ABSTRACT

Based on the specifications of a 100 kW single shaft automotive CGT, the aerodynamic design and the basic structure design of the radial turbine components, and the detailed design of each part were carried out. When designing, attaching great importance to the problems (brittleness and difficulty in manufacturing) unique to ceramic materials is required. The life was estimated with a strength reliability designing tool.

The aerodynamic characteristics were evaluated by testing metallic models, and the level of the intermediate target efficiency was achieved. The turbine rotor and the stationary structure parts were manufactured, and the strength reliability was evaluated with single and combined parts respectively. A peripheral speed of 830 m/s (113% of the rated value) with a turbine inlet temperature of 1200°C was obtained for the rotor hot-spin burst test.

The aerodynamic characteristics and the strength reliability shall be improved by repeating trial manufacture and test evaluation of ceramic parts.

INTRODUCTION

The Petroleum Energy Center in Japan is furthering the "Development of a 100 kW Automotive Ceramic Gas Turbine (CGT)" , a project supported by the Agency of Natural Resources and Energy, the Ministry of International Trade and Industry. This seven-year project, started in 1990, is aimed at designing and manufacturing a 100 kW automotive CGT featuring multi-fuel adaptability, excellent exhaust gas emissions and thermal efficiency of not less than 40%, and proving its superior potential by means of bench tests. This project does not include vehicle tests but focuses development activities on evaluating basic characteristics of the automotive CGT only by bench tests. The CGT engine to be developed is a single shaft type consisting of a centrifugal compressor, a radial turbine, a can type combustor and two regenerative rotary heat exchangers.

The pressure ratio is five at the rated speed and the maximum temperature at the turbine inlet is 1350°C (Fig. 1)

This paper outlines the aerodynamic and structural design of the radial turbine components, the design and trial manufacture of main component parts and the results of their test evaluation, all of which had been carried out by the end of FY1991.

AERODYNAMIC AND BASIC STRUCTURE DESIGN

Basic Design Specifications

In setting the basic design specifications of the turbine, important factors include raising the stress analysis accuracy...
and easing manufacture by making the parts smaller and simpler than those of conventional metallic turbines. To achieve this, the whole stationary structure section should be made compact, while keeping symmetry with respect to the axis, by making the rotor as small as aerodynamically possible. The rotor also should be shaped to sustain impact with foreign matter. Taking above mentioned conditions into consideration, the basic design set the specific speed higher while balancing the turbine with the compressor aerodynamically. Table 1 shows the design parameters and target efficiencies at the rated design point and at a typical point of partial load, that is, 80% rpm. Being for automotive use, it is important to attain a high efficiency under a partial load.

<table>
<thead>
<tr>
<th>Rotational Speed rpm (%)</th>
<th>110,000 (100)</th>
<th>88,000 (80)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas Flow (kg/s)</td>
<td>0.421</td>
<td>0.257</td>
</tr>
<tr>
<td>Inlet Pressure (kg/cm²)</td>
<td>4.87</td>
<td>3.01</td>
</tr>
<tr>
<td>Inlet Temp (°C)</td>
<td>1350</td>
<td>1350</td>
</tr>
<tr>
<td>Expansion Ratio</td>
<td>4.25</td>
<td>2.75</td>
</tr>
<tr>
<td>Efficiency (%)</td>
<td>87.5</td>
<td>87</td>
</tr>
<tr>
<td>Pressure Loss (%)</td>
<td>3.6</td>
<td>3.6</td>
</tr>
</tbody>
</table>

Table 1  Turbine Stage Design Parameters

Turbine Aerodynamic Design

Figs.2 and 3 show the basic configuration of the rotor and the nozzle and their velocity triangles. Although the partial load performance is better when the theoretical velocity ratio (Tip peripheral velocity/Theoretical expansion velocity) is high, the tip velocity was set to about 700 m/s at the rated point because of the strength limitation. The velocity triangles of the turbine rotor outlet at the mean effective diameter are shown. Their basic shapes were determined so that the outlet absolute velocity was as low as possible, while the swirl angle was also kept within a range that can maintain a high pressure recovery coefficient at the downstream diffuser section of the rotor.

The blade curvature, number of blades and blade thickness distribution were studied so that a smooth velocity change and uniform work distribution at the blade surfaces, and a uniform distribution with a small flow change at the blade outlet, were obtained through three-dimensional analysis of the flow between blades. Fig.4 shows the velocity changes along the mean line between the hub and the shroud and the flow distributions at the blade surfaces. Although uniform acceleration of the flow between the blades is desired, the flow tends to decelerate at the suction surface when it is nearer the hub side at an intermediate section in the flow direction, where the velocity at the suction surface tends to be the same as that at the pressure surface, or even reversed. At the pressure surface, the flow tends to greatly decelerate immediately after the rotor inlet, especially at the hub side. It is necessary to decrease the loss due to the secondary flow (swirl) at the intermediate section and the one due to the wake at the outlet by minimizing the amount of deceleration mentioned above. Making a trade-off with the three-dimensional stress analysis (mentioned later) a 14-blade structure was adopted and the blade shape was determined in detail.

The profile of the nozzle vanes was determined according to the basic profile which had actual results, and the trailing edge thickness and others were determined according to the requirements of manufacturing the ceramic part. After examination to satisfy the discharge characteristics and to make the flow distribution most suitable by analysis, a 21-blade structure was adopted. (Fig.5)
Rotor Inlet

\[
\begin{align*}
C : & \text{Absolute Velocity} \\
W : & \text{Relative Velocity} \\
U : & \text{Peripheral Velocity} \\
\end{align*}
\]

Rotor Outlet

\[
\begin{align*}
9' & \quad 17' \\
\end{align*}
\]

Fig. 2 Velocity Triangles of Turbine Rotor

Fig. 3 Turbine Basic Configuration

Meridian Distance
Relative Velocity at Mean Diameter

S.S.: Suction Surface
P.S.: Pressure Surface

Fig. 4 Three-Dimensional Flow Analysis in Rotor Blade Passage

Structure System Design

Since there is no stress relaxation effect such as plastic deformation seen in metallic materials, this design attempted to eliminate excessive stress concentrations and three-dimensional stress peak which is difficult to estimate accurately, and also attached great importance to manufacturing considerations. Fig. 6 shows the design principle and Fig. 7 shows the entire construction. The stacking structure shown in Fig. 7 consists of two systems, the scroll and its inner and outer support system, and the inner shroud, nozzle and roller system, and receives an elastic pressing force separately from the two balance pistons, each of which generates a load by a spring force and a pressure difference. Taking the molding characteristic into consideration, the scroll was divided in two radially.

To simplify stress evaluation and manufacture, the scroll section was kept simple despite aerodynamic drawbacks, while the flow turning section just upstream from the nozzle was kept symmetric with respect to the axis. Fig. 8 shows the scroll. Further, to lower the hoop thermal stress level at the end of the scroll inlet, the outlet section of the combustor was inserted into the downstream side.

Fig. 5 Flow Distribution in Nozzle Vane Passage

Fig. 6 Design Principle of Stationary Ceramic Structure Parts
at the radial key section. To reduce the stacking tolerance, the parts related to the centering were limited to the roller, nozzle and inner shroud.

The nozzle was divided into six segments. A solid-block nozzle ring was not adopted in this design structure because it is difficult to mold, and generates an excessive stress when axial load is not ideally symmetric and a local load is applied. The segment nozzle can reduce both the hoop stress due to radial temperature differences in each side plate when starting and stopping, and the vane stress due to the deformation of the side plates.

DESIGN OF STRENGTH RELIABILITY

Design conditions and materials

The final goal was to satisfy a failure probability of $10^{-5}$ under 300-hours of continuous running at maximum power, and 10,000 engine starts from cold state. This assumed the severest use conditions of a passenger car. Although there is no rapid drop in gas temperature when the engine comes to a stop, because of heat accumulation in the heat exchanger, the heat exchanger characteristics and the stop conditions shall be the subject for a future study. The strength reliability of the rotor depends upon both the rated continuous running condition and the cold starting condition, while that of the stationary structure parts is dominated by the cold starting condition. Therefore, these cases were considered in the design. Fig.9 shows the patterns of the turbine inlet temperature and the rpm which were set as the cold starting condition.

Silicon nitride was used for each component part because of its high temperature strength and toughness. Table 2 shows the materials used. Table 3 shows the properties of the material for the rotor. The rotors, the most important parts, were provided by three separate material manufacturers. A single material, with high strength up to 1400 °C, was adopted for the nozzle, inner shroud, and outer shroud. For the inner scroll, outer scroll, and support, a material with actual results in slip casting suitable for the large shell parts was used. For other parts such as the back plate, a material that has the highest strength at not more than 1200°C was adopted.
As a result of the reliability analysis, the failure probability under the rated 300-hour continuous running with the static fatigue life strength of the present material is 1/80. This failure probability was calculated from the stress distribution (Fig. 10), 300 hr fatigue strength 600 MPa and Weibull modulus 20 (Table 3). To satisfy the final failure probability of 10^-5, a 860 MPa level is required for the 300-hour static fatigue mean strength, which shall be the target value for improving the material strength. (Table 3)

Design of rotor

Fig. 10 shows the temperature distribution and the stress distribution at the rated condition. The maximum temperature is 1070°C at the outside tip of the blade, while the temperature at the blade root and the hub core, where the maximum stress is produced, is 850-900°C. The maximum stress is mainly a centrifugal stress of 293 MPa, but the hub core, having a wide range of high stress, dominates the lifetime. The maximum stress when cold starting is produced in the blade root after 25 seconds, and in the hub core after 40 seconds. Fig. 11 shows the temperature and the stress distribution at 25 seconds after starting.

As a result of the reliability analysis, the failure probability under the rated 300-hour continuous running with the static fatigue life strength of the present material is 1/80. This failure probability was calculated from the stress distribution (Fig. 10), 300 hr fatigue strength 600 MPa and Weibull modulus 20 (Table 3). To satisfy the final failure probability of 10^-5, a 860 MPa level is required for the 300-hour static fatigue mean strength, which shall be the target value for improving the material strength. (Table 3)
Design of stationary structure parts.

The detailed shape and plate thickness of each part was determined after analyzing and evaluating the transient thermal stress produced when cold starting, and also reflecting the molding requirements. The maximum stress produced had to allow for 10,000 cold starts and a failure probability of $10^{-5}$. The analysis results of the main parts are shown below:

- **Nozzle**: A solid-block nozzle, which produces higher thermal stress than segmented nozzles, was evaluated, and a maximum stress of 140 MPa was produced in the nozzle trailing edge near the side plate 10 seconds after starting (Fig.12).

- **Inner shroud**: A maximum stress of 168 MPa was produced in the surface attaching the nozzle slide plate 28 seconds after starting (Fig.13). The stress produced under the rated steady condition was 50 MPa.

- **Outer scroll**: The maximum stress of 209 MPa was produced in the inserted end of the combustor, where the temperature changes rapidly due to combustion gas, 100 seconds after starting (Fig. 14).

- **Inner scroll**: The maximum stress of 106 MPa was produced in the flange circumference 80 seconds after starting (Fig.15). The stress produced under the rated steady condition was not more than 10 MPa.
EVALUATION TEST ON AERODYNAMIC PERFORMANCE

To evaluate the aerodynamic performance prior to the trial manufacture of ceramic parts, a metallic rotor and nozzle were used to evaluate under similar aerodynamic conditions through low temperature air. Three kinds of nozzles, those complying with the basic design specifications and those with different vane angles and throat areas were tested. The passages of the scroll and so on were all made of metal. Fig.16 shows the test results of the efficiency to the corrected mass flow at rated pressure ratio when changing the rotational speed. The efficiency has reached the intermediate target value of 86%, but the greater the flow, the higher the peak efficiency. Further, the flow characteristic with the standard nozzle is shifted to the greater flow side. The degree of reaction of the rotor, the nozzle throat area and so on will be further studied to meet the required flow characteristic and to improve efficiency.

MANUFACTURE AND EVALUATION TEST OF CERAMIC PARTS

Manufacture of ceramic parts

Table 2 shows the molding method for each part. As mentioned before, three types of materials were used for the rotor, and the shape of the blade trailing edge or a part of the blade thickness distribution was changed upon proposals made partly because the respective manufacturing methods are different. Fig.17 shows the rotor, inner shroud and outer scroll.
Evaluation test on strength reliability

Fig. 18 shows the flowchart of the strength reliability evaluation for the rotor and the stationary parts. Each part is not necessarily required to follow all the evaluation steps shown in the figure, but steps can be selected according to the working purpose or conditions. The evaluation of the test pieces processed together with the actual parts is done to check the manufacturing process and to obtain evidence without destroying the actual parts. Regarding the stationary parts, however, it is possible to evaluate by cutting out a portion from an actual part. The evaluation of the test piece cut out from the actual parts is to set the manufacturing process and the material specification and to evaluate the strength dispersion in the real body, and is carried out in the early stages of the manufacture or by random sampling.

Regarding the large shell structure parts among the stationary parts, since there is a fear of residual stress or local defect of the material depending on manufacturing process such as sintering and machining, it is necessary to evaluate the soundness of parts where the high stress is produced. Therefore, proving tests were carried out by applying a load or pressure. Fig.19 shows an evaluation example of the inner shroud. Some large shell parts failed in these proof tests and the manufacturing process was improved to pass the tests.

As to the stationary parts, heat shock when cold starting and durability when running continuously under a maintained high temperature condition were evaluated. The nozzle itself was tested for cycle durability by letting high temperature gas and room temperature air through the nozzle alternately in a stepped pattern. Also, by letting high temperature gas through the stationary structure parts assembled as whole, heat shock and continuous durability are being evaluated on a stationary section combination tester (Fig.20).

In the strength reliability evaluation of the actual rotors, strength levels and dispersions in cold and hot states are obtained by burst tests under cold spin and hot spin, and correspondence to the cut out test piece is also studied. The results are fed back to the material, manufacturing process and to the design and analytical method. Fig.21 shows the results of the burst tests under cold spin and hot spin with three materials. Although the feature of each material is shown, the correlation between both data or with the data of the test pieces cut out from the real body shall be the subject of a future study since there remains unknown points due to lack of data. However, a peripheral speed of 830 m/s with a turbine inlet gas temperature of 1200°C has been obtained. The life evaluation of the rotor by the durability tests through repeated start-and-stop and continuous running have already been started on a high temperature turbine tester combined with the nozzle (Fig.22).
Fig. 20  Stationary Section Combination Tester and Sub-assembly of the Stationary Parts

Fig. 21  Cold & Hot Spin Test Results

Fig. 22  High Temperature Turbine Tester
CONCLUSION

(1) Tests are being carried out to evaluate the aerodynamic characteristics and strength reliability of radial turbine components of a 100 kW single-shaft type automotive CGT.
(2) In the evaluation test of aerodynamic characteristics with low temperature air, efficiencies at a level of the intermediate target value have been obtained. Discharge characteristics have been shifted from the target value to the greater discharge side.
(3) Heat shock and continuous durability tests are being carried out on each part of the stationary ceramic structure, and on the assembled structure at 1200 °C gas temperature.
(4) As to the ceramic rotor, the burst tests are being carried out under cold and hot spin. The maximum values of 124,700rpm and 830 m/s peripheral speed (113% of the rated speed) have been obtained at a turbine inlet temperature of 1200°C.
(5) Future improvements, to attain the target efficiency, will come from reexamining the aerodynamic design, while repeating trial manufacture and evaluation tests of the ceramic parts will produce the strength reliability that can withstand the engine test.

AKNOWLEDGEMENTS

The authors are grateful to the Agency of Natural Resources and Energy, the Ministry of International Trade and Industry, for making this development possible and to the program management at the Petroleum Energy Center. We would also like to thank members of the CGT organization in Japan Automobile Research Institute for technical supports.

Finally we would like to acknowledge the efforts of Kyocera Corp., NGK Insulators, Ltd. and NGK Spark Plug Co., Ltd. in manufacturing ceramic components.

REFERENCES