Effects of Low Reynolds Numbers on Performance of a One-Stage Axial Compressor*

Tae-Gon Kim,1) Young-Jin JUNG,1) Yohan JUNG,2) and Minsuk CHOI1)

1)Department of Mechanical Engineering, Myongji University, Yongin 449–728, South Korea
2)Fluid Machinery Research Department, Hyundai Heavy Industries Co., Ltd., Ulsan 682–792, South Korea

Three-dimensional numerical simulations were conducted to understand the effects of low Reynolds numbers on the performance of a transonic axial compressor. For the reference case, the computational results showed good agreement with the experimental data in total pressure ratio and exit flow. With decreasing Reynolds number, the mass flow rate and total pressure ratio of the axial compressor decreased by 4% and 1%, respectively, in comparison to the reference case. This study found that large viscosity significantly affects the location and intensity of the passage shock, which moves toward the leading-edge at low Reynolds numbers. Additionally, these results successively revealed changes in internal flow pattern such as pressure distribution on the blade surface, tip leakage flow and separation. An attempt has been made to explain the dependence of the performance and the total pressure loss on the Reynolds number in a one-stage transonic axial compressor. In addition, it was confirmed that there is a critical Reynolds number around 250,000 in axial compressors, below which total pressure loss increases rapidly.

Key Words: Low Reynolds Number, Axial Compressor, Stage 37

Nomenclature

BL: boundary layer  
C*: axial chord  
LE: leading edge  
p: static pressure  
p0: total pressure  
PS: pressure surface  
SS: suction surface  
STA: measurement station  
t: calculation time  
TE: trailing edge  
V: absolute velocity  
y+: non-dimensional distance from wall  
x: axial distance from leading edge  
ρ: air density  
Δ: increment or decrement

Subscripts

0: initial  
f: final

1. Introduction

The density and the viscosity of the air at the altitude of 20 km decrease to 7% and 77% of their sea-level values respectively, according to the U.S. standard atmosphere.1) Assuming that other conditions for unmanned air vehicles (UAVs) operating at the altitude of 20 km are the same, the Reynolds number is reduced to become 10% of the sea-level value due to the low air density. The boundary layer thickness on the wall increases significantly, and therefore flow separation occurs easily at low Reynolds numbers. Since the degraded flow has a detrimental effect on the performance and is a potential threat to the operation of gas turbines for high-altitude UAVs, much attention has been paid to understand the effects of low Reynolds numbers on the internal flow in axial turbomachinery.

For considering the variation in density and viscosity of the air, Castner et al.2) conducted numerical simulations for a gas turbine with changing Reynolds numbers and compared their results to the experimental data of Weinberg and Wyzykowski.3) Within Reynolds numbers from 30,000 to 295,000, the computed efficiency and pressure ratio matched well to the test data. Schreiber et al.4) observed that a laminar separation bubble appears on the blade surface in a compressor cascade at low Reynolds numbers. Van Treuren et al.5) found in their wind tunnel tests that the formation of a large separation on a turbine cascade has a detrimental effect on the performance within Reynolds numbers from 25,000 to 50,000. Matsunuma6) reported with his wind tunnel test data that Reynolds number and inlet turbulence intensity affect the internal flow in an axial turbine cascade, but have nothing to do with loss caused by the tip leakage flow. Matsunuma and Tsutsui7) analyzed the total pressure loss variation induced by low Reynolds numbers from 32,000 to 128,000 in a one-stage axial turbine and found that a low Reynolds number causes high turbulence intensity in the wake region and increases the flow fluctuation between the stator and the rotor.

Most of the research described above has focused on the effects of low Reynolds numbers on the performance and internal flow in a turbine. Recently, Choi et al.8,9) found, based on their numerical simulations, that an axial compressor...
might have a Reynolds number threshold between 200,000 and 300,000, below which the total pressure loss increases quickly. In these studies, the axial compressor consisted of a single rotor without a stator. To determine a critical Reynolds number value, however, more research is required using different types of compressors. The present work is a continuation of the authors’ previous studies for finding a Reynolds number threshold in an axial compressor. The effects of low Reynolds number on the loss and internal flow in a one-stage axial compressor are investigated in detail.

2. Test Configuration

2.1. Geometric specifications

A numerical study on the effects of low Reynolds number on the loss and internal flow was conducted using a one-stage axial compressor, Stage 37. This compressor was tested and well-documented by Reid and Moore to and extensive experimental data was available. Figure 1 shows the schematic diagram of Stage 37 with measurement positions. The compressor consists of 36 rotor blades and 46 stator blades. The rotor rotates at 17,196.8 rpm and the corresponding relative Mach number at the tip is 1.48. At the design condition, the total pressure ratio generated by the compressor is 2.05 and the mass flow rate is 20.2 kg/s. Detailed geometric specifications are summarized in Table 1. There are three measurement points where the spanwise distributions of pressure and temperature were measured by Reid and Moore. The axial locations of the measurement points are summarized in Table 2.

2.2. Computational method and grid

Simulations of the internal flow in the compressor were conducted using a commercial flow solver, Ansys CFX-14, which uses compressible Reynolds-averaged Navier-Stokes (RANS) equations as the governing equations. The convection term was discretized by a high-resolution scheme to obtain second or higher spatial accuracy, and the derivative of the diffusion terms was calculated from the shape function based on the finite element method.

The equation was solved using a fully implicit time stepping method with first-order accuracy to obtain steady solutions and with second-order accuracy to obtain unsteady solutions. The performance curves and the internal flow through the compressor were obtained from steady simulations, while the unsteady fluctuation caused by the rotor-stator interaction was captured from unsteady simulations. The unsteady simulations were conducted only at peak efficiency because of the long calculation time. The laminar viscosity was calculated by the Sutherland law and the turbulent viscosity was obtained by the k-ω shear stress transport (SST) model. Once the mass flow rate variation at the inlet was less than 0.001 kg/s per 300 iterations, the computation was considered to reach a converged solution. This convergence criterion has been used by Chen et al. for obtaining the numerical surge point (i.e., the last stable operating point). After obtaining a converged solution, the next simulation was re-started with the appropriate increase in the back pressure to form a constant speed curve on the performance map. In unsteady simulations, the pressure time history was used to determine whether the calculation reaches the quasi-steady state with periodic fluctuations.

Using the profiles of the blade, casing and hub given by Reid and Moore, the geometry of the compressor was reconstructed with the aid of Ansys Blade-Gen. Thereafter, a multi-block hexahedral unstructured mesh was generated using the Ansys Turbo-Grid as shown in Fig. 2. Each passage consists of an O-type grid around the blades and an H-type grid far from the blades to obtain the high-quality meshes. To capture the tip leakage flow correctly, the tip clearance was filled with an O-H type grid block of 12 nodes in the spanwise direction. Consequently, the rotor has 446,784 nodes and the stator 217,008 nodes, where the value of $y^+$ at the first grid point off the wall is less than 5. As mentioned above, the original compressor has a blade ratio of 36:46. For steady simulations, one passage in each blade row was used and the total number of nodes was 663,792. For unsteady simulations, the size of the stators was increased by a factor of 46/45 to fit the blade ratio 4:5 using Rai’s reconstruction method due to the long calculation time required for whole blade passages. Consequently, the computational domain in unsteady simulations had 2,872,176 nodes for four rotor passages and five stator passages.

Table 1. Geometric specifications.

| Specification             | Value          |
|---------------------------|----------------|
| Rotational speed (rpm)    | 17196.8        |
| Mass flow rate (kg/s)     | 20.2           |
| Total pressure ratio      | 2.05           |
| Rotor tip speed (m/s)     | 455            |
| Rotor tip solidity         | 1.3            |
| Tip clearance (mm)        | 0.356          |
| Aspect ratio              | Rotor 1.19     |
|                           | Stator 1.26    |
| Relative Mach number      | Hub 1.13       |
|                           | Tip 1.48       |
| Number of blades          | Rotor 36       |
|                           | Stator 46      |

Table 2. Location of measurement positions.

| STA | Axial location (m) |
|-----|--------------------|
| 0   | -0.0273            |
| 1   | 0.0378             |
| 2   | 0.1054             |
2.3. Boundary conditions and Reynolds number

The inlet total pressure and total temperature were specified as constant values. The inlet turbulence intensity was set to be 1%. For the outlet condition, the static pressure along the span was calculated using the simplified radial equilibrium (SRE) equation, where the static pressure at the mid-span was fixed. The no-slip condition was imposed on the solid wall and the scalable wall-function was applied to obtain the velocity on the nodes off the wall. The periodic condition enables flow variables to be continuous across the boundary, because a few blade passages in both the rotor and the stator were used for the simulations.

In steady simulations, the interface between the rotor and the stator was treated using the frozen rotor model, where the pitch change is accounted for in spite of a fixed relative position. For calculating the rotor-stator interaction in unsteady simulations, the interface was treated with the sliding plane, which is a boundary condition for the interface in relative motion between rotor and stator domains. The flow quantities in one domain were continuously transferred to the other domain through the interface at different relative positions.

To analyze the effects of Reynolds number on the performance in the compressor, the boundary conditions except for Reynolds number were kept the same. The reference Reynolds number at the design point for the compressor is $9.2 \times 10^5$, which was calculated based on the inlet velocity and meridional chord length of the rotor on the hub. Since the Reynolds number at the altitude of 20 km is reduced by the ratio of 1 to 10 in comparison to the value at sea level, the simulations were conducted at five different Reynolds numbers between $9.2 \times 10^4$ and $9.2 \times 10^5$. The Reynolds number was changed by varying the kinematic viscosity. For convenience, the lowest Reynolds number was referred to as Low Reynolds number, or Low Re, and the highest Reynolds number as Reference Reynolds number, or Ref. Re.

3. Steady Computational Results

3.1. Performance curves and exit flow data

The total pressure ratio was calculated at the design speed, as shown in Fig. 3, in which only three performance curves together with the experimental data were drawn for clarity. The computed total pressure ratio at Ref. Re agrees well with the experimental data and the predicted surge point corresponds to the experiment, although the numerical model choked at a slightly lower mass flow rate than the experiment. As the Reynolds number decreases with keeping all boundary conditions the same, the design operating point with peak efficiency moves along line A accompanied by the decrease in the mass flow rate and total pressure ratio. This is because boundary layers on the blade surfaces and endwalls at low Reynolds numbers cause the effective flow area to decrease. At Low Re, the mass flow rate and total pressure ratio were reduced by 4% and 1% of the reference values, respectively.

The spanwise distribution of the total pressure ratio and total temperature ratio is calculated at STA.2 for the peak efficiency condition, and the results are shown in Fig. 4. The computed values at Ref. Re are in good agreement with the experimental data, although the total pressure and temperature are slightly over-predicted near the hub. At Low Re, the spanwise distributions of the flow properties are similar to the reference case, but the total pressure is low and the total temperature is high for all span heights in comparison to the reference values. This means that the loss generated at Low Re is higher than the loss at Ref. Re. The effect of the low Reynolds number on the total pressure loss will be investigated in a later section.

3.2. Flow differences depending on Re

Figure 5 shows the comparisons of relative Mach number contours with different Reynolds numbers. At 90% span for the reference case, there are two shocks at the leading-edge of the rotor and at the mid-chord on the pressure surface. A strong shock at the leading-edge is called “bow shock” and affects the boundary layer on the suction surface. The static...
pressure rises rapidly across the bow shock and the increased static pressure causes a thick boundary layer and eventual separation on the suction surface. The other normal shock is observed at the mid-chord on the pressure surface to cause separation. At Low Re, the bow shock is detached from the leading-edge and its suction side leg also moves towards the leading-edge, which enables the boundary layer to separate more upstream than at Ref. Re. The movement of the shocks is caused by the reduction in the mass flow rate at Low Re. The shock on the pressure surface also moves forward and causes the boundary layer to thicken. The widely spaced contour lines around the shock waves at Low Re indicate that the intensity of the shocks becomes weak.

The non-dimensional static pressure distribution on the blade surface at different spans is shown in Fig. 6. At Ref. Re, the normal shock is clearly visible at the mid-chord of the pressure surface all the spans. The rapid pressure change caused by the bow shock at the leading-edge appears only near the casing, because the shock is detached from the leading-edge below 90% span. The pressure on the suction surface rises rapidly across the suction side leg of the bow shock. At Low Re, the bow shock is detached from the blade completely, so there is no pressure rise near the leading-edge on the pressure surface, while the bow shock still causes the pressure to rise on the suction surface. The normal shock also causes the pressure to change on the pressure surface. However, the pressure rise caused by the two shocks at Low Re is much smaller than that at Ref. Re, which implies that the strength of the shocks diminishes as the Reynolds number decreases. This is because the thick boundary layers on the wall at Low Re decrease the effective flow area and eventually reduce the mass flow rate.

Figure 7 shows the static pressure distribution on the casing under the design condition. The pressure difference between the pressure and suction surfaces across the tip is a main driving force of tip leakage flow. The trajectory of tip leakage flow coincides with the static pressure trough. The pressure trough at Ref. Re is longer than that at Low Re, implying that the tip leakage flow at Ref. Re might be stronger than that at Low Re. The closely spaced pressure contour lines at Ref. Re indicate the positions of the bow shock (A) and the normal shock (B), and imply an abrupt static pressure rise across them. It is well known that the
static pressure change across a shock is closely related to the intensity of the shock. The static pressure contour lines around the two shocks diffuse as the Reynolds number decreases. The static pressure distribution through the stator is similar in both cases regardless of the Reynolds number.

The boundary layers on the blade surfaces are significantly affected by shocks and corner-separations in an axial compressor. Figure 8 shows the limiting streamlines on the rotor blade surfaces to analyze the structure of separations. At Ref. Re, the boundary layer on the pressure surface separates at the mid-chord above 60% span due to the normal shock and reattaches quickly to the surface. Below 60% span, there is no evidence of separation in the boundary layer. However, the normal shock at Low Re causes flow separation and reattachment above 30% and the corresponding radial transport of low-momentum fluid from the hub to the mid-span. This is because the boundary layer at Low Re is thicker and more susceptible to separation due to a larger viscosity and lower mass flow rate than that at Ref. Re. On the suction surface, similar streamlines are found regardless of Reynolds numbers. An abrupt pressure rise across the suction side leg of the bow shock causes large separation without reattachment from the hub to the tip. It is evident that the low Reynolds number makes the bow shock move upstream and the boundary layer to separate early.

The flow on the stator blade surfaces is also affected by the Reynolds number. The limiting streamlines on the stator surfaces are shown in Fig. 9 for two different Reynolds numbers. The limiting streamlines on the pressure surfaces are the same independent of Reynolds numbers. However, the limiting streamlines on the suction surfaces are completely different from each other, depending on the Reynolds number. At Ref. Re, the limiting streamlines move smoothly on the suction surfaces in the core flow region in spite of two corner separations caused by the endwall boundary layer. At Low Re, however, two large separation lines appear at the leading-edge and trailing-edge due to an increased incidence angle. These separations are the main factor for the decrease in total pressure ratio.

3.3. Loss characteristics

Figure 10 shows the total pressure loss coefficients at five different Reynolds numbers, which were calculated by Eq. (1).

\[ \zeta = \frac{P_{t2} - P_{t1}}{0.5 \rho_1 V_1^2} \]
As shown in the figure, it seems that the compressor has a critical Reynolds number near 250,000. Total pressure loss remains nearly constant when the Reynolds number is greater than the critical value and the loss increases rapidly below the critical value. Referring to Choi et al.,9) a transonic rotor also has a critical value of Reynolds number around 250,000. These two results imply that an axial compressor has a critical Reynolds number between 200,000 and 300,000, below which total pressure loss rises quickly. For reference, total pressure loss in an axial turbine increases rapidly when the Reynolds number is below 100,000 as described by Fielding.13) In this study, total pressure loss of the one-stage transonic compressor varies according to the $-0.22$ power of the Reynolds number. These results are in a good agreement with the results reported by Matsunuma,6) Choi et al.9) and Fielding.13) In their results, total pressure loss in an axial turbomachinery is proportional to a power of Reynolds number between $-0.35$ and $-0.20$.

Figure 11 shows the entropy distribution just behind the rotor to evaluate the total pressure loss distribution. The wake and boundary layer on the casing affect the entropy distribution dominantly. The wake at Low Re is much thicker than that at Ref. Re, because the shocks in the rotor passage move forward and cause the boundary layers on the blade surfaces to separate further upstream as the Reynolds number decreases. By the way, it is evident that the boundary layer on the hub is less affected by the Reynolds number.

According to Denton’s loss model,14) total pressure loss can be divided into the profile loss, tip leakage loss, endwall loss and shock loss. In previous studies, Choi et al.8,9) analyzed the variation of each loss category depending on the Reynolds number in an axial compressor. Table 3 summarizes their results for total pressure loss variation in relation to the Reynolds number. Although the overall total pressure loss in an axial compressor increased rapidly with decreasing Reynolds number, the loss generated by the boundary layers on blade surfaces, tip leakage flow and shock waves decreased at a lower Reynolds number. The dependency of the endwall loss on the Reynolds number was weak. The flow separation and thick boundary layer on blade surfaces at a low Reynolds number caused a large loss in the wake, thereby making the overall loss increase. A more detailed description of the relation between total pressure loss and Reynolds number can be found in Choi et al.8,9)

The difference in total pressure between Low Re and Ref. Re was calculated using Eq. (2) behind the rotor and stator and normalized by the inlet stagnation pressure. The com-
puted results are presented in Fig. 12 to analyze the loss variation along the span depending on the Reynolds number.

\[ \Delta p_t = p_{t,Low \, Re} - p_{t,Ref \, Re} \]

(2)

Behind the rotor, the total pressure difference has negative values from 10% to 65% span because the wake loss is dominant and increases significantly at \( Low \, Re \). Above 65% span, the decreased shock loss offsets the increased wake loss. Around 95% span, the total pressure difference increases locally due to the weak tip leakage flow. The total pressure at \( Low \, Re \) is smaller than the total pressure at \( Ref. \, Re \) close to the endwall, because the wake loss is dominant rather than the boundary layer loss. There is a sharp increase in the total pressure difference around 5% span. This is because the flow directed to the hub region due to the reduction in effective flow area at \( Low \, Re \) raises the total pressure in this region. Behind the stator, the total pressure difference between \( Ref. \, Re \) and \( Low \, Re \) increases over the span in comparison to the values behind the rotor since the wake loss generated by the stator is dominant.

### 3.4. Discussion for critical \( Re \)

It is necessary to analyze why the critical Reynolds number is near 250,000 in an axial compressor when considering total pressure loss. As listed in Table 3, the wake loss plays an important role in increasing total pressure loss and it is closely related to the separation on the blade surfaces. Figure 13 shows the limiting streamlines on the blade surfaces at the second lowest Reynolds number (i.e., 30% of \( Ref. \, Re \)). In the rotor, the limiting streamlines look similar to \( Ref. \, Re \), although the separation line caused by the shock moves slightly upstream. The limiting streamlines on the pressure surface of the stator remain the same independent of the Reynolds number. On the suction surface of the stator, however, separation initiates at the trailing-edge due to an increased incidence angle, but the flow at the leading-edge is still attached. If the Reynolds number decreases a bit further and the separation on the suction surface of the rotor becomes larger due to shock movement, a more increased incidence angle causes larger separations on the stator surface as shown in Fig. 9(d). It is evident that separation at the leading-edge of the stator is a major cause for the abrupt increase in total pressure loss at the critical Reynolds number.

### 4. Unsteadiness for Different Reynolds Numbers

Two numerical sensors were installed at 10% and 90% spans from the hub between rotors and stators to obtain the static pressure histories shown in Fig. 14. The pressure history can be used to judge whether an unsteady simulation converges or not and to measure unsteady interactions between rotors and stators. The unsteady flow simulation for two different Reynolds numbers has been continued during 25 periods to obtain converged solutions, where one period means the time duration that it takes the four rotor blades to traverse the five stator blades once. During the last period, four large fluctuations appear periodically in each pressure history due to the rotor passing, implying the unsteady result reaches a quasi-steady state. The fluctuation amplitude is closely related to the unsteadiness caused by the rotor-stator interaction. At 10% span, the maximum fluctuation amplitude for \( Low \, Re \) is larger than that for \( Ref. \, Re \), implying that the flow near the hub becomes more unsteady as the Reynolds number decreases. However, the situation is reversed at 90% span. The fluctuation amplitude in the pressure history decreases significantly at \( Low \, Re \) in comparison to the counterpart. This implies that the flow near the casing is more stable at low Reynolds numbers.
Figure 15 shows the turbulence intensity distributions behind the rotors. Generally, the regions of high turbulence intensity coincide with the regions of high loss. At Ref. Re, high turbulence intensity is caused by the tip leakage flow, wake, separation and boundary layers on the wall. At Low Re, there are some differences when compared to the reference case. Firstly, the regions of high turbulence intensity near the casing almost disappear because the tip leakage flow weakens at low Reynolds numbers, as shown in Fig. 7. Secondly, the sizes of wakes and separations in the circumferential direction increase but their intensity decreases significantly with decreasing Reynolds number. This phenomenon might be related to the position and intensity of the shock waves. The weak shock waves at low Reynolds numbers alleviate the adverse pressure gradient in the boundary layers, resulting in relatively weak flow separation on the rotor suction surfaces. Thirdly, the boundary layers grow significantly on the hub surfaces. At low Reynolds numbers, the distribution of high turbulence intensity behind a compressor rotor is different from the distribution behind a turbine rotor reported by Matsumata and Tsutsui.7) In their experimental data using an axial turbine, the turbulence intensity behind the rotor increases significantly as the Reynolds number decreases.

There is no shock wave and tip leakage flow in the stator passages so that the boundary layers on the blade surfaces and the endwalls change turbulence intensity distribution considerably depending on the Reynolds number. As shown in Fig. 16, high turbulence intensity appears in the wake of the stator blades and the boundary layers on the casing. The passage vortex in the corner of endwalls and suction surfaces also increases turbulence intensity. In comparison to the reference case, a large separation occurs on the stator suction surfaces due to large incidence angles at Low Re, as shown in Fig. 9. This separation increases turbulence intensity considerably in the wake. Through the stators, flow unsteadiness increases over all spans as the Reynolds number decreases. This phenomenon was also observed in the turbine nozzle flow reported by Matsumata and Tsutsui.7)

Figure 17 shows circumferentially area-averaged turbulence kinetic energy to measure flow unsteadiness for two different Reynolds numbers. Behind the rotors, as the Reynolds number decreases, the turbulence kinetic energy also decreases significantly because of weaker tip leakage flow.
5. Conclusions

The effects of low Reynolds numbers on the flow in a transonic one-stage axial compressor were evaluated using numerical flow simulations with five different Reynolds numbers. The following conclusions were drawn from a comparison of the numerical results.

1. The mass flow rate and total pressure ratio at Low Re are reduced by 4% and 1%, respectively, in comparison to values at Ref. Re for the peak efficiency condition.

2. At low Reynolds numbers, large viscosity significantly changes the internal flow in an axial compressor. In particular, tip leakage flow and passage shock are weakened as the Reynolds number decreases.

3. It is well known that total pressure loss in an axial turbine increases rapidly when the Reynolds number is smaller than 100,000. In this study, it was confirmed that there is a critical Reynolds number around 250,000 in an axial compressor, below which total pressure loss rises rapidly.

4. Only the wake loss increases as the Reynolds number decreases. It was found that the separation on the suction surface of the stator is a critical factor for increasing the wake loss. The other losses generated by tip leakage flow, shock wave and boundary layer on the wall decrease at low Reynolds numbers.

5. The unsteadiness caused by the rotor-stator interaction does not always increase at low Reynolds numbers. Due to the weak tip leakage flow and shock waves, the unsteadiness is reduced above the mid-span but increases below the mid-span behind the rotors. Behind the stators, however, turbulence intensity increases significantly when a large separation appears at low Reynolds numbers.

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