Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-45778

DEVELOPMENT OF THE 5MW POWER GENERATION GAS TURBINE ENGINE

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ABSTRACT
Gas turbine engine has been applied to the aircraft and ship propulsion with its advantages of compactness and comparatively short starting time. With a significant improvement in gas turbine efficiency with development of super alloy materials and advancement in cooling technologies in the second half of 1990s, its importance as a source of base load as well as peak load power generation has been increasing. However, with increased demand in nuclear power and renewable energy in the 21st century, there seems to be speculations among the power generation industries that gas turbine will take more or less a buffering role supplementing the irregular inflow of electricity to the grid rather than acting as a base load power source. With the shift in the role of gas turbine from base to supplementary, CHP application based on small powered gas turbine utilizing biogas or syngas as its fuel is expected to increase in the future. In this context, this paper describes the development result of 5MW gas turbine engine for CHP application. It can be operated with LNG or syngas of low LHV fuel. Originally, the engine was designed for LNG as its primary fuel, but since the importance of syngas power generation market will be increasing in the future, a complementary work for modification of combustor part has been carried out and has been tested. However, this paper deals with the parts developed with the use of LNG fuel. The test result of emission characteristics meets the standards required in Korea. The development has been made through the cooperation of Doosan Heavy Industry (DHI, Korea) and Zory-Mashproekt (Ukraine).

NOMENCLATURE

CHP	combined Heat & Power
HIP	hot isostatic pressing
I	moment of inertia, kg-m^2
k	specific heat ratio
LHV	low heating value, kJ/kg
LO	Scheokiometric value of fuel consumption
m	mass flow rate, kg/sec
n	frequency, RPM
OL	operation line
P	pressure (Pa), power, kW
Pr	pressure ratio
sec	second
SM	surge margin, %
TIT	turbine inlet temperature, K
w	correlated mass flow rate,
Greek	
α	air excess coefficient
η	efficiency
π	constant =3.14
τ	Time, sec
Subscript	
3	turbine inlet
04	turbine exit
aux	auxiliary
C	compressor
Cor	correlated
Cr	critical
IN	Inlet
Gg	gas generator
GT	gas turbine
T	turbine

INTRODUCTION

As the gas turbine efficiency is a function of compressor pressure ratio and TIT, the power increase with a higher TIT and pressure ratio is achieved, and thereby the tendency to use a large powered gas turbine of high TIT and pressure ratio became prevalent in power generation industry during the past century. The improving capability in flow path design, blade profile and heat resistant materials allowed to have higher TIT and pressure ratio which financially enabled small gas turbines to compete with large gas turbines of more losses in power transmission. Furthermore, as the gas price increase each year, the demands for the syngas produced from various wasted sources are increasing, and so is the research on small gas turbines. For the efficiency of gas turbine power plant, most of the operation regime depends on effectiveness of recovering the heat in the exhaust gas from the gas generator. Thus, in order to increase the overall efficiency during partload as well as nominal load in CHP application, it is necessary to maintain the exhaust gas temperature as high as possible. However, increasing the turbine exhaust temperature for a given TIT will decrease the turbine efficiency and thereby lead to the reduction in electrical efficiency of the CHP power plant [1]. Therefore, selection of the gas turbine performance will have to be made based on yearly demand characteristic between electricity and heat from the customer. In order to meet such a demand, it is necessary for gas turbine engine to have the capability of controlling the air mass passing through the compressor. In many cases of industrial gas turbines, either IGV or VGV is installed to monitor the air flow rate for bleed off or digression to other components for optimal operation purpose. Such a function in gas turbine is very important for avoiding the surge phenomena during operation [2], especially in starting regime. Such a controlling method makes it possible to maintain the turbine exhaust gas temperature constant for a given operation range and ambient temperature conditions. It is also very important to reduce NO_x and CO emission for gas turbines of CHP application as more and more strict regulations are being applied to power drivers such as gas turbine and others all over the world. Currently, emission less than 20ppm for NO_x is required for the metropolitan cities whereas 25 ppm of NO_x is being applied for other cities in Korea. CO emission less than 50 ppm is set for the development.

DESIGN FEATURES

COMPRESSOR

This paper is to introduce general development history of the engine and only overall design features of the components are presented. For the development of DGT-5 gas turbine engine, gas dynamic calculation for various meridional flow paths with different inlet diameter ratios and compressor pressure ratio is carried out and mixed type of 10 stages axial and 1 stage centrifugal compressor is designed. In order to obtain acceptable last axial stage blade height, constant hub diameter for the first 4 stages and mitigated tip speed of the first stage are selected. Selection of centrifugal compressor stage is designed to meet the condition that last axial stage blade height become less than 26mm. In this case, compressor length can be reduced with a centrifugal stage replacing 3 stages of the axial part. In order to reduce the effect of radial tip clearance in the centrifugal part, shrouded impeller is adopted. During the course of determining the flow path design of the axial part, the followings are taken into account [3];

- length of the axial compressor
- last stage blade height of axial compressor
- load coefficient of the axial stage

The given restriction in load coefficients together with axial velocity components of each stage and fairly up-to-date blade profiling technologies make it possible to achieve highly efficient design of the engine. Fig. 1 shows the general layout of the 5MW gas turbine engine DGT-5 which is being developed. The design point selected is considered fairly competitive compared with similar gas turbines in the market today. In Fig. 2, distribution of load coefficients along the stage is shown. A rather small value of loading coefficient in the first stage is applied to secure effective control of the compressor while starting. Table 1 shows the design point of the engine.

Table 1. Design point of the engine

Component		Value	Unit
Inlet	pressure	101.325	kPa
	temperature	288.15	K
Compressor	mass flow	20.9	kg/sec
	pressure ratio	14.0	
	exit temp.	664.5	K
	exit pres.	1.4112	MPa
Combustor	exit temp.	1100	^oC
	exit pres.	1.32	MPa
	pressure loss	6.0	%
	fuel rate	1191	kg/h
	efficiency	99.0	%
Turbine	exit temp.	480	^oC
	exit pres.	0.104	MPa
	Expansion ratio	12.7	
Rotor	rotation speed	12840	rpm
Performance	power output	5250	kW
	efficiency	32	%

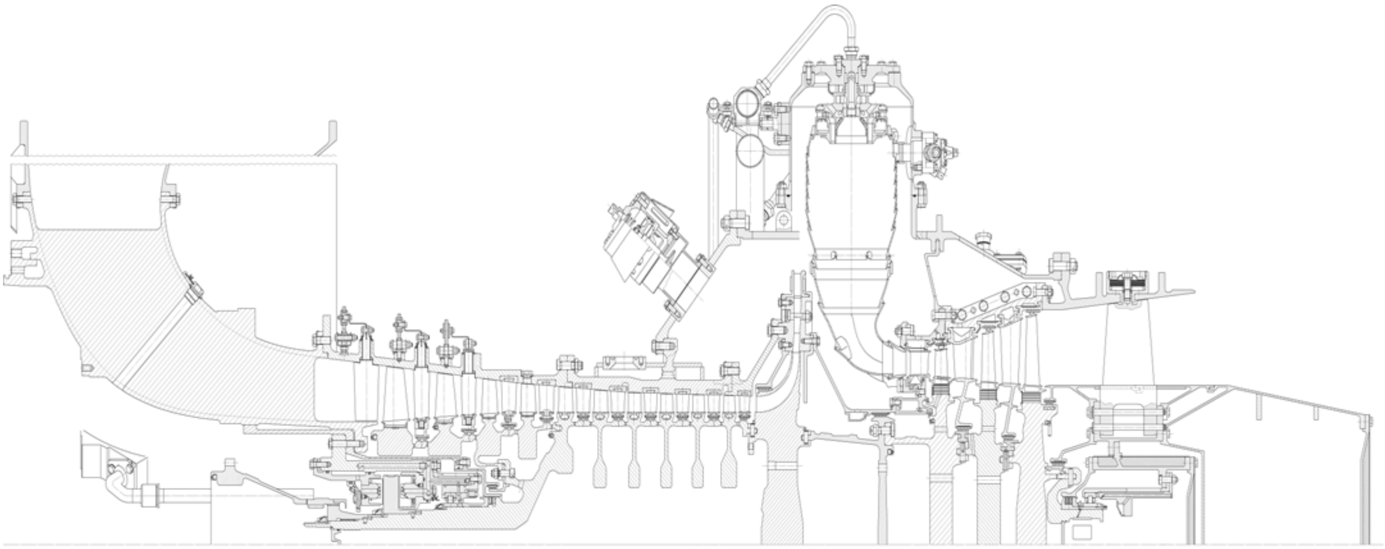

Figure 1. DGT-5 Gas Turbine Engine Layout

Selection of the rotating speed of the gas generator is made based on mass flow rate of the prototype model UGT-2500 gas turbine engine with stress limit in the centrifugal compressor taken into account and the equation is as follows [3],

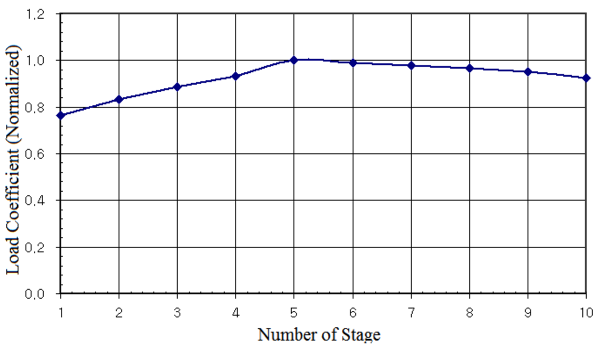

Figure 2. Load coefficient variation along the stages

Table 2. Input data for performance calculation

PARAMETERS	
Ambient temperature (K)	288.15
Compressor inlet pressure (kPa)	101.326
Compressor Pressure raise	14
Relative humidity (%)	60
Turbine inlet temperature (K)	1377
Inlet air mass flow rate (kg/sec)	21.35
Fuel low heating value (kJ/kg)	49300
Turbo compressor speed (RPM)	12840
Gear Box output power (MW)	5.25
Thermal Efficiency (%)	32.0

$$n_{5MW} = n_{2.5MW} \cdot \sqrt{\frac{m_{2.5MW}}{m_{5MW}}} = 12840 \text{ RPM} \qquad (1)$$

Based on thermal calculation with input data $m_a = 21.3$ kg/sec, $P_{in} = 100.8$ kPa, $T_{air} = 288$K, and with the rotation speed, inlet relative diameter of the 1st stage, the tangential velocity of the 1st stage rotor is set equal to U_{c1}= 380 m/sec. High tangential velocity at the tip of centrifugal stage impeller requires application of up-to-date technology for shrouded impeller manufacturing. The shrouded impeller is being manufactured by HIP method to be installed into the engine. Application of the shrouded impeller is estimated to bring a significant improvement in compressor efficiency of about 6%. Table 2 is the input data used for performance calculation.

COMBUSTOR

A combustor consists of central and peripheral swirl nozzles as well as ejector installed in the central swirl nozzle. Central swirl area and liner are designed based on the combustor prototype UGT15000 engine to utilize the stable combustion characteristics already proven through commercial operations [4]. The designed scheme of the combustor is shown in Fig. 3. Eight of can type reverse-flow combustors are installed in radial direction. Combustion liners have convective air cooling on external surface convective-film cooling on the internal surface. Transient piece is designed to provide smooth transition of hot gas flow from combustor exit to turbine with a minimum pressure loss.

TURBINE

Three stage axial type turbine is designed with due consideration of cooling system calculation. Secondary air system analysis is carried out to meet turbine performance with adequate air mass flow satisfying turbine parts strength limit. First stage rotor and stators, second stators are cooled by compressed air extracted from the compressor stages, and turbine rotor parts are of disk type design. In designing of the turbine geometry, meridional sections and number of stages, followings are taken into account;

- maximum possible efficiency of the stages
- convenient jointing with the combustor
- minimum possible number of turbine stages with minimal number of cooled blades and vanes.

In order to design a high efficiency turbine, a rather mild load coefficient distribution is adopted, something between 0.2 ~ 0.3. Based on the mean diameter flow path and given rotor speed (12840 RPM), three-stage turbine is considered to be optimum. In this way, meridional section of turbine is designed in such a way that maximum reduction of the axial velocity will be made to reduce the loss in the exhaust strut [5]. At this moment, the maximum possible 3rd stage turbine

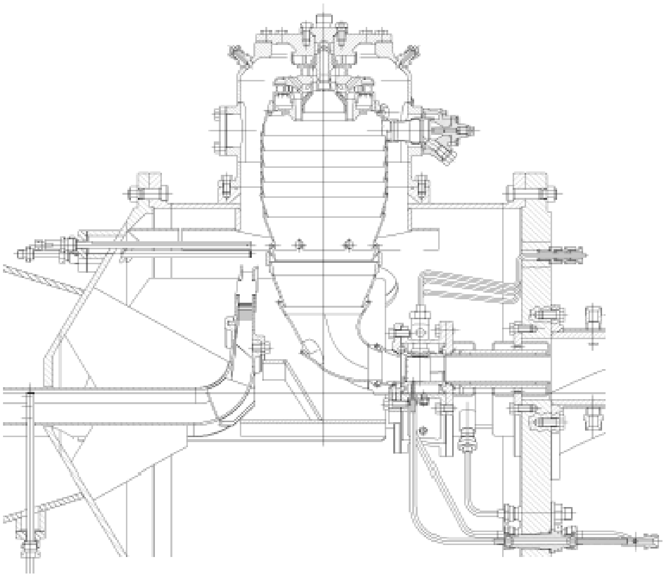

Figure 3. Layout of the combustor designed

blade height is possible with due consideration of the tensile strength exerted by centrifugal force. The maximum height of the 1st stage vane is limited by the exit flow angle. If this angle becomes too big, it will increase the tip loss and decrease in turbine efficiency as well. All cooled cascades were adopted with internal convective cooling. Thermal barrier coating was incorporated into the blades and vanes to ensure estimated parts life in nominal operation condition.

BEARING & GEAR BOX, STARTER

At this time of development, both front and rear bearings are of roller bearing type, but the front bear will be replaced with journal bearing in the commercial stage. Reduction gear box is employed to decrease the rotation speed of the gas generator and at the same time to increase the torque for transferring the power from the engine to the generator. The gear box is originally designed as 2 stages of 3 power transferring axial gears type. Starting power regime is estimated to confirm the starting time to the idle condition as well as to develop the control algorithm of starting system for the entire starting process. It takes about three minutes from initial start to the nominal rotating speed. During the starting period, the surge margin is calculated as,

$$\Delta SM = \left\{ \frac{(\mathrm{Pr}_c^{\ *} / m_{a.cor})_{surge}}{(\mathrm{Pr}_c^{\ *} / m_{a.cor})} - 1 \right\} * 100 \ \% \qquad (2)$$

where, $\mathrm{Pr}_c^{\ *}$, $m_{a.cor}$ are total pressure increase and mass flow
rate, respectively at correlated rotating speed. The equation representing the acceleration process is as follows [6];

$$I \frac{\pi}{30} \cdot \frac{dn_{gg}}{d\tau} = K \cdot \left(\frac{P_{st} + P_T + P_c + P_{aux}}{n_{gg}} \right) \qquad (3)$$

where, I is the total rotor moment of inertia of the gas generator rotor, n_{gg} is the gas generator frequency, P_T is the power generated in turbine, P_C is the power required from generator and gear box. This calculation result, through defining the possible operation range of the engine, will be able to provide input data for analyzing the structural and dynamic behavior and can also be used as a base material for the calculation of system control logic and, later, data for estimation of steam part of the cogeneration plant system.

PARTLOAD ANALYSIS

The steady state calculation of part-load condition is accomplished through application of a nonlinear mathematic model of different IGV conditions. Partload calculation of the DGT-5 for CHP application is carried out with the premise that a maximum possible exhaust temperature will be maintained for operation range possible. The maximum possible turbine exhaust temperature at nominal condition is kept to 480°C. The gas dynamic characteristic of axial-centrifugal compressor with $\Delta\alpha_{IGV}$ = -24 is shown in Fig. 4. In the figure, three thick lines represent engine operation lines for three cases of compressor inlet temperatures -45, 15, 45°C, respectively. The design point is located on the line of T_{IN}= 15°C which overlaps the speed line of RPM 12840. Relationship between compressor air flow rate and the inlet guide vane angles for various gas generator speeds is shown in Fig. 5. The load increase from engine start to possible turbine exhaust temperature is accomplished with IGV angle $\Delta\alpha_{IGV}$ = - 24° condition. Further increase in load is made by fuel and air flow rate control with IGV operation while the turbine exit temperature maintained less than 480°. Air mass flow rate increases with decrease in compressor inlet temperature and increase in air excess coefficient. The air excess coefficient is defined as follows,

$$\alpha = \frac{m_{air}}{m_{fuel} \cdot LO} \qquad (4)$$

The generator power output required for dry cranking is measured at gear box exit and is set to 50kW. The low heating value used in the calculation is 49,300kJ/kg [7] and the combustor efficiency was 99%. The steady state engine operation for different ambient condition is constrained by parameters as shown in Fig. 6. Power output of DGT-5 engine is limited to 5.25MW at gear box terminal with the compressor inlet temperature changes in the range -45 ~ 45^{o}C. The engine's operating range for compressor inlet temperature variation 15 ~ 45^{o}C is limited by gas turbine exhaust temperature (480^{o}C) and produces 5250MW at T_{IN} = 15^{o}C.

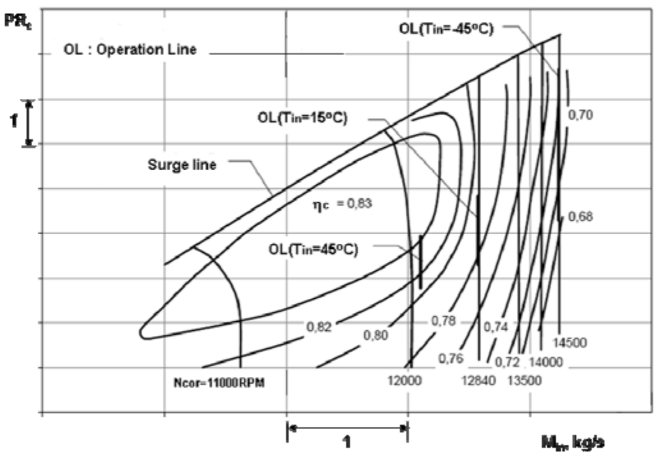

Figure 4. Axial-Centrifugal compressor characteristics of the DGT-5 gas turbine at IGV angle =-24

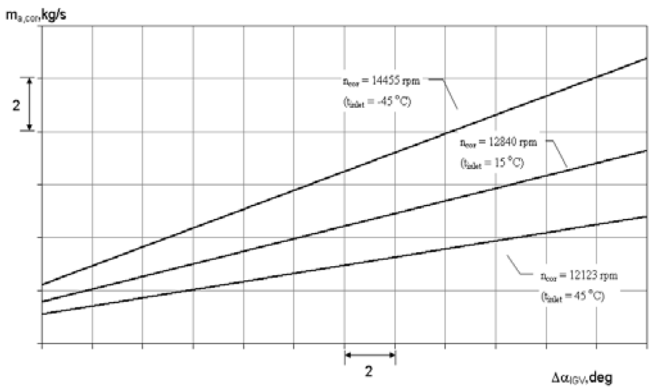

Figure 5. Relationship between correlated compressor air flow rate and Inlet guide vane angle $\Delta\ \alpha_{IGV}$ for different rotor speed and ambient conditions

This gas turbine power decreases to 3190kW as T_{IN} increases to 45^{o}C. Gas turbine power for the range of T_{IN} varying from -45 to 15^{o}C remains the same as 5.25MW. DGT-5 engine has a sufficient surge margin showing the safe operation characteristics. The surge margin is calculated using the equation (2). In Fig. 7, cycle thermal efficiency variation for different compressor inlet temperature is shown. As the load changes, the influence of compressor inlet temperature on thermal efficiency seems minimal. Fig. 8 shows the variation of turbine exhaust gas flow rate with load change. As seen in Fig. 5, turbine exhaust gas flow rate increases with decrease in compressor inlet temperature; however, the sudden increase of gas flow rate in each case is due to IGV control of air flow rate maintaining the constant turbine exhaust temperature according to governing control logic of operation. As seen in Fig. 9, the engine will produce higher power output with lower compressor inlet temperature at the same turbine exhaust temperature level.

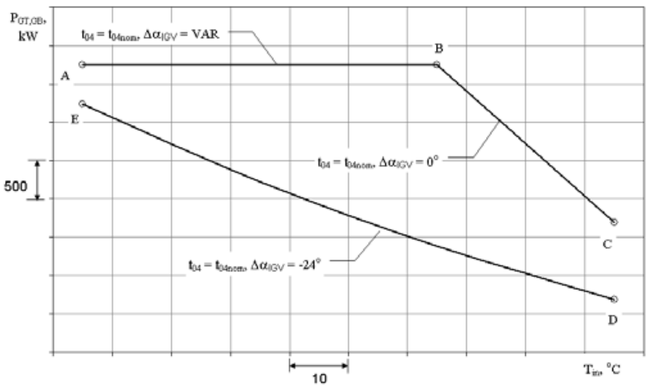

Figure 6. Variation of Power output (Gear box terminal) with compressor inlet temperature for different IGV angles

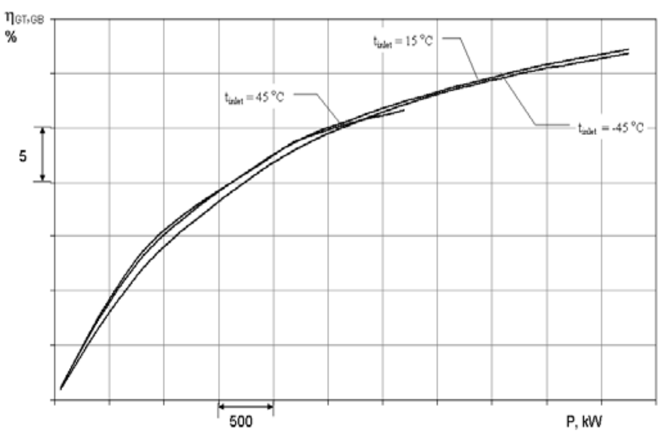

Figure 7. Relation between cycle thermal efficiency and power output for different ambient conditions

As turbine exhaust temperature reaches 480°C, power increase is made only through controlling the air mass flow rate by IGV/VGV operations. This allows the same heat rate for CHP application.

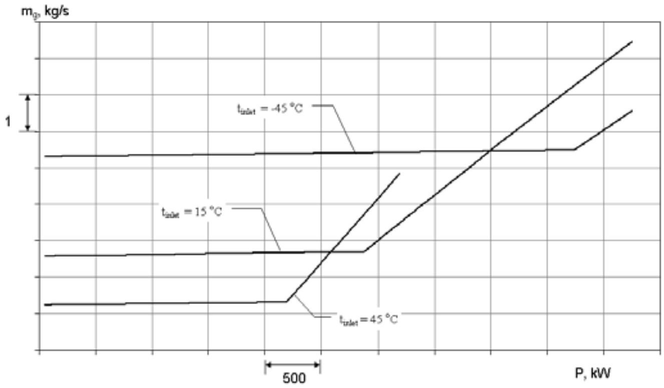

Figure 8. Variation of turbine gas flow rate with power output for different ambient conditions

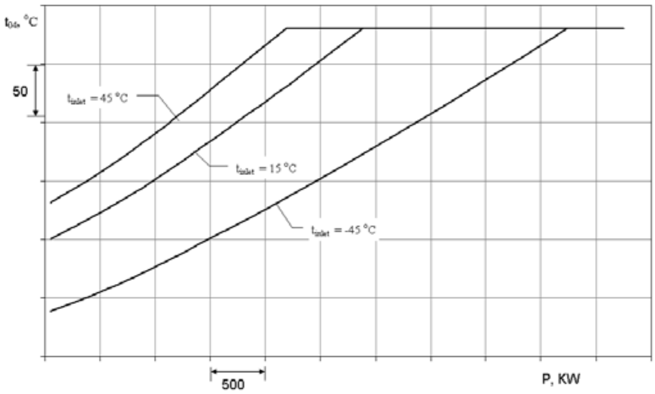

Figure 9. Turbine exit temperature variation with power output for different compressor inlet temperature ($\Delta\ \alpha_{IGV}$= varying)

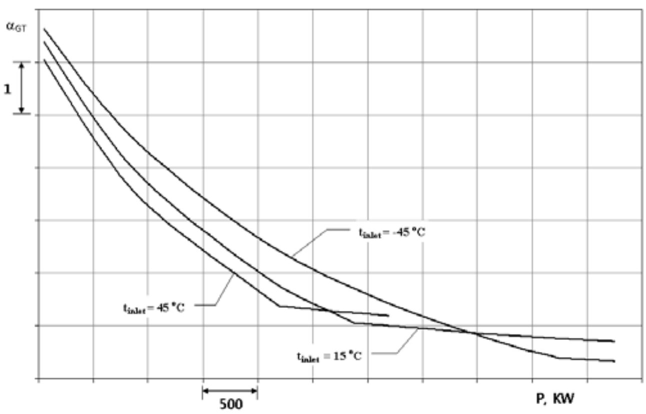

Figure 10. Air excess coefficient variation with power output for different compressor inlet temperature ($\Delta\ \alpha_{IGV}$= varying)

The air excess coefficient decreases significantly with load increase and is affected by the compressor inlet temperature reaching a minimum value near nominal load and compressor inlet temperature -45°C as seen in Fig. 10. While the engine's operating regime is controlled by IGV operation, the turbine inlet temperature remains the same. While the engine exhaust temperature varies from idle to 480°C, the engine is operated with IGV closed state, that is, $\Delta\alpha_{IGV}$= - 24. The engine's operating regime ABCDE in Fig. 6 is possible while maintaining the turbine exhaust temperature constant with IGV control[8]. Maintaining the nominal turbine exhaust temperature at ambient temperature to 15°C is possible through VGV control which enables the DGT-5 power output to vary between the range 0.55 ~ 1.0.

TEST RESULT

As mentioned in the introduction, throughout the development project four gas turbines were to be manufactured and two of them will be wasted in the process to confirm the design parameters through experiments. Third and fourth of the gas turbines will go through endurance tests after modifications are made to the engines. The fourth engine is scheduled to be installed in the biogas power plant at Kimpo, Korea to accumulate field test experience. Performance test of the first engine was done from November 2009 until April. 2010. The second engine had been tested from May 2 until September 2010. This paper shows the test result of the #1A engine which was tested from April 2010 through September 2010. The test result of power output was a bit lower than the target showing 4.97 compared with 5.25 MW of the target at ISO condition. After the test, a thorough investigation of the result found out that the defects came from several different locations such as centrifugal compressor part, diffuser, and combustor casing. Initially, the centrifugal part of the compressor was to be equipped with shrouded impeller, but, due to manufacturing difficulties involved with such a large outer diameter close to 600mm

Figure 11. Performance test facility of the 5MW gas turbine engine

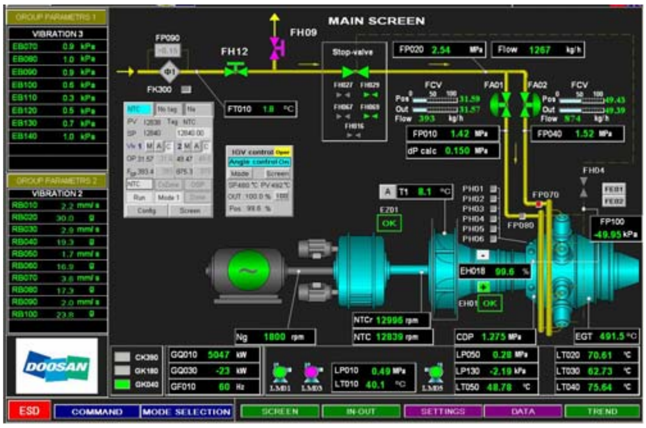

Figure 12. Automatic Control Panel (ACS) for DGT-5 Engine Performance Test

and its machining, unshrouded impeller was installed temporarily and tested. Currently, manufacturing of the shrouded impeller with HIP (Hot Isostatic Pressing) method is in progress and as the manufacturing is completed, the unshrouded will be replaced with shrouded and it is expected that deficiencies related to compressor will disappear. In Fig. 11, performance test facility of the 5MW gas turbine engine is shown. Fig. 12 shows the control panel to monitor the engine test. The power generated from the engine is consumed in the load bank.

Fig. 13 shows the engine's starting characteristics during the initial 60 seconds. Dotted line indicates the starting command schedule and real lines indicate various cases of starting conditions. Fig. 14 shows the results of full speed non load (FSNL) test. The surge margin for maximum permissible operable regime is measured as ΔSM= 13.1%. The designed surge margin of the compressor at nominal condition is estimated as ΔSM = 12%. Fig. 15 and 16 are load test results from where TIT and thermal efficiency were estimated to be compared with the designed.

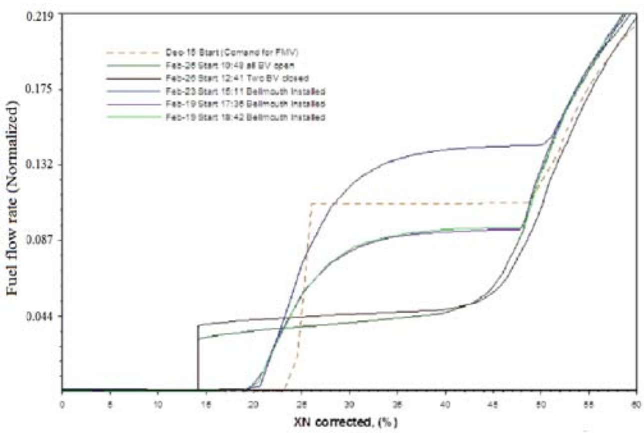

Figure 13. Test results of engine starting characteristics

As shown in Fig. 15, though the turbine inlet temperature for low speed region appeared to be higher than the designed, they are still in the range permissible in terms of metal temperature. As was explained above, the thermal efficiency turned out to be less than the designed due to the employment of unshrouded impeller during the test and improper design between compressor diffuser exit and combustor inlet. The decrease in compressor air flow rate is considered to be the major reason for decrease in power of the engine. It is considered that air flow rate generated with IGV and VGV operation is not enough to meet the target. The increase of inlet pressure loss to 350mm H_2O is also the factor in reduction of the performance and is caused by the installment of flow measurement devices to measure the compressor inlet air flow rate. The reduction in compressor efficiency 3.5% seems to be caused partly by the tip clearance increase at the tip of unshrouded impeller and loss characteristics there in order to improve noticed defects[9]. The test with shrouded impeller is scheduled in April 2011.

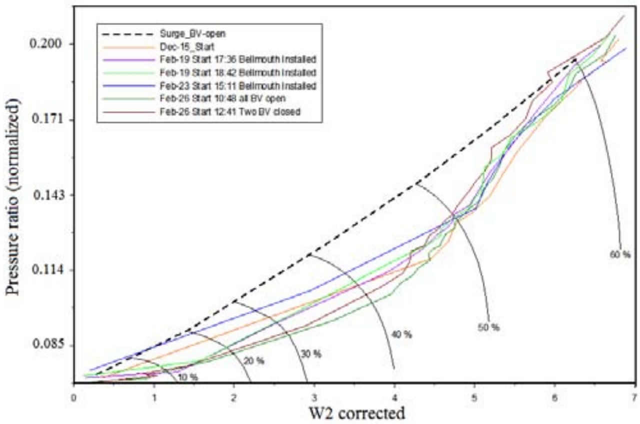

Figure 14. Test results of Full Speed Non-Load (FSNL) engine operation

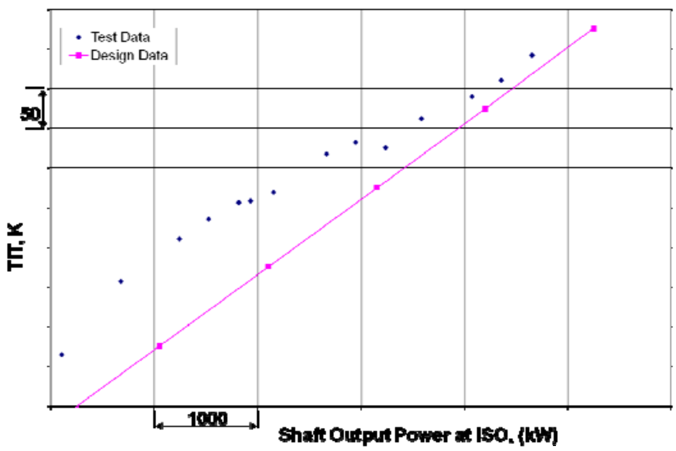

Figure 15. Engine test data for TIT increase with power output

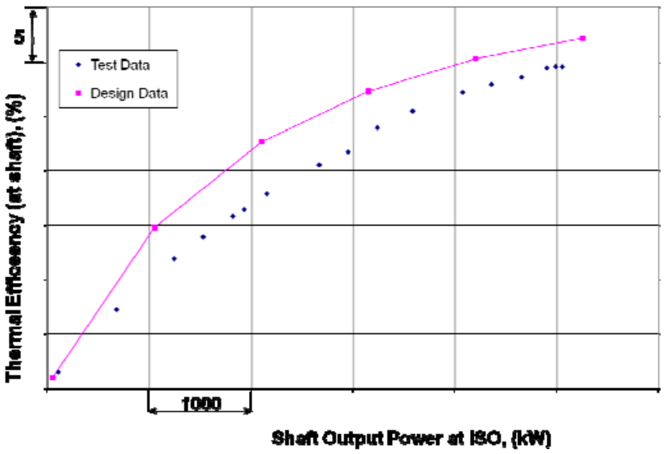

Figure 16. Engine test data for thermal efficiency vs. power output

CONCLUSION

This paper introduced the development results of 5MW gas turbine engine for power generation. From the test result of the developing gas turbine engine, the followings are noticed;

1. it is necessary to measure the flow rate to better define the turbine flow rate,
2. the precise measurements of the turbine flow passage area and compressor to better define the air leakage during the operation regime will be necessary,
3. the application of the shrouded impeller is considered essential to meet the target performance of the engine.
4. the design modifications to reduce the total pressure losses occurring from compressor diffuser exit to rhe first stage turbine inlet must be examined thoroughly and measures to overcome these obstacles should be taken.

ACKNOWLEDGMENT

This work is subsidized by Korea Institute of Energy Technology Evaluation and Planning (KETEP) and its support is greatly appreciated.

REFERENCES

[1] Philip P. Walsh and Paul Fletcher, 1998, “Gas Turbine Performance”, Blackwell Science.

[2] H. Cohen, GFC Rogers and HIH Saravanamuttoo, 2001, “Gas Turbine Theory”, 5th ed, Pearson Prentice Hall

[3] N. A. Cumpsty, 2004, “Compressor Aerodynamics”, Krieger Publishing Company.

[4] B.E. Spitsyn, et al., 2006, “Technical Report of UGT-5000-02PP, Nikolaev, Zorya-Machproekt.

[5] B.E. Spitsyn and A.A. Filonenko, et al., 2006, “Calculation of steady state performance of gas turbine unit GTD-5000 (DGT-5). Unpublished report.

[6] K.V. Holshevnikov, 1970, “Theory and calculation of Aviation Gas Turbine Engine”,.

[7] J.L Kerrebrock, 1992, "Aircraft Engines and Gas Turbines", 2nd Ed., MIT press.

[8] Kurzke, J., 1995, "Advanced User Friendly Gas Turbine Performance Calculations on a Personal Computer", ASME paper GT-147.

[9] J.D. Mattingly, W.H. Heiser and D.T. Pratt, 2002, “Aircraft Engine Design”, AIAA Education Series.

