Proceedings of the International Gas Turbine Congress 2003 Tokyo
November 2-7, 2003

The Development of 300kW Class High Efficiency Micro Gas Turbine "RGT3R"

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ABSTRACT

Niigata Power Systems Co., Ltd. has developed a continuous use 300-kW electric output power Recuperated Cycle Gas Turbine RGT3R with a liquid fuel Dry Low Emission (DLE) combustor from 2000 to 2003. The thermal efficiency specification at the generator end of the RGT3R is 32.5%. The latest prototype has been confirmed to achieve a thermal efficiency of 31.4% or higher and NOx emissions of 20 ppm or less in terms of 16%O2 conversion at load rates between 50% and full load.

The first commercial production is expected to start in the beginning of April 2004 when all the durability testing are completed.

INTRODUCTION

For the past few years, a number of manufactures have developed a Micro Gas Turbine (MGT) system or its peripheral applications. However, an assessment of these devices in terms of total life cycle cost, including the sum of initial cost, construction costs, operating costs, and maintenance costs, shows that a current MGT requires extremely high costs per unit output due to their limited power output. Cost breakdowns for 30-kW and 300-kW MGT are shown in Fig. 1. Although the difference in the outputs of these two turbines is significantly large, the difference in the initial construction costs, including electrical work, is small as shown in Fig. 1. Therefore, the MGT has not widely used due to this disadvantage.

In order to minimize the total life cycle cost per unit output, an appropriate output had to be determined. The result of this study is tabulated in Table 1, such as the target electric output and the targeted thermal efficiency at the generator end for two cases, a simple cycle and recuperated gas turbines. Based on this result, two types of gas turbines were developed, which were the RGT3C, a 300-kW-class simple cycle gas turbine, and the RGT3R, a recuperated cycle gas turbine.

In this paper, the development of the RGT3R recuperated cycle gas turbine is described, focusing on its history, features, and the performance on the latest testing result.

Table.1 Design Specification of Niigata 300kW Class MGT

Engine Name	RGT3R	RGT3C
Air Flow Rate	2.5kg/s	←
Pressure Ratio	4.02	←
Rotation Speed	36943 min-1	←
Electric Output Power	300kWe	360kWe
Thermal Efficiency @ Generator End	32.5%	16.3%
Remarks	Recuperated Cycle	Simple Cycle

Specifications are expected to be under the ISO condition.
T0=15℃, P0=1013hPa, Inlet & Exhaust Pressure drops=0kPa

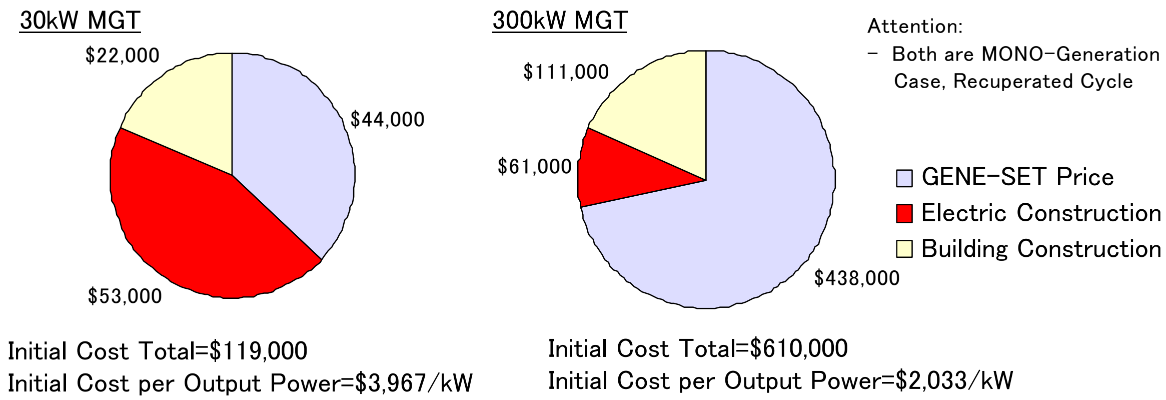

Fig.1 Initial Cost Comparison between 30kW and 300kW Recuperated MGT (300kW MGT GENE-SET Price indicates an example of the first testing GENE-SET price.)

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Manuscript Received on March 31, 2003

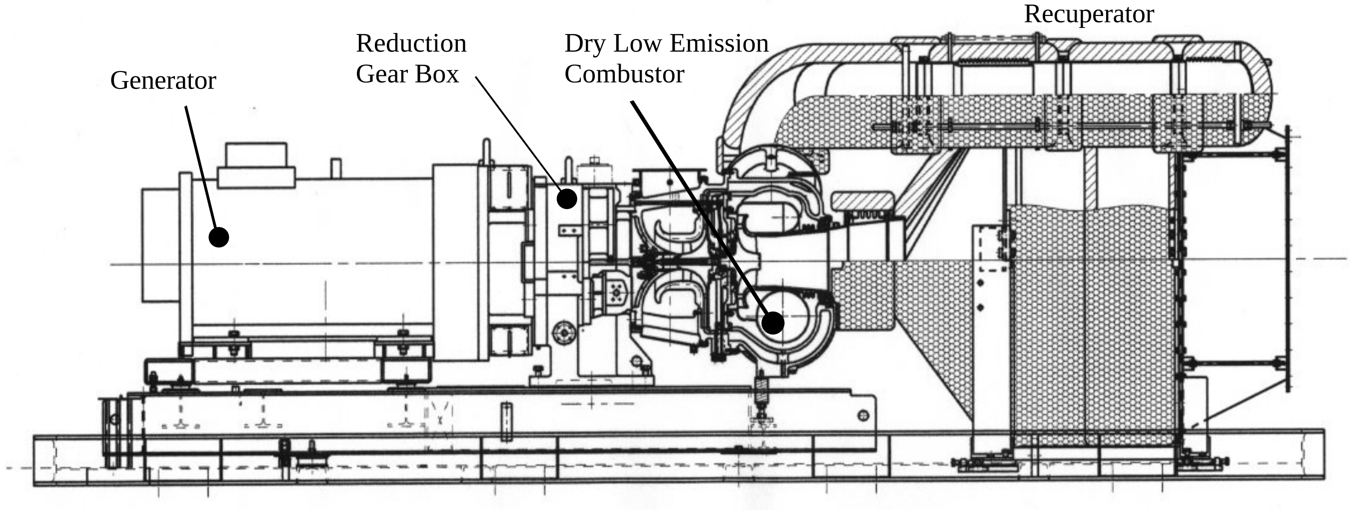


Fig. 2 Cross Sectional View of RGT3R

OVERVIEW OF THE DEVELOPMENT

Several important issues in the development of our MGT project are suggested as follows. First of all, turbochargers for both land and maritime diesel and gas engines have been manufactured for over 40 years in our facilities. This experience not only serves as the technical foundation for the development of the new MGT project; it also plays an important role in terms of production. Second, we have already released the "RGT3" gas turbine, an emergency-use in 1997 (Kobayashi et al.). This RGT3 provides the platform engine for the RGT3C and RGT3R continuous use gas turbines. Development programs for these MGT were launched at the end of 2000, a mere two years before.

Fig. 2 shows a cross sectional view of RGT3R. Both RGT3C and RGT3R are single shaft MGT which consist of the same single stage centrifugal compressor and radial turbines. These MGT are equipped with Liquid Fuel DLE (Dry Low Emission) combustor developed by collaboration between NAL (National Aerospace Laboratory of Japan, Tokyo) and Niigata Power Systems Co., Ltd.

This unique DLE combustor, it has successfully reduced emission levels of NOx, UHC, and CO to 20 ppm, 50 ppm, and 50 ppm or less, respectively, under conditions of 16%O_2 conversion and kerosene fuel burning. This DLE combustor can also operate in low-pollution mode at load rates of 50% or higher. This characteristics is achieved by adopting a two-stage parallel premixed pre-vaporized lean burn system, implemented in a simple configuration. Not only does such a configuration reduce costs; it also provides reliability superior to existing multi-stage premixed lean burn systems.

The Aerodynamics design at the centrifugal compressor was completely re-designed from the platform engine "RGT3" by using 3D CFD analysis. Since the required efficiency was high, 2D potential flow analysis was not appropriate for this MGT. Therefore, CDF Aerodynamics Optimization was used along with the conventional 2D potential flow aerodynamics analysis.

Among all the tasks associated with this development project, development of the Recuperator, including ducting system, was one of the most difficult engineering challenges during this MGT development program.

Another significant challenge was to develop a Man-Machine interface to meet the specific demands of the MGT users. It was determined that users would demand ease of use, high reliability, and an advanced remote monitoring system. In order to meet these requirements, an integral digital control system was selected. This system controls not only the main engine body, but also the generator and auxiliary equipment that is inside the enclosure.

R&D OF THE ENGINE COMPONENTS

1.Compressor

Development of the RGT3, an emergency-use gas turbine was already completed before RGT3C and RGT3R were designed. However, the RGT3 differs significantly from the RGT3C and RGT3R design specification. The characteristics of compressors of the conventional RGT3 and RGT3R(C) are compared and shown in Table 2. RGT3R(C) is required to satisfy an air flow rate of 2.5 kg/s, a pressure ratio of 4.02, and an adiabatic efficiency of 81% or higher. Compared to RGT3R, the air flow rate of RGT3 is 8% higher and the adiabatic efficiency is 5 point lower than RGT3R(C) rating point compressor performance. This suggests engine performance problems. In order to resolve these problems, a new compressor was designed by 2D potential flow analysis with 3D CFD technology, instead of the conventional compressor used in the RGT3.

Table.2 A Comparison between the RGT3 and RGT3R (C) Compressor Characteristics at the RGT3R rating point

Engine name	RGT3R (C)	RGT3
Rotating Speed	36,943 min^{-1}	37,005 min^{-1}
Air Flow Rate	2.5 kg/s	2.7 kg/s
Pressure Ratio t-t	4.02	4.2
Adiabatic Efficiency t-t	Up to 81%	76%
Number of Blades	Full Blade: 9 Splitter Blade: 9	Full Blade: 20
Impeller Shape		

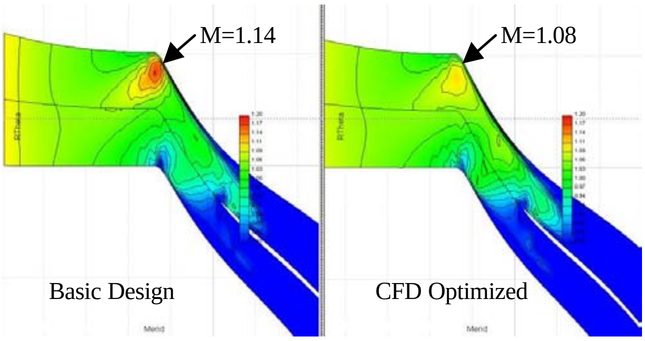


Fig. 3 Relative Mach Number distribution at the impeller inducer tip area.

CFD analysis of the relative Mach number distribution at the impeller inducer tip area is shown in Fig. 3. The left is the CFD result of an initial impeller design based on conventional 2D potential flow design. This indicates that the Mach number is large at the top end of the blade. In contrast, the right figure is the CFD results for an impeller with optimized inducer inlet angle using CFD. This represents that the relative Mach number is lower at the same point. The difference arises because of the viscosity of the boundary layer that develops on an impeller shroud. The 2D potential flow analysis can not handle the effects of the viscosity. In order to overcome this issue, 3D CFD optimization was introduced. In summary, by utilizing the 3D CFD optimization, the relative Mach number at the impeller inducer tip area was reduced from 1.14 of the initial shape to 1.08.

Velocity vector distribution on the suction side of the impeller splitter blade was estimated by CFD as shown in Fig. 4. The left figure, which is for the impeller initial shape based on 2D potential flow design, indicates the adverse flow circulation in the vicinity of a shroud wall surface. Such adverse flow circulation is likely to encourage the development of a boundary layer, which results in increased blockage and degrading of impeller performance. In contrast, an optimal curvature of the shroud wall surface by CFD is shown in the right and the adverse flow is eliminated as shown in the left figure.

Performance test results of a compressor optimized with CFD is represented in Fig. 5. The air flow rates and pressure ratio almost meet the design specification of RGT3R as shown in Fig. 5. On the other hand, adiabatic efficiency still does not satisfy the design specification, falling 1 point short. Since this causes a 0.5% shortfall in terms of engine thermal efficiency, this parameter needs to be improved before the first commercial production starts.

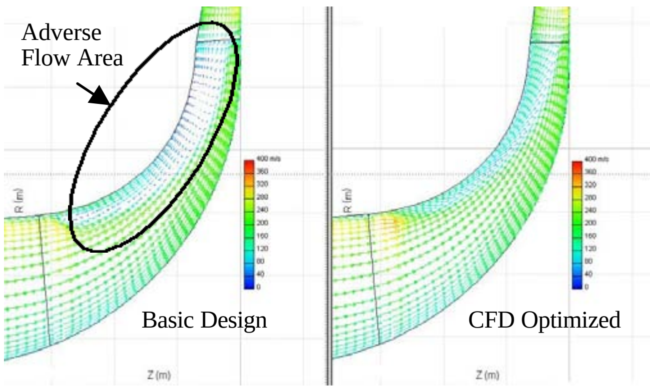


Fig. 4 Relative velocity vector distribution comparison at the suction side of the impeller splitter blade.

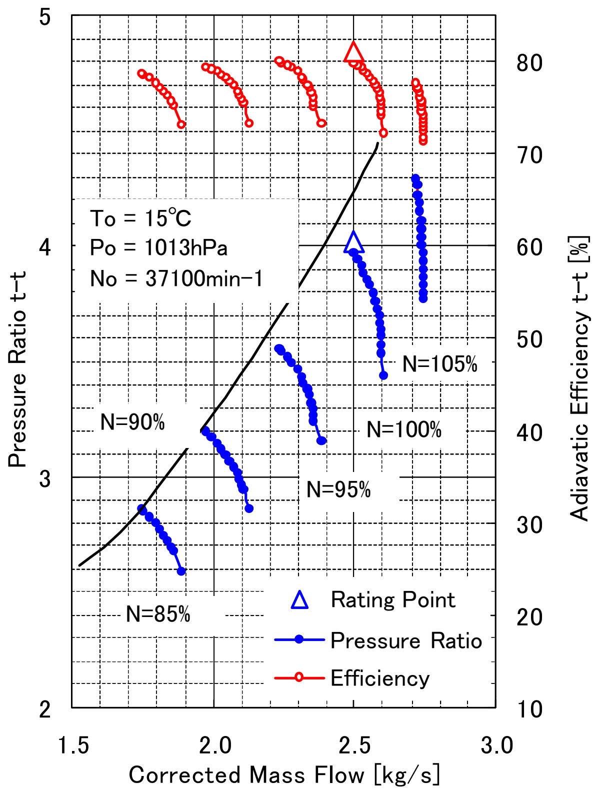


Fig. 5 Test Results of CFD Optimized Compressor Performance

2. Turbine Wheel & Rotor Dynamics

For both RGT3C and RGT3R, high-performance turbine wheels of RGT3 were used because of the extensive past results on these turbine wheels. Since both simple cycle and recuperated cycle require different turbine expansion ratios due to the recuperator pressure drops even at the same cycle pressure ratio, two types of turbine nozzles were designed for both cycles. The rotor assembly of RGT3R(C) with a power output coupling is shown in Fig. 6. The turbine wheel is made of nickel-base heat-resistant super alloy casting. The Rotor assembly is supported by tilting pad journal bearings in order to minimize a maintenance interval. The mechanical losses of RGT3R(C) are 40% greater than that of the RGT3, which used a ball-and-roller bearing. Reducing these excessive losses remains to be resolved in the future.

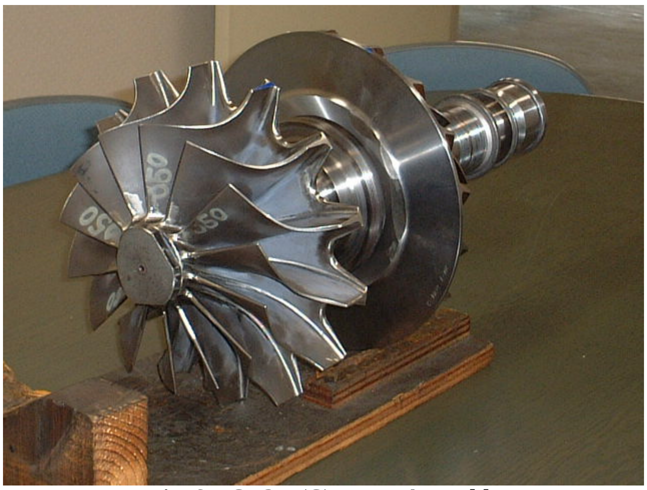


Fig. 6 RGT3R (C) Rotor Assembly

The predicted and measured unbalance response at the rotor output-coupling end is shown in Fig. 7. This figure indicates that there is no resonance point near the rating rotating speed and that the amplitude is very small. Therefore, the combination of the rotor and tilting pad journal bearing offers very good balance between bearing attenuation and stiffness as well as good stability across a wide range of velocities. If a direct-drive high-speed generator with a wide range of operating speeds is adopted in the future, this characteristic will be a significant advantage.

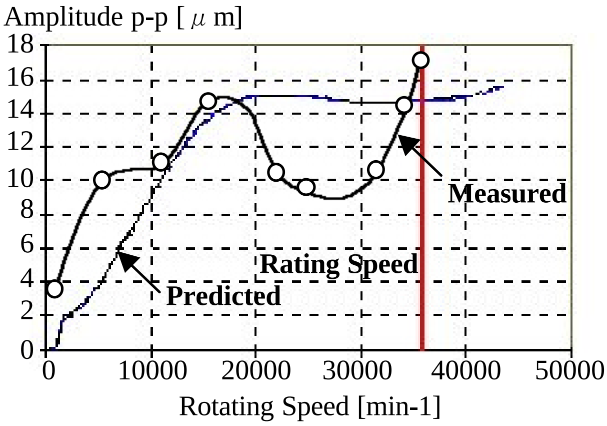


Fig.7 Comparison between Unbalance Response analysis Prediction and Test Data at the Rotor Output Coupling end.

3. Casing Structural Analysis and Material Development

Recuperated Cycle was chosen for the RGT3R to reduce the fuel consumption. Since the temperature of the combustion air discharged from the Recuperator becomes much higher than Simple Cycle case, the temperature gradient in the turbine casing of the Recuperated Cycle becomes much larger than that of Simple Cycle. This high temperature gradient induces strong thermal stress. Usually such a kind of high thermal stress causes the short-term fatigue of the turbine casing. In order to achieve sufficient fatigue life, new material called "PNX-TES3" was developed and used for RGT3R, instead of conventional stainless casting steel, SCH11. The PNX-TES3 was developed by collaboration between Niigata Power Systems Co., Ltd. and a casting manufacturer.

FEM analysis for the RGT3R is shown in Fig. 8. Evaluation of a low cycle life was carried out in accordance with ASME N-47 boiler and pressure vessel design code as shown in Fig. 9. The upper figure is the low cycle fatigue life of SCH11, indicating that the fatigue life under daily start-and-stop operation is only 8.3 years. The lower figure shows that in the case of PNX-TES3, low cycle fatigue life can be ensured for 25 years or longer. This is due to the low thermal expansion and high thermal conductivity characteristics of PNX-TES3, which reduce thermal stress levels and increase the high temperature fatigue strength of PNX-TES3.

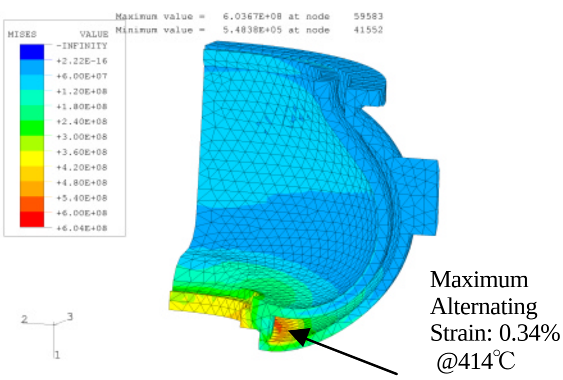


Fig.8 A FEM analysis example of the RGT3R Turbine Casing (SCH11)

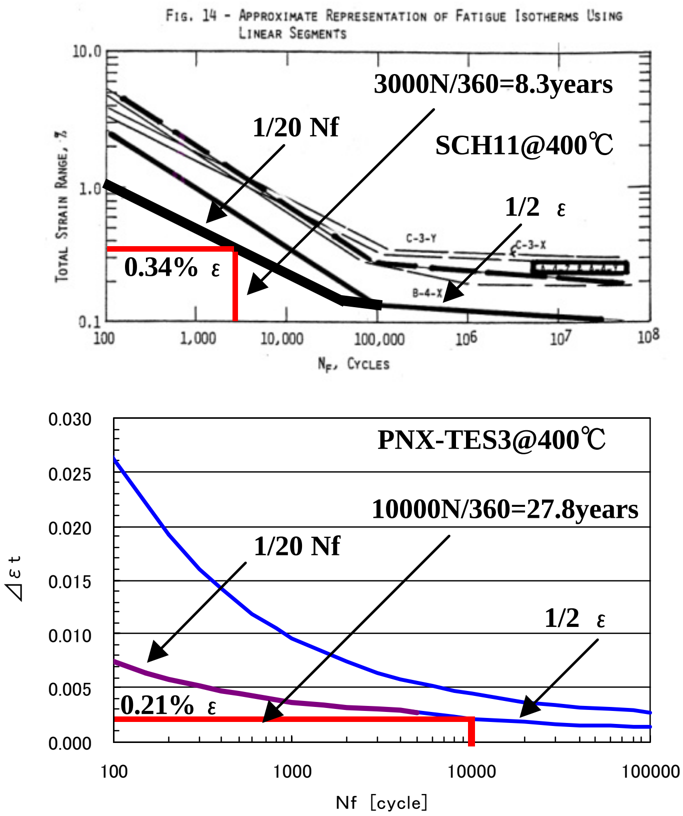


Fig.9 Low Cycle Fatigue Life of the RGT3R Turbine Casing Evaluated in accordance with ASME N-47 code (Upper: SCH11 / Lower: PNX-TES3 @400℃)

4. Development of the Recuperator, & Ducting system

The development of the Recuperator, including ducting system, was one of the most difficult engineering challenges during this MGT development program. The recuperated cycle is a well-understood area of thermodynamics. But this cycle had not been applied on the MGT because of inadequate heat exchange capacity of recuperator per unit volume and insufficient recuperator durability. Today, recuperator technology has made remarkable strides, thanks to sustained efforts by heat transfer engineers, materials scientists and technical staff at various companies. The success of the RGT3R development project owes much to joint efforts with a recuperator manufacturer.

At the start of this project, evaluation of the durability on the recuperator and a method to support the ducting system and recuperator are focused as a development goal.

Successful development of a high-performance recuperator involved finding a way to reduce the thermal stress of a heat-exchanging core and attached structural members of the recuperator. Moreover, since reactions resulting from pressure within the recuperator ducting system often resulted in unexpected damage on the recuperator. A design in which the reaction of the ducting system was balanced needed to be overcome. Furthermore, since the heat-exchanging core of the recuperator was an integral part by means of special heat-resistant brazing, this could not be repaired by any means, which made the design even more difficult. Thus, investigations including highly accurate low cycle fatigue life predictions were needed before manufacturing of an actual recuperator began. In order to overcome these design specifications, 3D CAD with FEM structure analysis system played a central role in studies of ducting configurations. It also facilitated effective communications with the manufacturer of the recuperator.

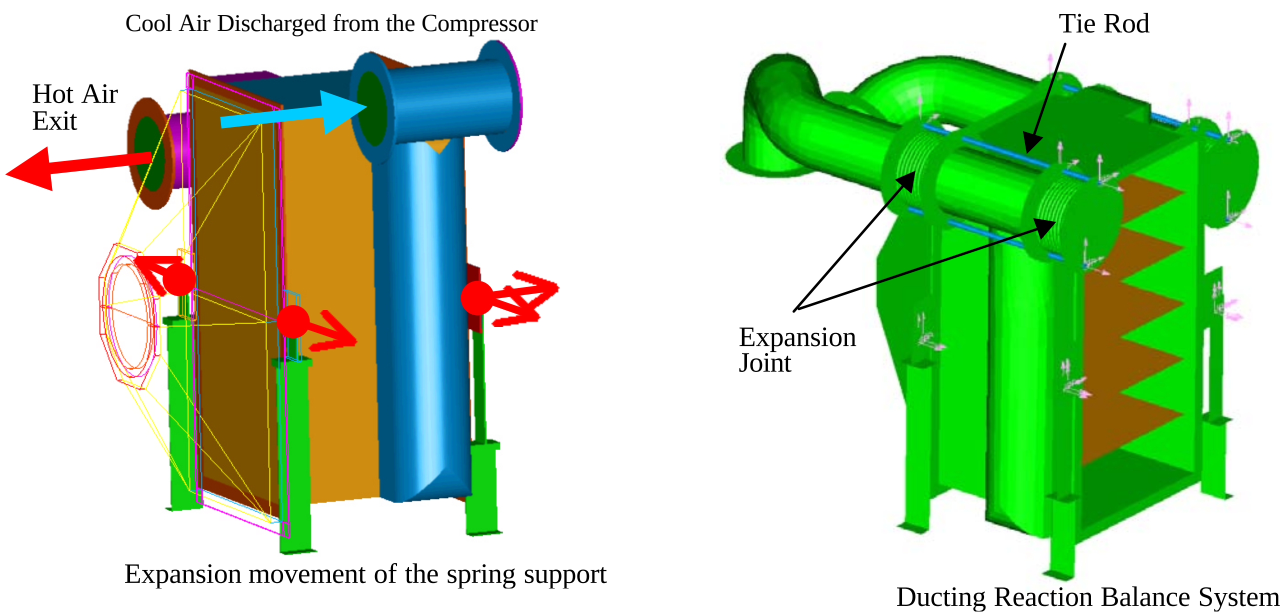


Fig.10 Schematics Recuperator Support (on the left) and Ducting Reaction Balance System (on the right).

A method of supporting the recuperator and a method of balancing the ducting system reaction are shown in Fig. 10. The recuperator is supported with spring supports, while the ducting reaction is balanced with the expansion joints, and the tie rod. Although these structures are simple, high reliability and low cost are achieved.

FEM analysis of the recuperator and the ducting system in the final configuration is shown in Fig. 11. Stiffness data for plate fin arrays within the recuperator required for the FEM analysis was provided by the recuperator manufacturer. This figure shows stress distribution on surfaces of the recuperator caused by thermal gradients, internal pressure and reaction force of ducting system. A maximum stress point was at a cold air header tank (settling chamber) of the recuperator. However, low cycle fatigue life and creep rupture at this maximum stress point would be sufficient because of the low temperature. The most critical point was a hot air header tank (Collector) of the recuperator as shown in Fig. 12. Since this portion was exposed to high temperatures, close to 600°C, under the rated operating conditions, low cycle fatigue and the possibility of creep rupture needed to be evaluated carefully.

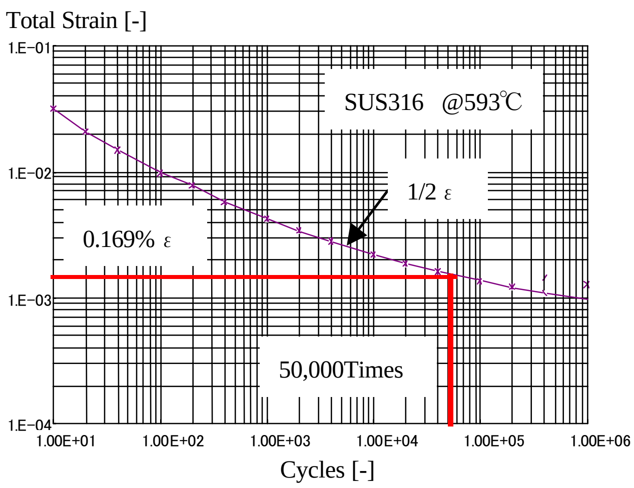


Fig.12 Evaluation of Low Cycle Fatigue at the Hot Air Header Tank (Collector)

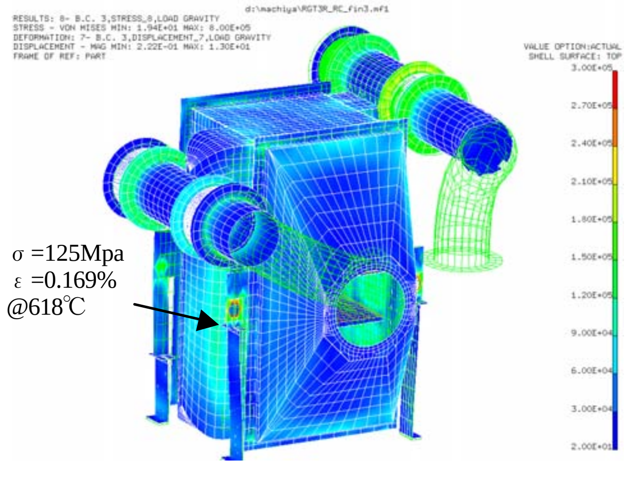


Fig.11 FEM analysis of the final Recuperator and Ducting configuration.

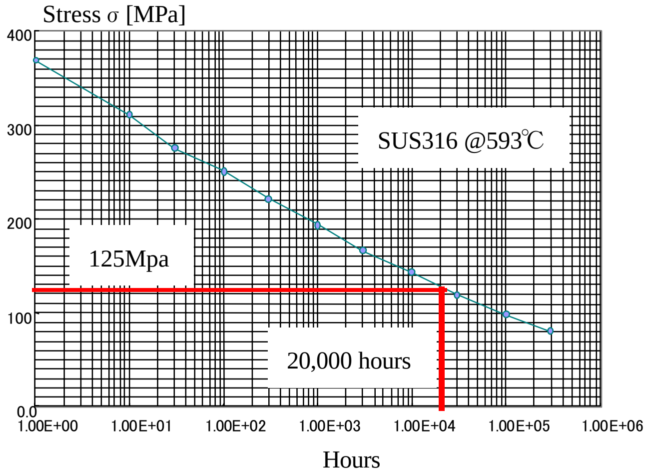


Fig.13 Evaluation of Creep Rapture at the Hot Air Header Tank (Collector)

The low cycle fatigue life prediction and creep rupture time prediction for the hot air header tank of the recuperator are shown in Fig. 12 and Fig. 13, respectively. The evaluation of the low cycle fatigue life and creep rupture time was based on the ASME N-47 boiler and pressure vessel design code. The life of the recuperator was set to be three years under the conditions of daily start and stop operations. Therefore, the required low cycle fatigue life and the required creep rupture times were at least 4,000 times and 20,000 hours, respectively. The low cycle fatigue life was estimated to be approximately 50000 times as shown in Fig. 12, which was well above the design specification. On the other hand, the predicted creep rupture time was approximately 20000 hours, which was the design specification. Thus, further consideration is required on the structure in order to improve the creep rupture time.

GENERATOR SET DESIGNS

Internal devices were an important design consideration of the gas turbine generator set. Since the configuration of the internal devices was constrained by a number of factors, including physical volume and weight, ventilation and cooling, vibration and noise reduction & pressure drops, electrical noise immunity and so on.

2D CAD had been used to determine the equipment configuration of the generator system for the past several years. For the RGT3R, 3D CAD was used for this task, making it possible to resolve physical restrictions on the arrangement of equipment in the generator set. The 3D CAD shortened the time required for design as well. The equipment configuration inside the latest RGT3R generator set is shown in Fig. 14. The reduction gearbox, generator, lubricating oil cooler, and AC starter motor shown in the figure are the devices that have been installed on MGT systems

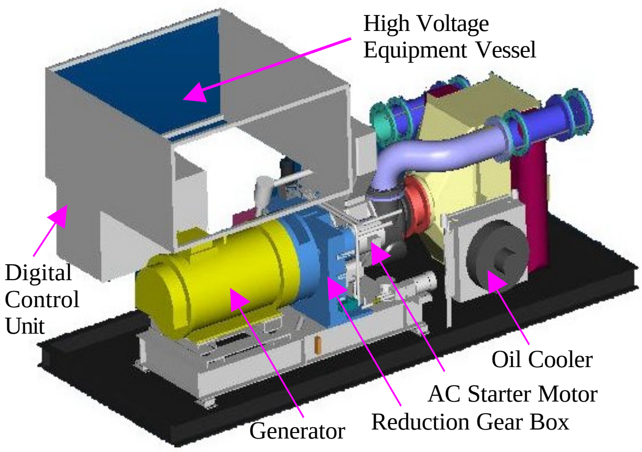


Fig.14 The Latest equipment's configuration in the Gene-set Enclosure

Fig.15 An Example of LCD Man-Machine Interface Panel

before. These auxiliary equipment inside the generator set come with satisfactory reliability, and are controlled by an integral digital programmable logic controller. An independent power-source system was selected for the auxiliary equipment in order to provide all electric power needed by the auxiliary equipment using a generator through a transformer, once the gas turbine is started.

Given the specific needs of the MGT users, Man-Machine interface was an important consideration. Important considerations include ease of use, high reliability. And an advanced remote monitoring system is required. An example of the LCD user interface and a schematic of the remote monitoring system are shown in Fig. 15 and Fig. 16, respectively. The RGT3R integral digital controller in a small control box controls not only all the functions of the GENE-SET but also entire Co-generation plant.

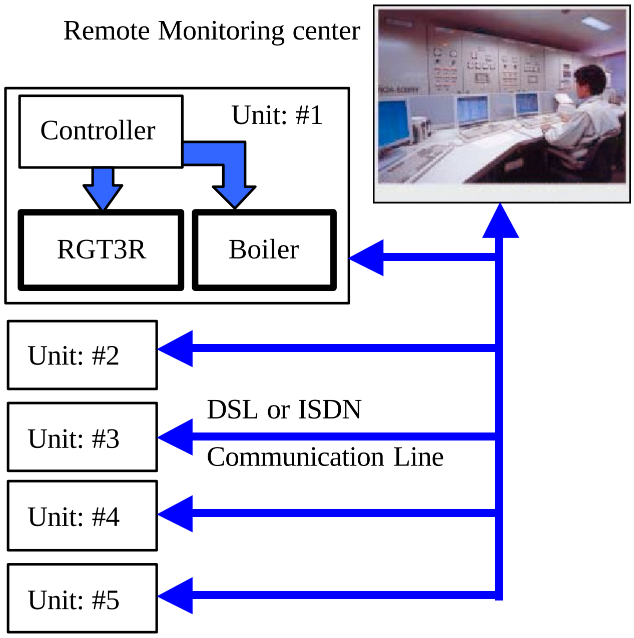


Fig.16 A Schematic of Remote Monitoring System

TEST RESULTS OF ENGINE PERFORMANCE

Testing of the first prototype engine began in May 2002. The first test was a simple cycle test of the RGT3C with a diffusion combustor. At the same time, unit tests of a liquid fuel DLE combustor and the generator set design were started as well. The primary objective of these testing was to confirm compatibility of a newly designed compressor, a rotor assembly including an existing turbine wheel, and a tilting pad journal bearings; specifically, to measure unbalance response and the performance of the new compressor.

In the second stage of testing, recuperated cycle RGT3R operations began at the end of August 2002. The response delay of the fuel control was measured and it was determined that the delay was unsatisfactory due to the modifications on the recuperated cycle, such as heat capacity of recuperator. In order to overcome this issue, the fuel control device and the software were immediately replaced to meet the specification.

For the third stage of testing, operation of the recuperated cycle RGT3R with a liquid fuel DLE combustor was started in January 2003.

Most of this chapter presents the most recent information on engine performance of the recuperated cycle gas turbine RGT3R.

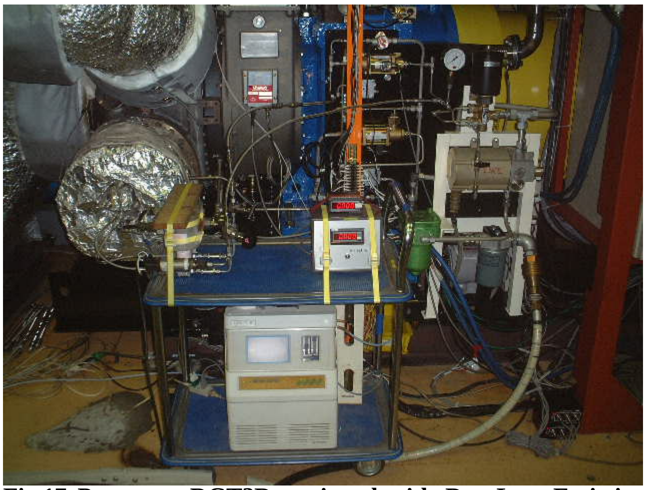


Fig.17 Prototype RGT3R equipped with Dry Low Emission Combustor

The RGT3R prototype equipped with a liquid fuel DLE combustor is shown in Fig. 17. Fuel metered by an electric governor is distributed to a pilot and main burners. A shut-off valve for the main burner fuel is opened at a staging point, which corresponds to a load rate of approximately 50%. At the same time, a servo valve controls the pilot burner fuel ratio. During preliminary testing, instead of a servo valve, a manual needle valve is attached to the pilot fuel line as shown in Fig. 17. The flow rate of the pilot and main fuel were measured with precise volume flow meters in the testing.

Except for the DLE fuel control hardware, the equipment of the testing were those to be used for the commercial generator set product. These equipment include the recuperator and ducting system, the newly designed reduction gear box, a case-coupled generator, integral digital control unit and high voltage control console. The prototype RGT3R engine as part of a complete system is shown in Fig. 18.

Fig.18 Prototype of RGT3R Engine Test Rig (Completed)

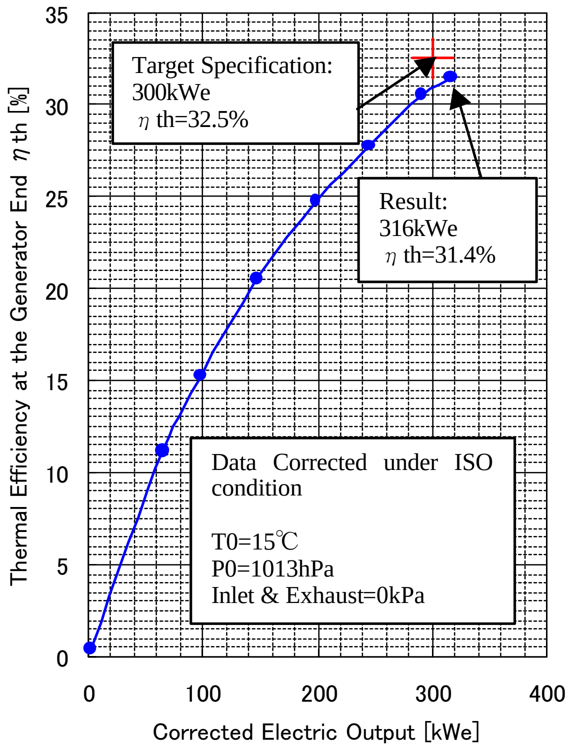


Fig.19. Thermal Efficiency of prototype RGT3R

Described in this paper, numerous component tests were performed, such as the metal sealing test of the recuperator ducting. Moreover, since an extremely high temperature air for combustion was required for the DLE combustor development, a DLE combustor test rig was significantly modified (the changes were so significant that the associated work equaled the work required building a new test rig). The compressor underwent similar modifications. Several types of diffusers were tested to assess their performance over a very short period as well.

The test result of the latest thermal efficiency and electric output of the generator-end is shown in Fig. 19. The values for engine performance given in the figure have been corrected to comply with the ISO standard condition. Since the generator-end thermal efficiency is currently 31.4%, which is 1.1 points lower than the target design specification of 32.5%, this parameter must be improved. No serious failure has occurred at this point and the accumulation of extensive data performance has achieved at this stage of the testing. This data will be evaluated to improve the performance. The next stage of the testing is to evaluate a newly designed diffuser and determine how to reduce mechanical losses.

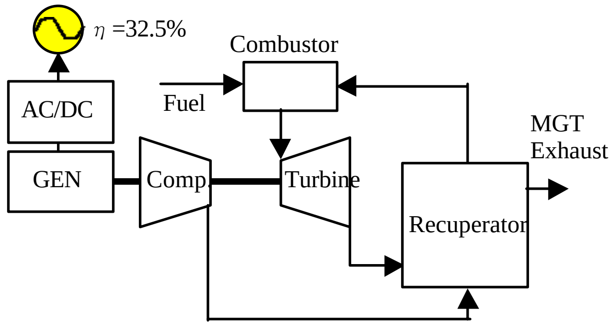
DURABILITY TEST, COMMERCIAL PRODUCTION & FUTURE PLAN

Due to the limited laboratory-testing environment, an on-site field test will be carried out at the actual premises of a customer from the end of 2003, for at least about a half year (4,000 hours). In this on-site field testing, durability will be evaluated. In the meantime, an engine performance improvement test of RGT3R and a development of a "Gas Fuel DLE" combustor will be conducted at the Niigata Power Systems Co., Ltd. When these testing are completed, the first commercial production will be started in the beginning of April 2004.

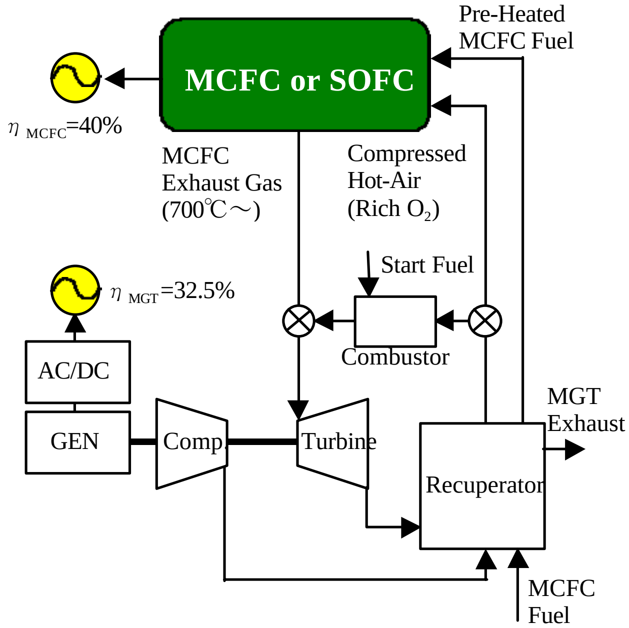
Although, direct-drive high-speed generator and a power converter system may be introduced to the RGT3R in order to improve thermal efficiency under partial loads. This improvement is possible since engine-rotating speed will no longer be a constraint. If the application of the direct-drive high-speed generator and a power converter system to the present engine is successful, the RGT3Rs can be used the combined power generation with a fuel cell as shown in Fig.20.

It is well known that venting high-temperature exhaust gas during operations and supplying air with higher oxygen content (boosted by compression) dramatically improve the thermal efficiency of molten carbonate fuel cells (MCFC) or solid oxide fuel cells (SOFC). The exhaust gas in this process is hot enough to drive a recuperated cycle gas turbine. In a combined MCFC (SOFC) / recuperated cycle gas turbine, high-temperature exhaust gas from the MCFC (SOFC) can be used to drive the gas turbine, while the recuperator provides high temperature compressed air (with increased oxygen concentrations) to the MCFC (SOFC). In such a system, the MCFC (SOFC) functions as a combustor for the recuperated cycle gas turbine. The thermal efficiencies of MCFC (SOFC) and recuperated cycle gas turbines are approximately 40% and 30%, respectively, when operated independently. However, overall system thermal efficiency can be improved up to 70% by combining these. Therefore, if the recuperated cycle gas turbine and MCFC (SOFC) are successfully combined, the gas turbine is able to gain the total thermal efficiency of power generating systems.

Another possibility is that the cycle power generation of a system consisting of a MCFC/recuperated cycle gas turbine system with a direct-drive high-speed generator and power converter system. Fuel cells generally have poor load variation characteristics, while recuperated cycle gas turbines are sensitive to load variations. The performance of recuperated cycle gas turbines with conventional generators typically degrades under partial loads. In contrast, a recuperated cycle gas turbine with a direct-drive high-speed generator and power converter system can operate at the optimal rotation speed for any load. These improve performance under partial loads as well as the load-subsequent characteristics.



(i) MGT System Flow



(ii) MGT+MCFC (SOFC) Combined System Flow

Fig.20 A Schematic of MCFC (SOFC) and Recuperated MGT Combined System

SUMMARY

1. The RGT3R achieved 316kW-output power and the world highest 31.4% thermal efficiency at the generator end.
2. The RGT3R has successfully reduced emission levels of NOx, UHC, and CO to 20 ppm, 50 ppm, and 50 ppm or less, respectively, under conditions of 16%O_2 conversion and kerosene fuel burning.
3. Predicted durability of both turbine casing and the recuperator, which were evaluated by the most severest ASME N-47 design code, achieved the fatigue life time development target of 25 years and 3 years, respectively.
4. The design specification of a thermal efficiency is 32.5% and will be achieved by the first commercial engine production.

AKNOWLEDGEMENTS

We appreciate Dr. Hayashi and Mr. Yamada, senior researcher at the National Aerospace Laboratory of Japan, for their significant contributions during the development described here. We would also like to acknowledge all the individuals at our various partner companies, who made significant contributions to the RGT3R development project.

Lastly, we are pleased to add that as of February 3, 2003, the Power System Company of Niigata Engineering has been renamed Niigata Power Systems Co., Ltd. as a member of the IHI group.

REFERENCES

Kazuyuki Kobayashi, Tsuneo Igi, Hiroshi Kawamorita, Satoshi Izumi and Sougo Kuwabara, 1998, “Development of Small Size Radial Gas Turbine” 22nd CIMAC International Congress on Combustion Engines, Proceedings Vol.2, pp.491-500.