was smaller than in a subsonic compressor, because the former had a relatively larger Reynolds number than the latter and also than in a subsonic compressor, because the former had a compressor with decreasing Reynolds number was smaller around 100,000. However, the increased loss in a transonic compressor with decreasing Reynolds number was smaller than in a subsonic compressor, because the former had a relatively larger Reynolds number than the latter and also because the decrease in boundary layer loss, tip leakage loss, endwall loss and shock loss in a transonic compressor compensated for the increase in trailing edge loss caused by full-span separation.

References