DESIGN AND DEVELOPMENT OF A TWO STAGE
TRANSONIC AXIAL FLOW COMpressor

Katsushi NAGAI, Kazuaki IKESAWA, Takao SUGIMOTO,
Toshinao TANAKA, Hiroshi UMINO and Takeshi RYUGUJI
Engineering Department, Industrial Gas Turbine Division
Kawasaki Heavy Industries, Ltd.
Akashi, Japan

ABSTRACT

A highly loaded two stage transonic axial flow compressor, which forms a front stages of a multi stage compressor for industrial gas turbines, has been
designed and tested. Overall pressure ratio is 2.25 and
the first stage rotor tip Mach number is 1.15. Two airfoil
types, Double Circular Arc airfoil and Multi Circular Arc
airfoil, were designed for a transonic rotor blade under
the same condition. MCA blade design method was
devised and introduced. The blade design relied heavily
on CFD techniques using an Euler code and a Navier
Stokes code to cope with a precise treatment. The rig
test was conducted by our compressor test facility to
verify a validity of the transonic compressor design
method and to compare the performance of the DCA and
the MCA airfoils. This report describes the aerodynamic
design and the test results as well as the test facility and
instrumentation.

NOMENCLATURE

C total chord
fi function
G mass flow rate
L front chord
M Mach number
mj ratio of front chord to total chord
mk ratio of front camber to total camber
Pt stagnation pressure
t maximum thickness/chord
\( \beta \) relative flow angle
\( \delta \) deviation angle
\( \eta \) adiabatic efficiency
\( \theta \) total camber angle
\( \iota \) suction surface incidence angle
\( \xi \) stagger angle
\( \pi \) total pressure ratio
s solidity

Subscripts:
1 inlet
2 outlet
d design point
le front side
r relative
te rear side

INTRODUCTION

In the field of industrial gas turbines, following a
demand of saving energy, a gas turbine with higher
efficiency has been required. Pressure ratio as well as
turbine inlet temperature has been increasing steadily
to get higher thermal efficiency. Since a futile increase
of stages is not allowed from the view point of cost and
rotor dynamics, pressure ratio of a stage is required to
increase. In order to get a high stage pressure, it is
essential to enlarge inlet Mach number and to introduce
a transonic flow compressor. At the same time a high
efficiency is indispensable in the field of industrial gas
turbines. A transonic compressor of which efficiency is
lower than a conventional subsonic compressor can not
be adopted anymore. Design target of a two stage
transonic compressor was chosen considering these
requirement. The compressor specification is shown in
table 1. The total pressure ratio 2.25 was the highest
level in the industrial use compressors. The mass flow
and rotating speed were settled respecting our test
facility. Both the first and second stages were chosen to
be a shock-in-rotor transonic compressor.

Table 1 Compressor Specification

<table>
<thead>
<tr>
<th>Air flow</th>
<th>21.2 kg/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure ratio</td>
<td>2.25</td>
</tr>
<tr>
<td>Rotation</td>
<td>15,160 rpm</td>
</tr>
</tbody>
</table>

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When developing a transonic compressor, an important subject is to design an airfoil shape which gives good performance in transonic flow region. Multi Circular Arc airfoil is a suitable airfoil for a transonic inlet flow and many MCA transonic airfoils are reported (Ref. [1],[2],[3]). But there is also successful reports of a transonic compressor using Double Circular Arc airfoil (Ref. [4]). Therefore, it was beneficial to test both airfoils and to judge which airfoil was suitable or desirable for this compressor. Further, a decisive blading method of both DCA and MCA airfoils for a transonic inlet flow has not been established and precise empirical data have also not been published. In this test program, both DCA and MCA airfoils were designed and tested on the same blading condition. They were designed by our own blading method. Profiles were evaluated by a quasi-three dimensional and a three dimensional Navier Stokes flow analysis.

In this paper, the aerodynamic features of a transonic compressor blade design and the test result of a two stage transonic compressor are described as well as Navier Stokes flow calculations.

AERODYNAMIC DESIGN

Stage design

When specifying this two stage compressor, it was taken into consideration that this compressor could form a front stages of an industrial multi-stage compressor and had to match its succeeding stages. In order to attain the total pressure ratio 2.25, the two stages were decided transonic. The stage pressure ratio of the first and second stages were 1.55 and 1.45, respectively. The stage pressure ratio of 1.55 is high level for an industrial heavy duty gas turbine compressor. The specification of the main design parameters of the transonic rotors are shown in table 2.

<table>
<thead>
<tr>
<th>Table 2 Rotor Design Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total pressure ratio</td>
</tr>
<tr>
<td>Hub/Tip ratio</td>
</tr>
<tr>
<td>Aspect ratio</td>
</tr>
<tr>
<td>Tip relative Mach number</td>
</tr>
</tbody>
</table>

The vortex design was done so as to avoid a large axial velocity deflection along span. A streamline curvature analysis method (Ref. [5]), which can handle an effect of streamline slope and curvatures, was used to calculate a radial equilibrium of the axisymmetric flow through blades. The relative inlet Mach number was settled not too high respecting efficiency. The relative inlet Mach number of the first stage rotor varies from 1.15(tip) to 0.95(hub) and that of the second rotor varies 1.02(tip) to 0.89(hub). The stator inlet Mach number was controlled not to exceed 0.8. The compressor inlet blockage, accounting for boundary layer growth along the casing wall, was estimated to be 3% of the inlet area. A wide chord with aspect ratio 1.1 was used for the first rotor blade with the purpose of improving compressor performance at a partial load and a large surge margin.

Rotor blade design

In the process of designing a transonic axial compressor, an important subject was how to get a blade configuration which does not produce a strong shock loss at a supersonic inlet flow. It is commonly known that significant airfoil design parameters for supersonic inlet are optimum incidence angle, critical channel area margin (the ratio of the minimum area to the critical area) and deviation angle. Incidence angle for supersonic inlet is usually defined by suction surface incidence. The incidence angle for a transonic flow was sometimes calculated following the unique incidence requirement (Ref. [1],[2]) and sometimes chosen to be constant, such as 0 deg in Ref. [4]. Deviation angle was often settled by a modified form of the Carter deviation correlation (Ref. [1],[4]). They were mainly based on a two dimensional treatment with an empirical modification of a three dimensional effect. But pressure ratio of a transonic compressor is actually so high that the outlet annular flow area is considerably reduced to control its velocity diffusion. Each stream tube height of a meridional plane alters from the inlet to the outlet of a blade and its variation depends on the pressure ratio. Consequently, the blade design of transonic compressors is no longer handled as a two dimensional treatment with a conventional three dimensional effect modification. Two dimensional cascade data can not be employed. Further, as inlet Mach number of a transonic compressor changes from subsonic at hub to supersonic at tip, it is difficult to get optimum incidence and deviation distributions along span continuously by a combination of a subsonic design method and a supersonic design method. From these consideration, blading of this transonic compressor was designed directly using a quasi-three dimensional Navier Stokes flow analysis which can cope with a stream tube height variation and a complicated flow of subsonic and supersonic regions.

Two types of airfoil, DCA airfoil and MCA airfoil, were chosen for transonic blade profiles. Both blades were designed on the same condition: velocity triangles, chord length, maximum thickness and solidity.

DCA blading

DCA airfoil has been mainly used for a high subsonic compressor and its profile parameters usually are designed by NASA SP36(Ref. [6]). It is possible to compute DCA profile parameters for a supersonic inlet flow by SP36. Table 3 is a Navier Stokes calculation result of a transonic DCA airfoil which was designed by the SP36 program. A two dimensional Dawes code was applied to this calculation. Although the outlet static pressure was varied, it was impossible to adjust the inlet Mach number and the flow angle to the design value at the same time due to a shock wave.
The nearest calculation result to the design condition was shown in the table 3. The outlet flow angle was considerably differed from the design value used in the SP36 and a strong passage shock was occurred. This suggested that DCA airfoil designed by SP36 did not satisfy the design velocity angle in the case of a supersonic inlet flow.

Table 3 N-S analysis of DCA airfoil designed by SP36
<table>
<thead>
<tr>
<th>Solidity</th>
<th>1.333</th>
<th>Maximum Thickness/Chord</th>
<th>0.062</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet Mach number</td>
<td>1.04</td>
<td>1.05</td>
<td></td>
</tr>
<tr>
<td>Inlet flow angle</td>
<td>55.55 (deg)</td>
<td>54.12 (deg)</td>
<td></td>
</tr>
<tr>
<td>Outlet flow angle</td>
<td>37.61 (deg)</td>
<td>31.12 (deg)</td>
<td></td>
</tr>
</tbody>
</table>

In order to design transonic DCA blades, several airfoil sections along span were designed on conical surfaces. A design method which could settle optimum incidence angle and deviation angle was studied. As DCA airfoil is consist of one circular arc for each suction and pressure surface, variable parameters to adjust incidence angle and deviation angle are only camber and stagger angle if maximum thickness and leading and trailing radius are given as this case. In order to find satisfactory parameters which satisfy the design velocity angle, Navier Stokes flow analysis was adopted. Dawes code was applied for a Navier Stokes solver. It was essential to respect streamtube variation because it fairly affected the flow in high pressure ratio compressor. Therefore, every cascade flow was analyzed by a quasi-3D Dawes code, which has provision for variable streamsheet thickness. The streamsheet thickness was assumed constant up and down stream of the cascade with a sine curve variation between blade leading and trailing edges. The streamtube height ratio was determined by the radial equilibrium calculation at inlet and outlet stations of a blade in the streamline curvature calculation. Actual procedure was as follows. As a first approximation, DCA airfoil profile was designed by SP36 and its blade-to-blade cascade flow was calculated using the quasi-3D Navier Stokes solver. If the calculation result did not satisfy the design velocity triangle, then the stagger angle and/or the camber angle were changed. Several combinations of camber and stagger (combination of incidence and deviation angle) satisfied the velocity triangle. An optimized blade shape was chosen among them complying with our assessment guideline. The guideline was to minimize the loss calculated by the Dawes code and to achieve a smooth velocity deceleration on the blade surface. The suction surface incidence angle of the final first rotor DCA airfoil was -2.4 deg at hub and -0.4 deg at tip. The deviation angle varied from 5 deg to 10 deg.

MCA blading

Many MCA airfoils have been used for transonic flow axial compressors. The reason is that MCA can be adjusted more freely to fit supersonic inlet flow. Front camber angle and rare camber angle could be separately modified in order to control maximum Mach number on a leading suction surface (See Fig. 1). A ratio of front camber to total chord "mj" (location of the maximum thickness was also settled at mj in this program) and a ratio of front camber to total camber "mk" are the additional parameters to form a MCA airfoil in comparison with DCA. But this means that there are additional parameters to be set in order to shape MCA airfoil. It would be a hard work to seek optimum value of these parameters for each airfoil sections and for each transonic compressors separately. Further, a method to settle these parameters for any flow conditions was not made public. Therefore, a preliminary study was conducted to get an approximate design rule of MCA airfoil. Flow analysis of many MCA airfoil configurations at various inlet and outlet conditions were calculated by a 2-dimensional Denton code, which is a non-viscous Euler solver. It was used because of a short cpu time. The airfoil thickness, solidity and inlet-outlet flow conditions were restricted within the expected values for a transonic compressor. Airfoils which did not produce a strong shock were chosen among the MCA airfoils and regulated with their flow conditions. By statistical management of these result, design formulas of each mj, mk, and deviation angle were formed with a function of inlet Mach number, camber angle, solidity, stagger angle and maximum thickness ratio. Incidence angle was adjusted to satisfy the unique incidence. The optimum mj and mk were ruled by the following expressions.

\[
m_j = f_1(M_i, \theta, t) \cdot \sin (\xi / \sigma)
\]

\[
m_k = f_2(\theta)
\]

The deviation angle of the prescribed airfoils was written by the next expression.

\[
\delta = f_3(\theta, t) \cdot M_i \xi / \sigma
\]
When designing this transonic compressor blades, firstly, MCA profile parameters were calculated from iteration of the above formulas so as to satisfy the design condition. As this profile was founded on the non-viscous 2-dimensional flow analysis, its blade-to-blade flow was assessed again by the quasi-3D Navier Stokes analysis in order to respect a boundary layer viscous effect and a streamtube height variation effect. The profile parameters $m_j$, $m_k$, incidence angle and/or deviation angle were adjusted to achieve the design velocity triangles and to satisfy the assessment guideline. The guideline was that a good transonic profile must have a smooth velocity deceleration without a strong shock. The $m_j$ and $m_k$ as well as the incidence angle and the deviation angle were smoothly varied from hub to tip. In the case of the 1st rotor blade, the $m_j$ varied from 0.4 at hub to 0.6 at tip and the $m_k$ altered from 0.25 to 0. The suction surface incidence angle $\alpha_s$ varied from -2 deg at hub to 2 deg at tip. The deviation angle varied from 8 deg to 12 deg.

Figure 2 shows the airfoil profiles of the first rotor blades at hub, mean and tip. It is recognized that the leading suction surfaces of the MCA are more straight than that of the DCA and the MCA has a peculiar shape. Figure 3 and 4 are the calculated Mach number distributions on the DCA and MCA blade surfaces of the 1st rotor airfoil at mid span. Mach number of the MCA airfoil decelerate more smoothly than the DCA airfoil. As this selected DCA airfoil was the best choice among our examined DCA airfoils. This was a limitation of DCA type airfoil because it was difficult to get both the desirable Mach number distribution for supersonic inlet and the design velocity triangle by controlling only one circular arc.

Stator vane design

DCA airfoil was adopted for the stator vanes of both the 1st and 2nd stages as their inlet flow was subsonic. They were designed using the NASA Sp36 program. For the inlet guide vane, NASA 63 series airfoil was applied.
effect and to confirm appropriateness of the transonic blade design method. The mesh size was 33 x 59 x 33. It was equipped with a tip clearance of 0.5% span. The mesh is shown in Fig. 5. The boundary conditions were as follows:

1) The hub was rotated with rotor blades but the casing surface was stationary.
2) Total pressure and temperature at the inlet boundary were held constant as the design values.
3) The inlet absolute tangential velocity were fixed to allow the inlet angle to emerge from the unique incidence condition.
4) For the outlet boundary condition, the static pressure at hub was held at the design value.

The rotating speed was 15160 rpm. To achieve convergence which was defined as a three order of magnitude RMS residue reduction, 2000 time steps were required.

Mach number and flow angle at the inlet boundary agreed very well with the design values. Relative Mach number and flow angle distributions at the outlet boundary are shown in Fig. 6 and Fig. 7, respectively. These are the mass averaged values along peripheral direction at each radius. Both Mach number and flow angle at the outlet boundary were almost consistent with the design value except at hub and tip regions, where casing wall boundary layer and jet flow from the tip clearance influenced and disturbed its flow. Therefore, according to the 3D N-S flow calculation, it was suggested that our blade design method for a transonic compressor was proper and reliable.

Figure 8 is Mach number distribution of the blade-to-blade flow at mid span. In the MCA blade, an attached oblique shock was recognized and flow was decelerated smoothly through the channel and there was not a strong shock which might produce a large shock loss. On the other hand, in the DCA blade, a detached normal shock across the passage was observed. Besides, the shock-boundary layer interaction led to the separation. The change of the relative total pressure from inlet to outlet was shown in Fig. 9. The relative total pressure of the DCA blade dropped larger than the MCA blade. The mass averaged adiabatic efficiency was also calculated based on the inlet and outlet total pressure and total temperature. The calculated efficiencies were 93.4% of the MCA and 90.8% of the DCA. The efficiency of the MCA rotor blade was 2.6% higher than that of the DCA blades.
TEST FACILITY

Test compressor

A sectional view of the two stage transonic compressor is shown in Fig. 10. The test compressor is composed by inlet casing, inlet guide vane (IGV), the first stage rotor and vane, the second stage rotor and vane, diffuser and exhaust casing. The axial spacing's between the IGV, the 1st rotor, the 1st stator and the second rotor were saved to traverse pitot probes. The casing was made of 5 sections. Two sections could contain probes and rotated circumferentially in order to traverse. Their position was controlled by the electric motor. Although stagger angles of the IGV, 1st stators and 2nd stators were variable, their angles were fixed mechanically at the design position during this test. Two types of rotor blades, MCA and DCA, were manufactured and installed. The rotor blades for the 1st and 2nd stage were replaced and the stator vanes were used in common. Photograph of the compressor rotor is shown in Fig. 11. Hereafter, a test compressor assembled by MCA rotor blades is called MCA compressor and a compressor by DCA rotor blades is called DCA compressor.

Test facility

The test program was conducted in our compressor test facility shown in Fig. 12. A variable speed induction motor with a gearbox to provide speed range capability was used as the driver of the test compressor. Torque meter was co-axially connected with the test compressor to measure compressor consuming power. The inlet airflow was drawn through filters prior to a ventury, then through an inlet plenum to provide a uniform total pressure and temperature profile to the test rig. The air flowed in from the side perpendicular to the compressor axis. A throttle valve and a blow-off valve were equipped at the discharge duct in order to adjust the exhaust pressure of the compressor and to escape surging respectively. After overall performance test was finished, traverse measurement near the design point was conducted.
INSTRUMENTATION

Airflow was measured by means of a venturimeter which was designed to the specifications defined by the British Standards. The overall performance was measured using four total pressure rakes and four temperature rakes installed at the outlet of the 2nd stator. Each rakes were equipped with six sensors. The wall static pressures were measured at a position between each stage on the outer and inner casing as shown in Fig. 13. The temperature rakes were calibrated by our wind tunnel over the expected Mach number range to determine recovery factor. The total pressure rakes were calibrated as functions of Mach number and pitch angle. All pressures from rakes and wall static pressure taps were measured with transducers on scanivalves and recorded by an automatic data acquisition system. Flow rate and temperature rise of the lubrication oil for the compressor bearings were also measured to grasp mechanical loss power of the test compressor. After measuring the overall performance of the compressor, three-hole cylindrical probes(Yaw-meter) were installed and traversed radially and circumferentially to measure internal flow at the outlet plane of IGV, the 1st rotor and the 1st stator. These probes were calibrated as the functions of Mach number, yaw and pitch angle. During the traverse tests, the compressor operating condition was held steady. These probes were driven automatically through the scheduled matrix of test positions. These traverse probes were removed during the overall performance test.

TEST RESULTS

Overall performance of the DCA and MCA compressor were measured at 80 %, 90% and 100% of the design speed. They are shown in Fig. 14 and 15. Compressor characteristic curves are drawn by the non-dimensional expression as bases of the design pressure ratio $\pi_d$, the design air flow $G_d$ and the design adiabatic efficiency $\eta_d$. The adiabatic efficiency of the compressor was measured by two ways. The one was based on the temperature rise and the other was the consuming power, which was calculated using the measured torque and mechanical loss. Both efficiencies were identical within 0.5 % at the design speed. The long dotted line shows the estimated surge limit except the measured points of 80% speed since the compressors were not dropped into surge even at the throttle valve closed due to a leakage of the valve. It can be seen that the target pressure ratio and mass flow were achieved in both compressors. But regarding the adiabatic efficiency at the design pressure ratio, DCA compressor was 2 % lower than the design target. On the contrary, MCA compressor succeeded the target efficiency. The mass flow of both compressors at the design pressure ratio were 3-4% larger than the design value. This was mainly attributed to miss estimation of the inlet blockage. The maximum efficiency of the MCA compressor was 2% higher than that of DCA at the design pressure ratio.
This corresponds to the Navier Stokes flow analysis. The DCA airfoil blade-to-blade flow calculation suggested that there existed normal shock at inlet and this shock loss caused the deterioration of efficiency. On the other hand, at the partial speed (90% or 80% speed), the adiabatic efficiency of the MCA and DCA compressor are almost equal.

Figure 16 and 17 show total pressure distribution at the outlet of the 1st rotor. They were gotten by the pitot tube spanwise traverse measurement at the outlet of the 1st rotor near the design pressure ratio operating point. They were compared with the three dimensional Navier Stokes calculation. The measured total pressure of the MCA was almost the same as the calculation. But in the case of the DCA compressor, the measured total pressure at the span of 75%-100% was lower than the calculated distribution. It corresponded with a high inlet Mach number region. It was estimated from the flow analysis that this was caused by the passage shock loss. But this large pressure drop could not be predicted by the Navier Stokes calculation.

CONCLUSION

1) A two stage transonic compressor was manufactured and tested. Two types of transonic airfoil, MCA and DCA, were designed and compared on the same condition. The test results showed that both the DCA and MCA compressor achieved the design values successfully. But the adiabatic efficiency of the DCA compressor was lower than the design target.

2) The design procedures of DCA and MCA profile for a transonic compressor was described. One MCA blading method to get an optimum profile for a transonic compressor was introduced. The transonic profile design was relied on the quasi-3D Navier Stokes flow calculation and considered the smooth velocity deceleration on the blade surface. This test result suggested that our design procedure and methods were useful in developing a industrial transonic compressor.

3) In this test program it was concluded that MCA airfoil was more suitable than DCA for a transonic compressor blade thanks to the shaping flexibility.

REFERENCES