DEVELOPMENT OF A DRY LOW NOx COMBUSTOR FOR 1.5 MW GAS TURBINES

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ABSTRACT

To meet strict NOx regulations, Kawasaki Heavy Industries, Ltd. has been conducting a development program of dry, low-NOx combustion system since 1989. In a first step of the development program, a multi-burner type, can combustor has been developed. The test engine, with an output of 1.5 MW, demonstrated NOx emissions below 42 ppm at 15% O2. Following the engine performance tests, a 500 cycle endurance test and a 300 hour test at full load have been conducted to assess the reliability of the combustion system developed. During these tests, no mechanical durability issues arose.

INTRODUCTION

During the past 10 years, the regulations for NOx emissions have become more strict in the world. In Japan, new regulations for NOx emissions have been introduced in 1992. For example, in Tokyo area, NOx emissions should be below 28.6 ppm (15% O2) for gas turbines larger than 2 MW and 42.9 ppm for that under 2 MW. Although these NOx levels may be attained using commonly employed water injection, water injection consumes a large amount of water purified at high level, deteriorates gas turbine efficiency, and increases initial and operating costs of the facility. A selective catalytic reduction (SCR) system can reduce NOx emissions to ultra-low levels and does not affect gas turbine efficiency. However, SCR is more costly than water injection, and increases the complexity of facility operation to prevent a nonreacted ammonia leak. As the cost issue is more severe for small-to-medium size gas turbines, the cost increment for the NOx abatement impedes the growth of small-to-medium size gas turbine co-generation facilities. Therefore a development of dry, low-NOx combustion technology is the most urgent task for gas turbine manufacturers.

From 1989, Kawasaki Heavy Industries, Ltd., a powerful small-to-medium size gas turbines manufacturer, has been conducting, a development program of a dry, low-NOx combustion system burning natural gas that can be applied to 1.5 MW Kawasaki M1A-13A gas turbine. The operating characteristic of M1A-13A gas turbine is shown in Table 1.

Table 1 OPERATING CHARACTERISTIC OF M1A-13A

<table>
<thead>
<tr>
<th>Output(kW)</th>
<th>Pressure Ratio</th>
<th>Combustor Inlet Temperature(K)</th>
<th>Turbine Inlet Temperature(K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1500</td>
<td>9.4</td>
<td>613</td>
<td>1293</td>
</tr>
</tbody>
</table>

This paper describes the test results obtained throughout the first step of the development program.

DESIGN CONCEPT

Over past 15 years, Kawasaki Heavy Industries, Ltd. has been investigating a dry, low-NOx combustion technology suitable for small-to-medium size gas turbines. These experiences lead to consider that a lean-premixed combustion and a catalytic combustion are the most promising technology for burning natural gas. However, the catalytic combustion will not be able to be commercialized in the near term because of the short life of the catalyst (Kajita, et al., 1990). Consequently the combustor used in the development program was designed based on the concept of lean-premixed combustion.

Since NOx formation increases exponentially with the flame temperature, the objective of the lean-premixed combustion is to supply a homogeneous mixture for elimination of the locally hot zone and to reduce average flame temperature. However, the lean-premixed combustion has a narrow operating range over which both low NOx and CO are achieved. To broaden the operating range, a control of combustion air flow and/or fuel flow is required.

Although a variable geometry air control system has already been accomplished in some large industrial gas turbines (Aoyama and Mandai, 1984), there have been...
few successful applications of this technique in small-to-medium size gas turbines because of size and cost limitations. Moreover the operation reliability of the variable geometry is questionable in small-to-medium size gas turbines requiring a quick response in operation. On the other hand, a fuel staging procedure is quite simple and reliable because only a limited number of regulating valves are required to control the fuel flow to the combustor.

Therefore a combination of lean-premixed combustion and fuel staging (multiple burners) was applied to the combustor design. The air flow through the combustor changes slightly since the engine speed is constant from idling to full load. However, the fuel flow changes by a factor of about 2.5. To maintain a premixed burner equivalence ratio within a suitable range, the number of operating burners will be changed according to the load. The optimum shape and number of the premixed burners was determined by the results of preliminary tests.

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Figure 1. SINGLE BURNER COMBUSTION TEST FACILITIES

Figure 2. BURNER CONFIGURATIONS

PRELIMINARY TESTS

The development of the gas turbine combustor relies heavily on previous experience and experimental work because of the complexity of the combustion process. In the case of a dry, low-NOx combustor, however, there is little experience and experimental data. Therefore a design of the premixed burner must be supported by the experimental tests performed at the early stage of the development.

Figure 1 shows a single burner combustion test facility where the burner set in the premixing duct injects fuel upstream of the swirler. The fuel is mixed with combustion air and burnt at the swirler exit. The combustion tube equips an igniter, thermocouples for the measurement of burner outlet temperatures, and a gas sampling tube. Tests were conducted at atmospheric conditions and the combustion air temperature: T=300K using pure methane as a fuel.

Figure 2 shows the typical burners tested. Type I is a premixed burner that has a long mixing distance between the injection tubes and the swirler to produce a homogeneous fuel-air mixture. Type II injector contains a newly devised, dispersal fuel burner, that consists of a series of multi-orifice, radial tubes and a swirler to shorten a mixing distance. Type III is a diffusion burner that consists of a conventional multi-orifice gas injector and a swirler. The emission characteristics of the type III are compared with those obtained from the other burners.

The emission data of all burners are indicated in Figure 3. In the case of the diffusion burner, NOx emissions are over 14 ppm when the burner equivalence ratio is between 0.5 and 0.7, but CO emissions are less than 30 ppm. NOx emissions of the premixed burners tend to increase as the equivalence ratio increases, and are 70 to 90% less than that of the diffusion burner over the operating range. CO emissions of the premixed burners increase significantly as the burner equivalence ratio: $\phi$ decreases below 0.55, and at $\phi=0.5$, CO emission of the type I is higher than 1000 ppm and the type II causes a flame out. From these results, both type I and type II were expected to be a premixed burner for a low-NOx combustor, and the lower equivalence limit of the burner was set to 0.55.

COMBUSTOR DESIGN

Figure 4 shows a cross sectional view of the multi-burner combustor developed that can be retrofit to the current production combustor without any modification. The combustor consists of the can-type combustor liner and the burner module. The liner is of sheet metal, double-wall construction and incorporates impingement and film cooling. The exit geometry of the liner is the same as that of the current production one to maintain a retrofitability, but the diameter of the combustion zone is increased to increase combustor volume and ensure complete combustion. The liner head equips one pilot swirler at the center and eight main swirlers surrounding it.
The burner module is shown in Figure 5. The pilot burner is of conventional multi-orifice type and a small amount of pilot fuel is injected directly into the combustion zone. During engine operation, the pilot burner holds a stable diffusion flame downstream of the pilot swirler to enhance combustion stability. As the diffusion flame produces higher NOx emissions than the lean-premixed flame, pilot fuel flow is limited to as low as 5% of maximum fuel flow. The main burner consists of a series of multi-orifice, radial tubes that inject main fuel upstream of the main swirler. The fuel is mixed with combustion air prior to entering the combustion zone. The eight main burners make four groups of two as shown in Figure 6, and the first group operates incorporated with the pilot burner for light-off at 5% of engine speed. The second group joins at 22% speed to increase fuel flow for engine acceleration. This group is stopped when engine reaches rated speed to avoid flameout. From idling to 25% load, only the first group operates, and successive groups of burners are supplied sequentially according to every increment of 25% load. Consequently all burners operate from 75 to 100% load.

DEVELOPMENT PROCEDURE

Following the preliminary tests, a prototype combustor was designed and fabricated. This combustor was installed in the combustor test rig for the initial check of performance. After several modifications had been made on the combustor through a series of tests, engine tests were carried out to demonstrate a low-NOx capability of the combustor developed. Finally the whole combustion system was installed in the generator unit to confirm the system reliability in a typical operation environment.

Figure 4. CROSS SECTION OF MULTI-BURNER COMBUSTOR

Figure 5. BURNER MODULE

Figure 6. BURNER GROUPS IN THE MULTI-BURNER COMBUSTOR

Figure 7. SCHEMATIC DIAGRAM OF COMBUSTOR TEST FACILITIES
COMBUSTOR RIG TESTS

Test Facilities

A schematic diagram of the combustor test rig is shown in Figure 7. The combustor inlet-air is supplied from a centrifugal compressor, heated by an indirectly fired pre-heater, and regulated at a constant pressure level by a butterfly valve installed in an exhaust duct. Pure methane is supplied from commercial gas cylinders, regulated by a valve and metered by an orifice-meter. The exhaust gas temperatures are measured by 24 R-type thermocouples at the combustor exit. Continuous monitoring of the emissions is accomplished for NOx, CO, UHC and O2.

Combustion tests were carried out both in the reverse flow type test section and the engine simulator one. Though the configuration of the reverse flow type test section does not simulate the air flow condition of the actual engine, the straight-through exhaust design allows the combustion flame to be observed. The engine simulator test section was used to evaluate the combustor performance at the condition that is the same inlet-air flow as the actual engine.

Combustor Rig Test Results

The primary objective of the combustor rig test is to determine the basic combustion characteristics of the dry, low-NOx combustor prior to the engine test. Tests are run at the air pressure of 0.3 Mpa because of the limitation of compressor capacity. The other conditions, such as combustion air temperature and velocity are adjusted to the engine operating conditions.

To clarify the characteristics of the flame of the multi-burner combustor, visual observations of the flame were carried out preceding the combustion performance test. The color of the flame for the premixed burner was light blue, and blue with a yellow front for the pilot burner. Flame shapes of premixed burners were always stable and no signs of unstableness were observed even at the near lean limit condition.

Figure 8 indicates the typical emission results of the type II premixed burners obtained in the reverse flow test section. NOx emissions are converted into the dry, low-NOx combustor prior to the engine test. NOx emissions decrease rapidly as fuel/air ratio decreases. However, when the equivalence ratio of each premixed burner decreases below 0.55 that is near the lean limit of this type of combustor, the combustion efficiency decreases rapidly with the result of significant increase of CO and UHC emissions. Under all-burner operating conditions, 25 to 42 ppm (at 15% O2) of NOx level was attained while CO and UHC were at a low level. For the type I, almost the same results were obtained.

To examine the effect of the reference velocity on the flashback, combustion tests were carried out changing the reference velocity from 5 to 15 m/s. For the type I, flashback occurred at 7 m/s, that is half of the designed velocity. On the other hand, no flashback occurred for the type II over the velocity range of 5 to 15 m/s. Therefore the type II was selected as a practical burner.

Emissions results obtained in the engine simulator test section are shown in Figure 9. Since the turbine scroll is installed in this test section, some part of the combustion air is used for the scroll cooling. Therefore the fuel/air ratio in Figure 9 includes the effect of the scroll cooling. As a result, the emission curve shifts to the left. As was reported previously (Mori, et al., 1977) the combustion air flow rotates around the combustor liner in this type of test section. As the result, flame stability was improved and 12 ppm (at 15% of O2) of NOx was achieved at partial load condition without significant reduction of the combustion efficiency.

Throughout the rig tests no flashback problem occurred, and 30 ppm or less of NOx emissions was expected at the rated engine operating condition.

ENGINE TEST

Test Facilities

Following the combustor rig check, the multi-burner combustor was installed in the 1.5 MW class gas turbine engine, Kawasaki M1A-13A. To evaluate the engine performance and the emission characteristics, the engine was connected with the hydraulic dynamometer and controlled manually. The performance was measured at every 185 kw load. The fuel staging points were determined by the data of NOx emissions and combustion efficiency.

Figure 8. EMISSION RESULTS AT THE REVERSE FLOW CONDITION

Figure 9. EMISSION RESULTS WITH A SCROLL
The engine was instrumented with eight thermocouples downstream of the last stage turbine to measure exhaust gas temperature. The exhaust gas was continuously sampled to analyze NOx, CO, UHC, CO2 and O2. Combustion efficiency was calculated by the emissions data.

To detect flashback during engine operation, eight thermocouples were installed just upstream of each main swirler. In the case of flashback, the fuel supply to the burner causing flashback is stopped momentarily and the same fuel flow to the stopped burner is added to the pilot burner.

After the engine performance and emission characteristics were confirmed, a 500 cycle, cyclic endurance test was conducted to evaluate the reliability of the combustion system. The pattern of the cyclic test is shown in Figure 10 which consists of a startup, a stepwise loading, a continuous running at maximum load, a stepwise downloading, a shutdown and a purge for the next start. These operations are automatically controlled by a sequencer. One cycle needs approximately 11 minutes which includes 210 seconds of maximum load operation.

Finally the complete combustion system was installed in a gas turbine generator unit and operated to assess the system reliability and durability in a typical operating environment. The total operating time in the generator unit was approximately 300 hours. NOx and O2 emissions were sampled continuously at the exhaust duct. In addition, the engine response for the stepwise load change was evaluated, and the operation of the flashback protection system was confirmed by the use of a sham signal.

Test Results

NOx emissions and combustion efficiency obtained in the engine tests are shown in Figure 11. The NOx emissions indicate a saw-tooth pattern because additional groups of injectors are switched on as load increases. From 25 to 100% load, NOx emissions were below 42 ppm (at 15% O2), and at the rated load NOx indicated 26.7 ppm that has a large enough margin against the 42 ppm target. Combustion efficiency was over 99.5% from 80 to 100% load though it decreased near the fuel staging points when load decreased. NOx emissions and combustion efficiency were strongly affected by the degree of swirl imparted to fuel-air mixture. A high degree of swirl increased both NOx emissions and combustion efficiency because the effective area of the swirler passages decreased, and caused flashback. Therefore the swirler angle was decreased from 45° to 35° to weaken the rotation. This modification caused a slight increment of CO and UHC emissions. After this modification, neither flashback nor flameout occurred even in the cyclic endurance test.

The responses to the loading and unloading of the generator unit was the same as that of conventional combustion systems. Further the flashback protection system was confirmed to operate properly by input of a sham signal and no increment of NOx emissions was observed in this test.

Figure 12 is a photograph of the multi-burner combustor after the cyclic endurance test and the full load test. No damage was observed in the combustion liner.

CONCLUSION

To realize a dry, low-NOx combustion, a multi-burner type, can combustor has been developed. This combustor is designed based on the principle of lean premix combustion, and the fuel staging technology is applied to broaden the operating range. To enhance flame stability, a conventional pilot burner is used.

The performance and reliability of the combustion system were assessed through the engine performance tests, the 500 cycle, cyclic endurance test and the 300 hour full load test.
At the engine tests, the combustor demonstrated a NOx level below 42 ppm at 15% O2 between 25% and 100% load. During the cyclic endurance test and the full load test, no mechanical durability issues arose. The next step of the development program is to reduce NOx emissions below 20 ppm. Now the combustion improvement is continuing.

REFERENCES


Aoyama, K. and Mandai, S., 1984, "Development of a Dry Low NOx Combustor for a 120 MW Gas Turbine," ASME Paper No. 84-GT-44.
