Development of the Centrifugal Compressor for the Nissan Vehicular Gas Turbine

K. NAKANISHI  
Research Engineer

T. WATANABE  
Research Engineer

S. YAMAZAKI  
Senior Research Engineer

New Power Source Research Dept.,  
Central Engineering Labs.,  
Nissan Motor Co., Ltd.,  
Yokosuka, Japan

This paper outlines the development process of the centrifugal compressor of the Nissan vehicular gas turbine YTP12. The compressor should provide a solid performance as a component of a vehicular prime mover. As we have managed to achieve the design performance by modifying the components of the compressor within the limited sphere permitted by the dimensions of a given engine, this paper will cover the key processes of improvement, as well as some additional information which may be applicable to future compressor designs.


Copies will be available until December 1, 1977.
Development of the Centrifugal Compressor for the Nissan Vehicular Gas Turbine

K. NAKANISHI T. WATANABE S. YAMAZAKI

NOMENCLATURE

G = corrected airflow, kg/sec
U1 = circumferential velocity of inducer tip, m/sec
Clm = meridional flow velocity at inducer tip, m/sec
W1 = relative flow velocity at inducer tip, m/sec
W1' = local flow velocity at inducer tip, m/sec
M1 = relative Mach number at inducer tip
U2 = circumferential velocity of impeller tip, m/sec
C0 = absolute flow velocity at impeller exit, m/sec
C2m = meridional flow velocity at impeller exit, m/sec
C2d = circumferential flow velocity at impeller exit, m/sec
W2 = relative flow velocity at impeller exit, m/sec
M2 = absolute Mach number at impeller exit
\eta = compressor adiabatic efficiency, percent
Cf = friction factor
\Delta Pt = total pressure loss through diffuser, kg/m²
PD = dynamic pressure at impeller exit, kg/m²
B1 = inducer throat blockage, percent
Bp = diffuser throat blockage, percent
b = blade height at impeller exit, mm
t = axial clearance at impeller exit, mm
\beta p = pressure surface angle of diffuser vane, deg
\beta s = suction surface angle of diffuser vane, deg
\beta = centerline angle of diffuser vane, deg

INTRODUCTION

Since 1963, we have been engaged in the survey and development of a two-shaft regenerative gas turbine for heavy-duty vehicles. Experimental gas turbine engine No. 1 was completed in 1968, and went through four subsequent redesign stages to perfect its mechanism; continual efforts have been made to improve performance through better components built in. Along with this, gas turbine buses have been road tested to provide important practical information.

As general descriptions of the YTP12 versions and their vehicular applications have already been given in footnotes 1 and 2, this paper will concern itself with describing the improvement process of the centrifugal compressor of the YTP12.

Although neither the initial design specifications nor the present performance status can be accepted as the component of the gas turbine to compete with the diesel engine in fuel economy, the development process has been a difficult task for us whose experience with turbomachines was so limited.

The compressor development program was started just after the dimensions of the YTP12 were first specified. During the development program, many requirements were forwarded from the results of road tests on gas turbine buses as well as from bench tests of engines, and the course of compressor improvement has been pushed forward by demands stemming from the engine and experience accumulated year by year.

DESIGN

In any gas turbine, the most important factor is the proper matching of the compressor and

turbine.

To begin with, the compressor of a two-shaft vehicular gas turbine with variable geometry power turbine nozzle must operate over a wide flow range, from 50 to 100 percent speed, because the variable nozzle opening is controlled to maintain the gas temperature entering the gasifier turbine at the rated value even under partial load. Fig. 1 shows the compressor operating lines corresponding to each nozzle opening.

In addition, as the gasifier should be accelerated without excessive delay, a sufficient surge margin between 50 percent speed (idling) and 100 percent speed is necessary.

In the YTP12, as in most vehicular gas turbines, a regenerative rather than a recuperative heat exchanger is used. When a regenerator is used, there are many unsolved problems with respect to the air sealing mechanism. The variation in air leakage rate at the regenerator changes the matching condition of the compressor and turbine.

Axial turbines are choked at their nozzles, and it is difficult to adjust the rated airflow through them because nozzles—for reasons of cost—are one-piece castings.

Thus, the compressor must compensate for the variation in air leakage rate in the heat exchanger.

The specifications of the YTP12 and the data on compressor design are listed in Tables 1 and 2. Fig. 2 shows the design point velocity triangles.

**Impeller**

The compressor of a vehicular gas turbine must have a low manufacturing cost and high durability against vibratory resonance.

Therefore, the impeller was manufactured of high strength aluminum alloy as an integral precision casting.

For reasons of strength, the use of thin inducer blades was avoided, but flow path blockage was minimized by means of a splitter blade design.

Unfortunately, we could make no practical plans to implement the backward leaning blade design at the impeller exit in spite of the strong demand for wide flow range operation, because at the time the program was starting up, we could not get casting material of the extremely high strength necessary for the backward leaning blade design.

Fig. 3 shows the dimensional configuration of the impeller.

**Diffuser**

As for the diffuser, 13 straight wedge-like vanes were used to bring manufacturing costs down and to keep the number of parameters affecting the performance of the diffuser at a minimum for quick development.
Table 1 Specifications of the YTP12 at 15°C, 760 mm Hg

Maximum power available .......... 272 ps/3400 rpm
Turbine inlet temperature .......... 920°C (850°C)\textsuperscript{a}
Gasifier rotational speed .......... 40,000 rpm
Specific fuel consumption .......... 245 gr/ps/hr

\textsuperscript{a} The figure in parentheses ( ) indicates the initial target.

Table 2 Design Specifications of the Compressor

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Airflow rate</td>
<td>2.0 kg/sec</td>
</tr>
<tr>
<td>Revolution speed</td>
<td>40,000 rpm</td>
</tr>
<tr>
<td>Pressure ratio total-static</td>
<td>4.0 (3.8)\textsuperscript{a}</td>
</tr>
<tr>
<td>Inducer outer diameter</td>
<td>40 mm</td>
</tr>
<tr>
<td>Inducer inner diameter</td>
<td>21.1 mm</td>
</tr>
<tr>
<td>Impeller tip diameter</td>
<td>250.6 mm</td>
</tr>
<tr>
<td>Diffuser inlet diameter</td>
<td>500 mm</td>
</tr>
<tr>
<td>Diffuser exit diameter</td>
<td>941 mm</td>
</tr>
<tr>
<td>The number of impeller blades</td>
<td>30 (including 15 splitter blades)</td>
</tr>
<tr>
<td>The number of diffuser vanes</td>
<td>13</td>
</tr>
</tbody>
</table>

\textsuperscript{a} The figure in parentheses ( ) indicates the initial target.

Fig. 3 Compressor dimensional configuration

DEVELOPMENT

Test Apparatus

1. Inlet air pressure was reduced by the use of a multi-hole baffling plate in the suction line so that the compressor could be driven by a d-c 440-kw dynamometer with 11 to 1 step-up gears.

2. Airflow rate was measured by means of a metering orifice in the suction line.

3. Surge was detected at high speeds by audible sound and at low speeds by a microphone installed near the inlet bell-mouth.

4. Each diffuser vane was pivoted near the leading edge so the throat area could be easily altered (Fig. 4). Vanes were omitted when the characteristics of the impeller alone were measured. Later, the exit of the impeller was directly opened to a large collector to avoid stall in the vaneless diffuser.

Fig. 4 View of compressor

Process

The performance data from the series of experiments were found to be rather disappointing in that the pressure ratio was 3.53 and the flow range at the rated speed was 70 percent of the predicted range, though the surging flow rate agreed well with the prediction.

Thus, began the effort to raise the performance up to the design target.

As the progress of the engine test and the road test by buses required a much higher compressor performance, we decided to follow a development course consisting of repeated flow rate-pressure ratio measurements followed by macroscopic analysis and additional modifications to the basic compressor, instead of following the development path of designing and testing different
compressors or analyzing detailed flow phenomena for further improvement.

The following code is used throughout this paper to indicate the configuration of the compressor.

<table>
<thead>
<tr>
<th>Code</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>OGL or OLA</td>
<td>the outer diameter of the inducer, being 138 or 142 mm</td>
</tr>
<tr>
<td>91 or 85</td>
<td>the blade height of the impeller exit and the diffuser, being 9.1 or 8.5 mm</td>
</tr>
<tr>
<td>NC or 6C</td>
<td>the axial length cut from the inducer blades, being 0 mm (no cut) or 6 mm</td>
</tr>
<tr>
<td>26 or 28</td>
<td>the width of the diffuser throat, being 26 mm or 28 mm</td>
</tr>
</tbody>
</table>

Thus, OLA-91-NC (26) means that the inducer's outer diameter is 142 mm with no cut inducer blades, the blade height of the impeller is 9.1 mm and the width of the diffuser throat is 26 mm.

Surface Finishing

Measurement of the surface roughness of the cast inducer blade and machined vaneless and vaned diffuser showed a maximum roughness much greater than the average thickness of the laminar sublayer; this was considered to be the cause of the inadequate pressure ratio and choking flow rate.

Thus, we polished both the inducer blade surface and the diffuser surface with a buffer down to a maximum roughness of $4\mu$. As a result, appreciable improvement was attained.

Diffuserless-Test Rig for Impeller

Experimentation with the use of a vaneless diffuser suggested that the inadequate performance of the compressor was mainly due to mismatching of the impeller and diffuser.

Although knowledge of the characteristics of impeller alone is the key to correct matching of the impeller and diffuser, the experiment did not produce sufficient data about the flow range of the impeller due to the stalling of the vaneless diffuser.

Therefore, we modified the rig, removing the sidewalls composing the vaneless diffuser, and let the discharged air from the impeller flow directly into the collector. Fig. 5 shows a comparison of the two test rigs. The use of the "diffuserless-test rig" resulted in a better understanding of the impeller characteristics and contributed to a better matching of the impeller and diffuser.

Fig. 6 shows a comparison of the characteristics of an impeller measured on both test rigs.

The Flow Range of the Impeller

The target for the rated airflow was set at between 2.0 to 2.1 kg/sec, 5 percent away from the surging flow limit. The value of 5 percent was taken from the experience gained in engine tests.

Therefore, in consideration of manufacturing accuracy, the target for the surging flow limit of the impeller alone was set at just under 2.0 kg/sec.

Efforts were also made to achieve a wider impeller flow range so that a wider working range for the diffuser might be effected.

![Fig. 5 Impeller test rigs](A) Test rig with diffuser walls (B) Diffuserless-test rig)

![Fig. 6 Characteristics of an impeller measured on both test rigs](chart)
Although choking usually occurred at the inducer throat, some impellers, which had the same inducer dimensions (OLA-85-6C and OLA-91-6C), were found to have different maximum flow rates, as is seen in Fig. 7. Other impellers, which had different inducer dimensions and the same blade height at the impeller exit (GLA-85-NC and GLA-85-6C), had the same maximum flow rate. Thus, it was uncertain whether choking would always occur at the inducer throat.

Concerning the surging flow rate, some of the impellers, which had the same inducer dimensions but a different blade height at the impeller exit (GLA-85-NC and GLA-91-NC, GLA-85-6C and GLA-91-6C), had a different surging flow limit, most likely due to the difference in the diffusion factor.

Matching of the Impeller and the Diffuser

We successively tested many combinations of impellers and diffusers, in the order of from top to bottom in Fig. 7 (all combinations are not shown).

Through experiments, efforts had been made to achieve correct matching by causing the surging flow limit of the diffuser to coincide with that of the impeller. The difference between them meant not only an inadequate pressure ratio, but also too narrow a compressor flow range [GL-85-6C(28)] because the gradient of the pressure ratio versus the airflow rate was falling on the right-hand side.

Making the two surging flow limits coincide, however, was not easily achieved because the surging flow limit of the diffuser was affected by the individual impeller which showed an individual pressure ratio and flow distortion at the exit.

Now, GLA-85-6C(26) and GLA-91-6C(26) showed a nominal airflow rate which fell within the targeted range and also a nominal pressure ratio higher than the initial target; moreover, the lightening of the blade loading on the impeller resulted in GLA-91-NC(26) having the nominal pressure ratio of 4.96 (the adiabatic efficiency of 80.0 percent).

The performance of GLA-91-NC(26) is presented in Fig. 8.

CONSIDERATIONS OF TEST RESULTS

Influence of Surface Roughness

For the test Reynolds number, the smooth limit of the maximum surface roughness was approximately 3µ and the friction factor, \( C_f \), was 0.0026. On the other hand, the measured maximum roughness was 20µ, both at the inducer blade and the diffuser. Then, both the surface of the inducer blade and diffuser were smoothed down to 4µ in consideration of manufacturing cost.

The data of \( C_f \) for the diffuser are as follows:

<table>
<thead>
<tr>
<th>Targeted Range of Nominal Air Flow Rate</th>
</tr>
</thead>
<tbody>
<tr>
<td>GL 91 NC</td>
</tr>
<tr>
<td>GL 85 6C</td>
</tr>
<tr>
<td>GL 85 6C(28)</td>
</tr>
<tr>
<td>GL 85 NC</td>
</tr>
<tr>
<td>GL 85 NC(28)</td>
</tr>
<tr>
<td>GL 85 6C</td>
</tr>
<tr>
<td>GL 85 6C(28)</td>
</tr>
<tr>
<td>GL 91 6C</td>
</tr>
<tr>
<td>GL 91 6C(28)</td>
</tr>
<tr>
<td>GL 91 NC</td>
</tr>
<tr>
<td>GL 91 NC(28)</td>
</tr>
</tbody>
</table>

Fig. 7 Matching of the impeller and diffuser.

Fig. 8 Characteristics of GLA-91-NC(26)
As a result of surface finishing of both the inducer blade and the diffuser, a stage efficiency rise of 4.4 percent was attained.

The total pressure loss through the vaneless and vaned diffuser was calculated as follows:

\[ \Delta P_{t/d} \text{ (impeller exit), } 20 \mu = 0.52 \]
\[ \Delta P_{t/d} \text{ (impeller exit), } 4 \mu = 0.42 \]

Although the relation between \( C_f \) and \( \Delta P_{t/d} \) could not be described as proportional, it could be said that \( C_f \) was the key factor in estimating \( \Delta P_{t/d} \).

Stall and Choke of the Impeller

**Stall.** From Fig. 9, it can be seen that the lower \( M_{r_1} \), or revolution speed, the greater the stalling incidence. Fig. 10 shows a larger \( W_2/W_1 \) which corresponds to a larger incidence angle at lower \( M_{r_1} \).

The reason for the larger \( W_2/W_1 \) is considered to be as follows:

The flow velocity on the suction surface of the inducer blade becomes high (\( W_1^* \)) locally as the incidence increases. Thus, it is inevitable that \( W_2/W_1 \) will become larger than 0.54 in order to keep \( W_2/W_1^* \) at a level above 0.54. Then, the correlation between the margin, \( W_1^*/W_1 \), and the incidence can be made for the individual impeller, as shown in Fig. 11.

The reason why a larger incidence angle is permissible for a small \( M_{r_1} \) is that \( W_2/G \) at lower speed is larger than \( W_2/G \) at higher speed due to the lower pressure ratio at the impeller exit, while \( W_1/G \) differs hardly at all.

**Choke.** Fig. 12 shows, assuming the choked condition at the rms radius, the blockages at the inducer throat when several compressors operate at their maximum flow conditions.

GL-91-NC, GLA-85-NC, and GLA-91-NC have the same choking blockage of 5.5 percent, while GL-85-6C, GLA-85-6C, and GLA-91-6C each have a blockage larger than 5.5 percent.

As the maximum flow of the impeller changes only when the blade height at the impeller exit

\[ C_f (20 \mu) = 0.0041 \]
\[ C_f (4 \mu) = 0.0029 \]
\[ C_f (3 \mu) = 0.0026 \]
is changed at the impeller with cut-off inducer (-6C), we surmise that the maximum flow rate is regulated not only at the inducer throat but, in some cases, somewhere downstream near the exit.

**Stall and Choke of the Diffuser**

**Stall.** The airflow angle at the vaneless diffuser inlet was calculated assuming a uniform flow when the compressor was running at its minimum flow condition. The correlation between the calculated flow angle and the revolutional speed is indicated in Fig. 13. The flow angle is approximately 22.5° and rarely affected by the revolutional speed.

Therefore, it is understood that diffuser stall occurs when the flow angle coincides with the centerline angle of the leading edge of the diffuser vane. This result agrees with the experimental results obtained in a study made by Dr. Senoo's group, wherein it was demonstrated that diffuser stall occurs when the measured pressures at both sides of the diffuser blade near the leading edge become identical.

**Choke.** The blockage at the diffuser throat in the choked condition was calculated under the assumption that due to friction loss, the total pressure at the throat was lower than the total pressure entering the vaneless diffuser. The result is indicated in Fig. 14, where the abscissa is the inlet Mach number at the vaneless diffuser.

According to Fig. 14, the blockage increases as the inlet Mach number exceeds 1.0. The explanation posited is that the shock wave generated by the transonic flow condition causes the increased blockage.

**Effect of Running Clearances**

Fig. 15 shows the correlation between the ratio of the axial running clearance to the blade height at the impeller exit and the deterioration of compressor efficiency.

This result is similar to the data correlations by Pampreen and Dean.3 It can be seen that the ratio of the work factor decrease and the deterioration of compressor efficiency becomes larger when the relative clearance increases.

**CONCLUSIONS**

1 A considerable performance improvement and flow range adjustment were achieved through methods which had no adverse effects on the rest of the engine; those methods included modification of inducer outer diameter, change in blade height at the impeller exit and diffuser throat, and cutting-off of the inducer blades.

2 The characteristics of the impeller alone were better understood from use of the diffuserless impeller test rig, resulting in easier and better matching of impeller and diffuser.

---

3. The stage efficiency can be improved if the surface roughness at the inducer and at the entry region of the diffuser is reduced down to hydraulic smoothness, to the maximum roughness of 3µ. Therefore, grinding or the use of a buffer is an absolute must so as not to lose efficiency due to additional friction, even though, in consideration of manufacturing cost, 4µ is the limit.

4. We considered the minimum stalling diffusion factor, \( \frac{W_2}{W_1} \), as 0.54. The inducer had a greater stalling incidence at lower speed than at higher speed due to larger volumetric flow at the impeller exit. At low speed, however, high velocity was locally generated on the inducer suction surface because of a large stalling incidence. Accordingly, the apparent diffusion factor must be greater than 0.54.

5. Impellers are usually choked at their inducer throat. But our experiments raised some doubts, indicating that, in some cases, the maximum flow limit may be regulated somewhere downstream near the exit.

Concerning the impeller choked at the inducer throat, the blockage was approximately 5.5 percent.

6. The blockage of the diffuser throat in the choked condition was within the range of 1 to 10 percent.

This blockage increased as the inlet Mach number at the vaneless diffuser increased.

7. It was indicated that diffuser stall occurred when the inlet airflow angle at the vaneless diffuser reached the centerline angle of the vane.

8. There could be a correlation between the deterioration of stage efficiency and the relative running clearance. It was also shown that the work factor rapidly decreased when the relative clearance exceeded 0.06.

ACKNOWLEDGMENTS

The authors wish to express their thanks to Dr. Yasutoshi Senoo for his consistent guidance and encouragement in the design and development of the compressor.

They would also like to thank Nissan Motor Co., Ltd. for permission to publish this paper.

Finally, the authors wish to express their appreciation to Y. Nomura and S. Ohtaki for their experimental work.