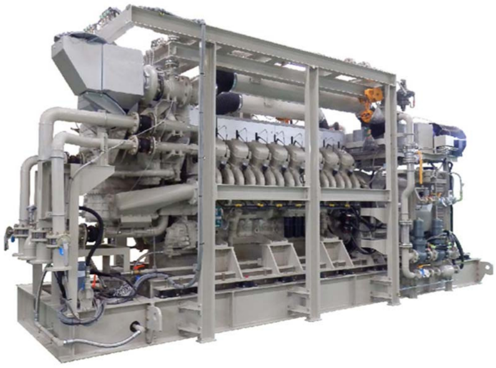
Development of High-efficiency Gas Engine with Two-stage Turbocharging System



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A new G16NB gas engine for generator use adopting a two-stage turbocharging system was developed for the improvement of electrical efficiency and output. Two-stage turbocharging, intake valve close timing and expansion ratio optimization, as well as an increased peak firing pressure, were adopted to achieve the development target of 10% (4% points) or higher efficiency improvement. The test engine operated by city gas (13A) displayed the results below: (1) Turbocharger efficiency was improved by 14% points compared with that of conventional models. In addition, the pressure ratio of compressor was increased by 1.5 contrasted with that of conventional models at the rated output. (2) The electrical efficiency of the engine was increased by 4% points, and attained the world's top level efficiency of 44.7% or higher. Additionally, the break mean effective pressure (BMEP) was raised by 0.5 MPa in contrast to that of the current model.

1. Introduction

Recently, gas engines for generator use are required to have both high efficiency and high power. The Miller cycle is well known as one of the means to achieve high efficiency while avoiding abnormal combustion conditions such as knocking. However, the Miller cycle inherently lowers the volumetric efficiency, and the output power is lowered at the same boost pressure. A turbocharger is required to ensure a higher pressure ratio and higher efficiency to compensate for the disadvantages of the Miller cycle.

Contrarily, a turbocharger tends to lower the turbocharging efficiency in proportion to the rise of the pressure ratio at a certain pressure ratio and above, and so a high pressure ratio and high turbocharging efficiency are a trade-off. One solution to achieve both a high pressure ratio and high turbocharger efficiency is two-stage turbocharging with intermediate cooling. In a two-stage turbocharging system, each pressure ratio at the lower and higher stages can be designed to be relatively low, and the high pressure ratio can be attained at the high-efficiency operational point of each stage. Moreover, the effect of intermediate cooling improves turbocharger efficiency.

In two-stage turbocharging, the pressure ratio of each stage can be set to be relatively low, as can the compression temperature after the compressor becomes low compared with that of single-stage turbocharger. This is advantageous in designing the compressor strength.

This report describes the development of the new G16NB high-efficiency gas engine adopting a two-stage turbocharging system. The development target of this engine is a higher efficiency of 10% (4% points) or higher than that of conventional models. In this development, two-stage turbocharging, the optimization of the inlet valve closing timing (IVC) and expansion ratio, and a higher peak firing pressure (higher Pmax) are adopted to achieve the desired results. The development of the two-stage turbocharging system and the design of the component parts of the combustion chamber are detailed in this report.

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2. Development of G16NB gas engine design

The main specifications of the G16NB engine are shown in **Table 1**. The output is increased from 1500kW to 2000kW with the same cylinder bore, stroke length, rated speed and cylinder numbers. The break mean effective pressure (BMEP), which refers to the load per cycle and unit displacement volume, is raised by 0.5MPa.

With these specifications, one dimensional performance simulation was conducted using the IVC, target turbocharger efficiency, the target compressor pressure ratio and the target peak firing pressure as parameters to clarify the conditions to achieve the target of efficiency improvement.

GT-Power V7.2 calculation software (Gamma Technologies Inc.) was used for the calculation investigation. Single-stage turbocharging with a turbocharger efficiency equivalent to two-stage turbocharging was calculated, although the actual engine uses two-stage turbocharging. In the calculation, the engine output, expansion ratio and excess-air ratio were set to be constant, and the ignition timing was adjusted to the knocking limit. BMEP was set to the increase target value by 0.5 MPa. The heat-receiving rate was estimated by extrapolation by conducting a sensitivity analysis of BMEP from the test results of the current model.

Table 1 G16NB main specifications

		Conventional engine GS16R2	Developmental engine G16NB
Cylinder bore	mm	170	170
Stroke	mm	220	220
Rated engine speed	min^{-1}	1500	1500
Cylinder number	—	16	16
Rated output (at generator end)	kW	1500	2000
Break mean effective pressure	MPa	1.6	2.1
Combustion type	—	Pre-chamber	Pre-chamber
Ignition system	—	Spark ignition	Spark ignition

2.1 Determination of inlet valve close timing, target turbocharger efficiency and target peak firing pressure

The calculation results of thermal efficiency, compressor pressure ratio and peak firing pressure (Pmax) to IVC are shown in **Figure 1**.

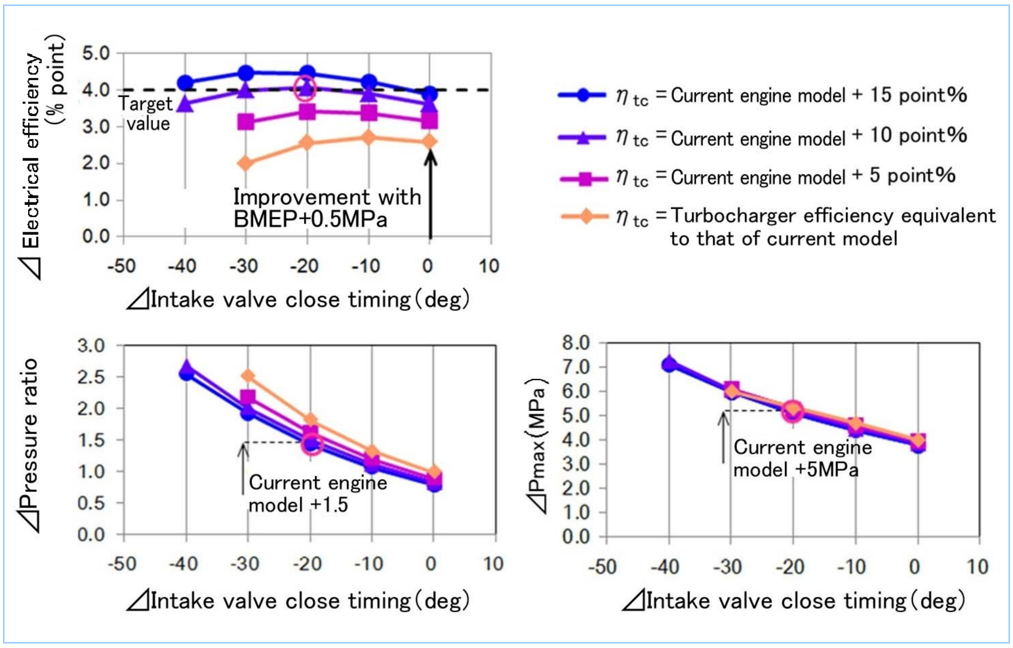

Figure 1 One-dimensional performance simulation

As the thermal efficiency calculation shown in the left upper position of Figure 1, the electrical efficiency improvement cannot be attained by advancing IVC in the case of current turbocharger efficiency (◆).

The advancement of IVC by 20 deg, the improvement of target turbocharger efficiency by

10% points from that of the current turbocharger, the increase of the compressor pressure ratio by 1.5 from the current model, and the increase of Pmax by 5MPa from that of the current model are required to achieve the target electrical efficiency.

2.2 Design of two-stage turbocharging system

The two-stage turbocharging system was selected to attain the target turbocharger efficiency and required pressure ratio clarified in the previous section, and the specifications were investigated.

2.2.1 Determination of parameters

In the target 10% points increase of turbocharger efficiency, 5% points were allotted to the improvement of single-stage turbocharger efficiency, and the remaining 5% points to the two-stage turbocharging system. An outline drawing of the two-stage turbocharging system is shown in **Figure 2**.

The parameters below are required to design the two-stage turbocharger system.

・Loading ratio of high-pressure compressor

・Intermediate cooling temperature

・Allowable pressure loss of intermediate cooling

Figure 3 shows the results of the parameter study to determine the items above.

The loading ratio of the high-pressure compressor was defined as the required loading ratio of the high-pressure compressor in the total required compressor loading. In the calculation, the total compressor pressure ratio including the low and high pressure stages was constant, and the single-stage compressor efficiency, turbine efficiency and mechanical efficiency were constant regardless of the condition.

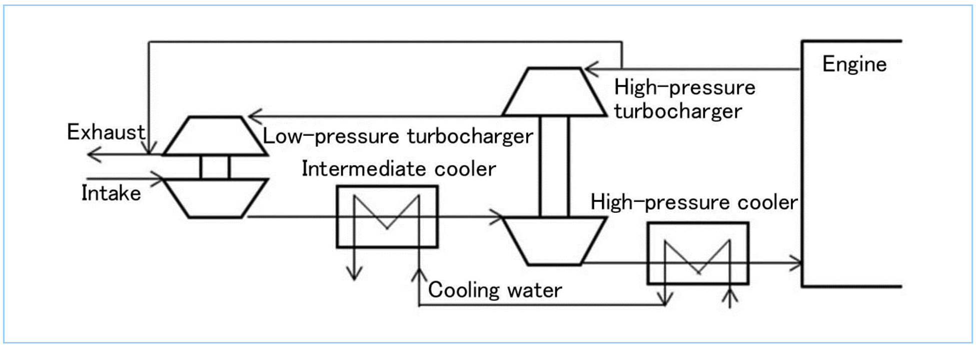

Figure 2 Outline drawing of two-stage turbocharging system

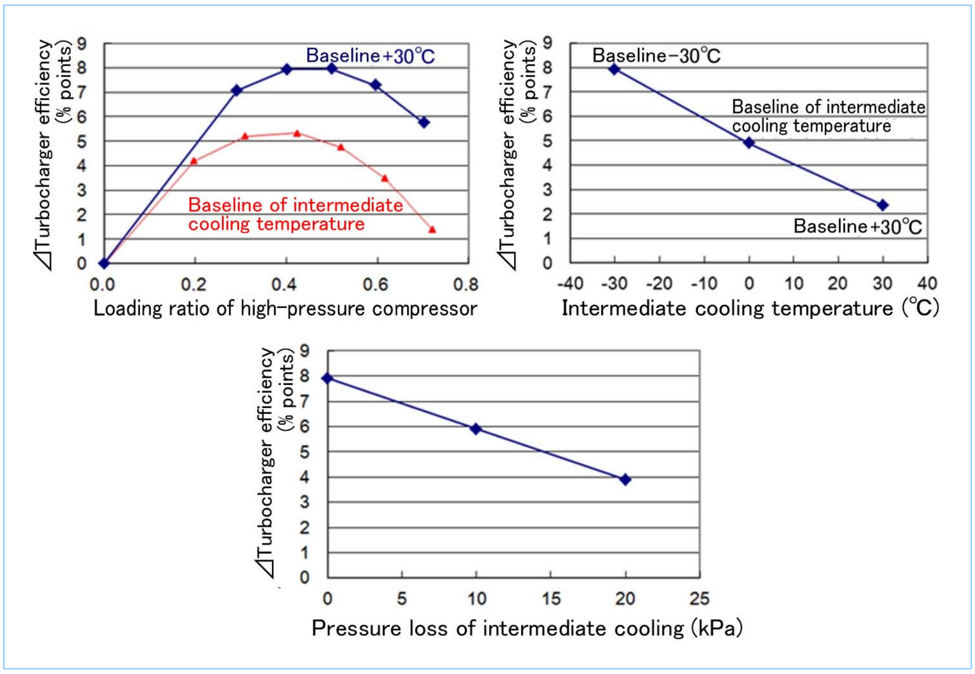

Figure 3 Parameter case study of two-stage turbocharger system

Figure 3 shows the results below.

・The optimum point of the loading ratio of the high pressure compressor differs depending on the intermediate cooling temperature. The approximate value of 0.5 shows the maximum turbocharger efficiency.

・Turbocharger efficiency change is almost linear to the intermediate cooling temperature, and is approximately 0.1% point/K.

・Turbocharger efficiency change is almost linear to the intermediate cooling pressure loss, and is approximately 0.2% point/KPa.

From the results above, the verification calculation was carried out with the design values below, and the prospect of satisfying the target turbocharger efficiency was achieved.

・High pressure compressor loading ratio: Approximately 0.5

・Intermediate cooling temperature: Lower than the baseline by 30° C or less

・Intermediate cooling pressure loss: Less than 10kPa

2.2.2 Determination of parameters

The ratio of the equivalent area was determined to satisfy the loading ratio of the high pressure compressor investigated in the previous section.

In addition, the high-pressure compressor pressure ratio and volumetric flow rate were calculated, and the high- and low-pressure stage compressors were designed. The compressor was designed to have the maximum efficiency near the operating pressure ratio when considering the low pressure ratio of each single stage. The turbocharger efficiency of a single stage was expected to rise by 5 % points contrasted with the current model averaging the low- and high-pressure stage compressors.

A design study of the intermediate cooler layout was conducted to satisfy the target temperatures of the high- and low-pressure stage turbochargers and the intermediate cooler. **Figure 4** shows the results of the pressure loss calculation.

The intermediate cooler pressure loss calculation was made in several layouts, and finally a layout which could satisfy the target value of less than 10kPa was determined.

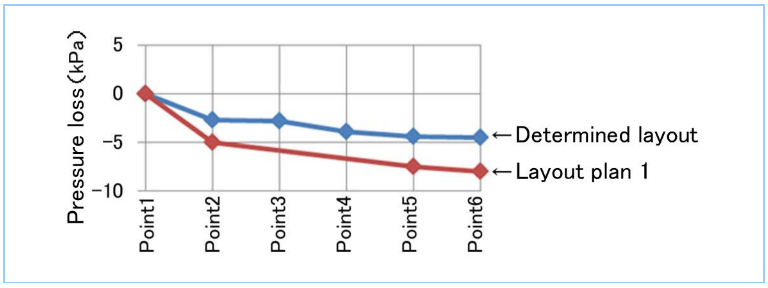

Figure 4 Pressure loss calculation

2.3 Design of combustion chamber

The flow analysis and flame propagation simulation in the pre-chamber was conducted to optimize the pre- and main combustion chambers. In addition, thermal structure analysis was carried out to design a combustion chamber that can be realized considering the mechanical strength under the high Pmax and BMEP conditions.

2.3.1 Flow analysis and flame propagation simulation in pre-chamber

The flow analysis in the pre-chamber was conducted to stabilize the combustion in the pre-chamber. **Figure 5** shows the methane concentration distribution at the timings of 180 deg and 90 deg before compression top dead center, and the ignition timing. Figure 5 shows the dilution of the supplied fuel gas by lean mixture flowing from the main chamber, and a uniform mixture formed at the ignition timing. The optimization of the pre-chamber shape was conducted using the gas trap rate (fuel gas amount remaining in the pre-chamber at the timing of ignition to the supplied fuel gas) and the flow speed around the ignition plug as the evaluation indicator.

The flame propagation simulation was carried out using the three-dimensional combustion analysis code for the design of the main chamber shape, and the heat-receiving rate, the gas temperature in the combustion chamber and the heat transfer coefficient were calculated. For the three-dimensional combustion analysis code, KIVA was used to solve 42 chemical species and 213 elementary reactions.

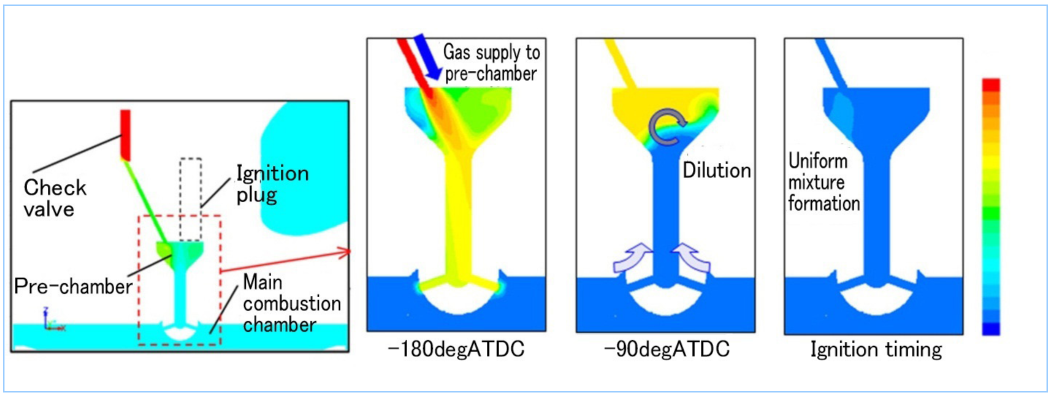

Figure 5 Flow analysis in the pre-chamber

The calculated heat-receiving rate and isothermal surface in the analysis are shown in **Figure 6**. The figure shows the isothermal surface of 1500°C from 5 deg before to 25 deg after the compression top dead center. The torch flame discharged from the pre-chamber and the flame surface propagation from the torch are shown.

The heat-receiving rate calculated in the analysis was fed back to the one dimensional performance simulation in section 2.1 to improve the accuracy of performance calculation, and the gas temperature in the combustion chamber and the heat transfer coefficient calculated in the three-dimensional combustion analysis were used as the thermal boundary conditions in the strength analysis of the piston, cylinder liner, and cylinder head.

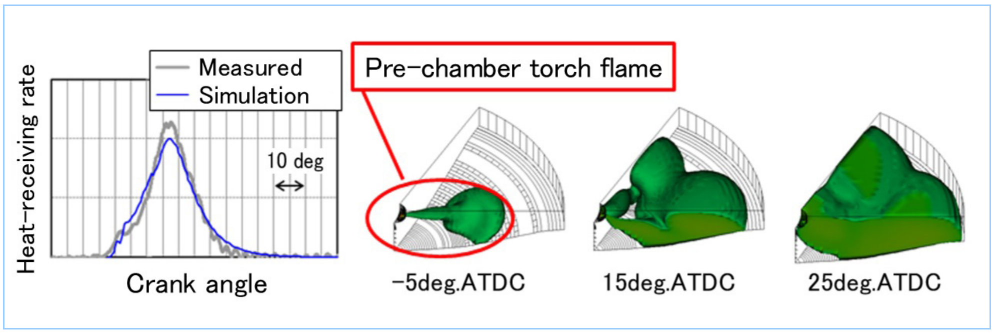

Figure 6 Flame propagation simulation

2.3.2 Thermal structure analysis of combustion chamber component parts

For the strength analysis of the combustion chamber component parts, the thermal boundary conditions in the combustion chamber were determined by the combustion analysis in the previous section, and the thermal boundary conditions on the cooling face were determined by the one dimensional flow network analysis. The MHI flow network analysis code was used for the flow network analysis of cooling system.

The flow channel cross section, flow channel circumferential length, and flow channel length were used as parameters, and the cooling water channel of the whole engine was calculated as a one dimensional model for the flow network analysis.

The flow network analysis can calculate the heat transfer coefficient on the main parts such as the cylinder liner and cylinder head, in addition to the flow speed calculation at each channel. As an example, the developed view of the heat transfer coefficient distribution around the cylinder liner and one dimensional model are shown in **Figure 7**.

The cooling water passage was designed to equalize the heat transfer coefficient in the circumferential direction at the upper part of the cylinder liner using the results of the analysis. FEM analysis was conducted using the determined thermal boundary conditions on the combustion chamber side and cooling water side, and the cylinder was designed to have sufficient fatigue strength.

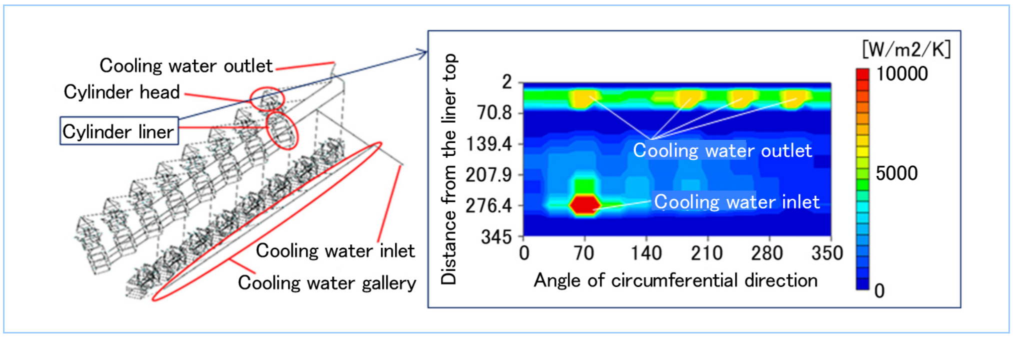

Figure 7 Flow network analysis of cooling system

3. Test results

A test engine equipped with a two-stage turbocharging system was built, and a performance test was conducted. The main measurement results are shown in **Table 2** and **Figure 8**. city gas (13A) was used for the fuel.

The results were sufficient to attain the target values of intermediate cooling temperature and pressure loss in the two-stage turbocharging system. Then, the target values were satisfied by improving the turbocharger efficiency by 14% points and the compressor pressure ratio by 1.5 in contrast to those of the current model at the rated operational point.

In addition, the relationship of the compressor pressure ratio and turbocharger efficiency at engine partial load was compared. As a result, it was found that the boost pressure rate of increase at engine starting is advantageous, and the startability is free from problems with the improvement of turbocharger efficiency from the low pressure ratio condition that is used during engine starting.

The electrical efficiency and BMEP of the G16NB engine were increased by 4% points and 0.5 MPa, respectively, compared with conventional models with a single stage turbocharger of the same cylinder bore, stroke length, and engine speed. As a result, the world's top level electrical efficiency of 44.7% or higher was achieved.

Table 2 Test results

		Target	Measured
Intermediate cooling temperature	℃	＜Baseline−30	Baseline−35
Pressure loss of intermediate cooling	kPa	＜10	5
⊿Pressure ratio of compressor	−	Current model＋1.5	Current model＋1.5
⊿Turbocharger efficiency	% points	＋10	＋14
⊿Break mean effective pressure	MPa	＋0.5	＋0.5
⊿Generation efficiency	% points	＋4	＋4

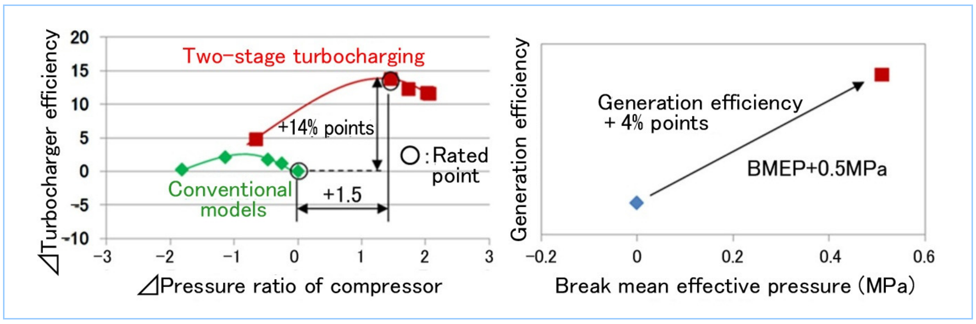

Figure 8 Test results

4. Conclusion

The new G16NB generator gas engine was designed, and the test engine displayed the test results below:

(1) The optimization of the two-stage turbocharging system was conducted. Turbocharger efficiency was raised by 14% points, and the pressure ratio was raised by 1.5 contrasted with those of conventional engine models.

(2) A two-stage turbocharging system was installed on the engine, and the engine electrical efficiency of the world's top level of 44.7% or higher was attained with an efficiency increase of 4% points. Furthermore, the BMEP was raised by 0.5 MPa in contrast to that of conventional models using city gas (13A) fuel.

This development was conducted with the aid of the fiscal 2012 "Program for Strategic Innovative Energy Saving Technology" of the New Energy and Industrial Technology Development Organization (NEDO).

Endurance testing using the verification facilities of our company shown in **Figure 9** will be conducted. In addition, plans call for the engine to be launched on the market after evaluating engine reliability including the two-stage turbocharging system.

Figure 9 Verification facilities