Al-Khwarizmi Engineering Journal, Vol. 9, No. 4, P.P. 54- 64 (2013)

Al-Khwarizmi Engineering Journal

The Power Enhancement of a Mini-Gas Turbine by Adding Ethanol to the Compressor Inlet Air

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(Received 10 October 2013; accepted 17 December 2013)

Abstract

An experimental study is conducted on the utilization of the inlet ethanol injection technique in order to evaluate its impact on the performance of a two-shaft T200D mini-gas turbine engine. The maximum degradation recorded in power output was 32.8% at the climate temperature of 45°C. Nevertheless, at that temperature, adding ethanol with Eth/LPG ratio of 20% by volume brought an enhancement in power output of 19.2% compared to normal LPG run. SFC of the dual-fuel engine ranked a level of 22% higher than that with pure LPG consumption. The overall efficiency suffered a maximum reduction of 14.4% with Eth/LPG fuel ratio of 20%, but when the loading was raised beyond 70% of the engine full load; the efficiency of dual-fuel engine excelled the normal LPG engine by 3.1% at the corresponding Eth/LPG ratio maintaining a power turbine speed of 12000 rpm.

Keywords: *Alcohol gas turbine, Inlet cooling, Alternative fuels, Bio-fuels.*

1. Introduction

The vast urban constructions accompanied by the development in commercial and industrial sectors led to a continuous growth in power demand during the last two decades in Iraq. This situation requires urgent remedies to come out of the electricity and energy demand crisis.

The 1973 oil crisis in the Middle East caused a global search for alternative fuels that could replace petroleum-based fuels. Utilization of biofuels with fossil fuels is a more economic option to provide renewable power. And, much like the automotive industry, the power industry will need to institute design changes to accommodate these fuels [1]. The gas turbine power plants have found increasing service in the last forty years due to the great amount of energy they produce for their size and weight. Gas turbine compactness, low weight, flexible load accommodation, start-up easiness, and multiple fuel applications make it a natural power plant for off-shore platforms [2].

All of the current liquid biofuels used or being considered exhibit a lower heating value less than standard petroleum-based fuel ; thereby requiring a step increase in fuel flow to deliver the same power output. Ethanol has already been tested in gas turbines and found to be an attractive renewable fuel option. Its lower lubricity and low heating value, though, require changes to most standard gas turbine fuel and combustion systems [3].

Ethanol has been known as a fuel for many decades since Henry Ford expected that ethanol, made from renewable biological materials, would be a major automobile fuel. However, it is not widely used because of its high price. In Brazil, ethanol has been used as automotive fuel since 1925, due to its large territorial areas utilised for sugar cane planting. This motivates the automotive industry to start the development of engines and components adequate for this source of energy [4].

A land based gas turbine for power generation has the characteristics that the output falls on hot summer days when the electricity demand peaks. This is because the high temperature causes air density to be less, reducing the mass flow of compressor intake air [5].

There are certain methods to reduce gas turbine inlet temperature, e.g. water evaporation, adsorption and mechanical cooling. This work focuses on the injection of liquid ethanol directly into the inlet air making use of its high latent heat of vaporization to be absorbed from the air itself causing its temperature to decrease. The reduction in inlet air temperature will in turn increase its density, as well as, its mass flow rate and hence, boost the power output.

Ethanol itself could be considered as an organic fuel that contributes to the heat addition to the engine. Therefore, it may replace the petroleum fuel in purpose of improving the environment impact. Ethanol injection into engine inlet air has the advantage of avoiding the harmful effects of ethanol on seals and gaskets of conventional fuel system if straight blended with the petroleum fuel.

2. Literature Survey

One of the first studies on using alcohol in gas turbine engines was conducted by National Advisory Committee for Aeronautics NACA [6]. The investigation was carried out on a 4000 lbf thrust centrifugal flow type turbojet engine at maximum rotor speed of 11500 rpm by injecting water-ethanol mixture at compressor inlet. The maximum thrust augmentation of 26% was obtained by the injection of 31% alcohol by weight. The fuel flow and the tail-pipe gas temperature was the same as for no injection run.

A blend of ethanol/Jet A fuel in varying concentration ranging from 25 to 100% by volume was tested in a small-scale 30 kW gas turbine. The performance characteristics including the thrust, thrust specific fuel consumption were compared with that of Jet A over a range of throttle settings. The results indicate that Ethanol-Jet A blend is a viable alternative fuel for gas turbine engines [2].

It is a common practice to use excess air to control the combustion temperature. But, this excess air could be reduced if methanol vapour reformed by the turbine wasted heat is injected into the combustion gas [7]. This scheme would increase the net power output of the plant. When the time comes that allows us to use turbine inlet temperatures of 1,400^{o}C, the proposed cycle could achieve approximately 60% overall thermal efficiency.

The air, as auxiliary medium is replaced with ethanol-water compound steam that produced with waste heat of a Capstone C33 gas turbine. Ethanol mass flow was increased up to 10% presenting the expected cooling on the inlet temperature [8]. But, when more ethanol is injected it cause a reduction in efficiency due to the unburnt ethanol that leaves with the exhaust gases. The steam of ethanol-water compound will not absorb significant amount of heat inside the combustor and lowering the flame temperature as the mixture of air and evaporated ethanol does.

The results of performance testing of a 30 kW gas turbine engine supplied with biodiesel from palm oil, soy and ethanol showed that the latter have a 65% higher SFC than diesel fuel. Ethanol showed the lowest efficiency among the fuels tested as a decrease of about 10% with respect to diesel was recorded [9].

General Electric (GE) Energy's Aeroderivative class of gas turbines has tested ethanol among many other biofuels to evaluate its sustainability as fossil fuels replacement, and the extent of environmental impact resulted from that conversion [3]. Tests indicated a need for 30% increase in fuel flow concerning the combustion chamber and injectors redesign for burning the much lighter fuel. GE Co. has been contracted by Petrobras to convert one of the two units at a power plant near Rio de Janiero to burn ethanol [10]. A 43.5 MW, LM6000 gas-fired plant is believed to be the first power plant in the world powered by ethanol. The turbine burns 18000 litres per hour if operated continuously at base load capacity. The results were very satisfactory and very equivalent to that of any other liquid fuel. Nevertheless, the cost of compatible dual-fuel turbine is between 5 to 10% more than a gas-only turbine considering the additional cost to install liquid fuel system to the gas turbine.

3. Thermodynamic Background

It is known from Carnot cycle that the maximum efficiency of a heat engine is determined by the average temperature of heat addition and heat rejection. In this case, we focus on reducing the inlet temperature of a gas turbine, which represents the lower cycle temperature. Hence the cycle efficiency will increase because the average temperature of heat rejection has decreased. Referring to Figure (1), due to the divergence of constant pressure lines on the (***T-s***) diagram, the final delivery air temperature will reduce as the compression commences from lower intake air temperature like T_a instead of T_1. Consequently, the required compression work is reduced as the delivery temperature reduced. Considering the overall pressure ratio yields [11];

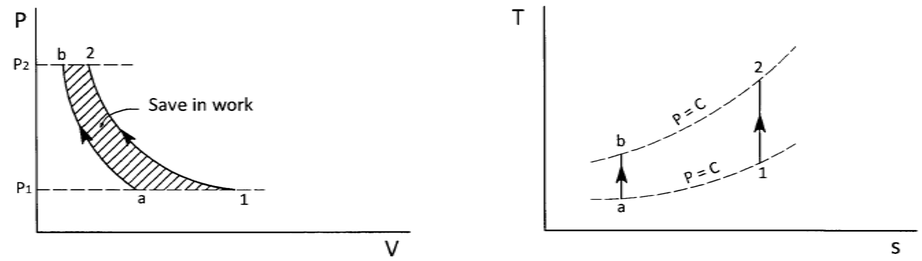


Fig. 1. Effect of Inlet Temperature on the Compression Work.

$$W_c = C_p \frac{T_{01}}{\eta_c}\left[\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \qquad ...(1)$$

This obviously shows the great effect of the inlet air temperature on the reduction of the compression work required. Hence, the intended ethanol injection in the present work should decrease the compression power, as well as, increase the power output contributable to the increase in air mass flow resulting from the density improvement due to the inlet cooling.

4. Cycle Analysis

The gas turbine performance is to be analysed by evaluating the cycle parameters which affected by the addition of ethanol to the inlet air. These parameters are to be used in demonstrating engine performance with the dual fuel as will be shown later in the results section.

The air flow rate measured is calculated by the equation [12];

$$\dot{m}_a = 0.01027\sqrt{\Delta h}\,\sqrt{\rho_1} \qquad ...(2)$$

then, this rate should be corrected as;

$$\dot{m}_{a,C} = \dot{m}_a\left(\frac{1.013}{p_a}\right)\sqrt{\frac{T_a}{288.15}} \qquad ...(3)$$

The ethanol fuel flow is calculated by the rate equation;

$$\dot{m}_{Eth} = \rho_{Eth}\left(\frac{V_{Eth}}{\theta}\right) \qquad ...(4)$$

then this rate is also corrected as;

$$\dot{m}_{Eth,C} = \dot{m}_{Eth}\sqrt{\frac{288.15}{T_{Eth}}} \qquad ...(5)$$

also, the main LPG mass fuel flow rate as measured directly, needs to be corrected as [12];

$$\dot{m}_{LPG,C} = \dot{m}_{LPG}\left(\frac{f_p}{1.581}\right)\sqrt{\frac{288.15}{T_{LPG}}} \qquad ...(6)$$

where, f_p is a pressure based correction factor. Hence, the total amount of dual fuel consumed is given by;

$$\dot{m}_f = \dot{m}_{LPG,C} + \dot{m}_{Eth,C} \qquad ...(7)$$

At this stage, it is now possible to determine the heat liberated by the burning of the dual fuel as;

$$\dot{Q}_f = \dot{m}_{LPG,C}\,H_{LPG} + \dot{m}_{Eth,C}\,H_{Eth} \qquad ...(8)$$

In order to estimate the performance criteria of the cycle, it first needs to determine the power produced by the engine which is an electric power calculated by;

$$\dot{W}_E = I \times \Phi \qquad ...(9)$$

therefore, the specific fuel consumption of the cycle is given by;

$$SFC = \frac{\dot{m}_f}{\dot{W}_E} \qquad ...(10)$$

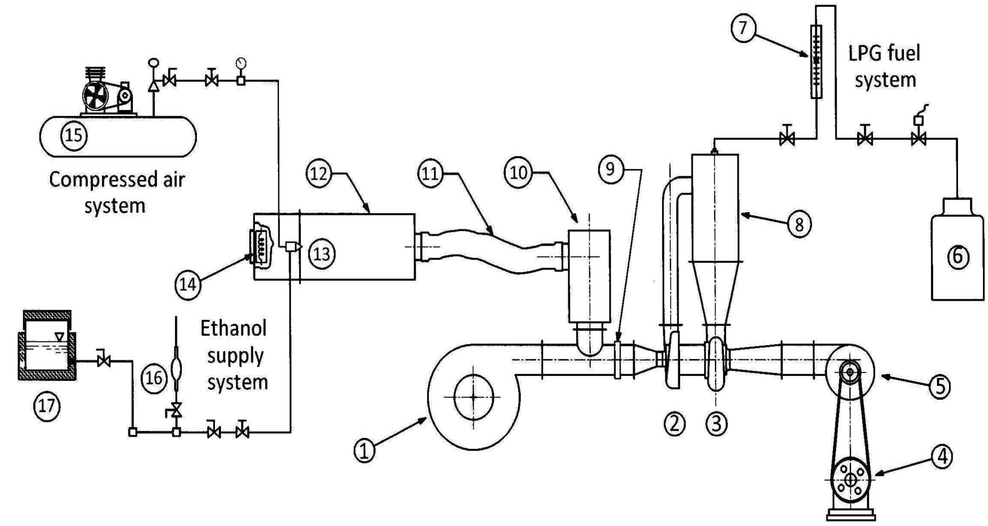
and the overall efficiency is determined as;

$$\eta_{ov} = \frac{\dot{W}_E}{\dot{Q}_f} \qquad ...(11)$$

5. Experimental Set-Up

5.1. The Test Rig

In purpose of carrying out the cooling needed to reduce the temperature of the gas turbine inlet air, a test rig has been constructed specially to introduce ethanol in the way of the incoming air such that increasing its mass flow by increasing its density.



1	Starting blower	7	LPG fuel flow meter	13	Ethanol atomiser
2	Gas generator compressor	8	Combustion chamber	14	Air pre-heater
3	Gas generator turbine	9	Orifice air flow meter	15	Compressed air plant
4	Dynamometer	10	Admission box	16	Ethanol flow meter
5	Power turbine	11	Flexible conduit	17	Ethanol reservoir
6	LPG fuel vessel	12	Treatment duct		

Fig. 2. Experimental Set-Up Showing the Spraying System Attached to the Gas Turbine.

The experimental rig shown schematically in Figure (2) is mainly a duct that inducing the ambient air inside to pass across the sprayed ethanol source and driving the treated air down to the gas turbine inlet. At the end of the spray chamber, an air assisted atomiser was installed to create a mist of very tiny ethanol droplets which increase the surface area of the liquid exposed to the air in order to enhance the mass and heat exchange process.

A galvanised sheets box of 30 cm cube is used to store the ethanol and it has an external lagging to avoid any heat interactions with the ambient. A glass bulb flow meter is used to measure the time needed to consume 10 ml of ethanol from the bulb, see Figure (3).

The ethanol atomiser has an orifice diameter of (1.2 mm) , and it is working at a safe pressure of (4 bar). The atomization air is generated by a *LIBA* two-stage reciprocating compressor works under a maximum pressure of 10 bar, having a 100 litres accumulator.

The gas turbine used in this study is a Two-Shaft T200D mini-gas turbine unit fabricated by *Didacta Italia*. The combustion chamber burns the *LPG* gaseous fuel that ignited by a spark plug. Ethanol needs to be introduced in such a way that no disturbance is created within the air inflow. This task could be satisfied by means of inlet injection as ethanol easily mix and penetrate the bulk flow of air. The attachment of the gas turbine unit with the spraying system is done by adding the “Admission Box” which encloses the machine inlet matrix to decelerate and separate the non-evaporated droplets escaping out of the treatment duct before entering the unit, see Figure (4).

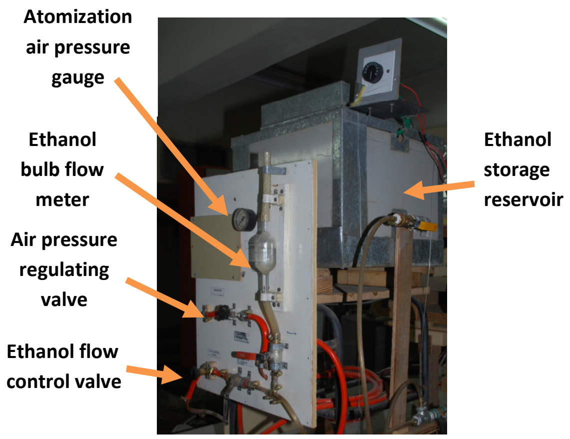


Fig. 3. Ethanol Reservoir with Flow Meter and Control Valves.

As shown in Figure (5), the T200D unit is equipped with a synoptic panel contains the main measuring instrument that built in it and covering the vital parts of the gas turbine cycle. The pressure gauges are all mechanically operated with a range of (0 to 10 bar) for high pressure side, while for low pressure side after the combustion chamber the gauges work with a range of (0 to 600 mbar). The temperatures are all digitally displayed with signal comes from probes inserted into air path around the cycle. The both turbines rotational speed are indicated by digitally displaying tachometers. The dynamometer output is indicated digitally in terms of electric current and voltage developed by the alternator connected to the power turbine via a toothed belt assembly. This output represents the electric power produced by the unit. The main LPG fuel has a mechanically operated pressure gauge indicating the supply pressure with a range of (0 to 4 bar), as well as, a rotary float meter measuring the LPG mass flow rate up to 3 g/s.

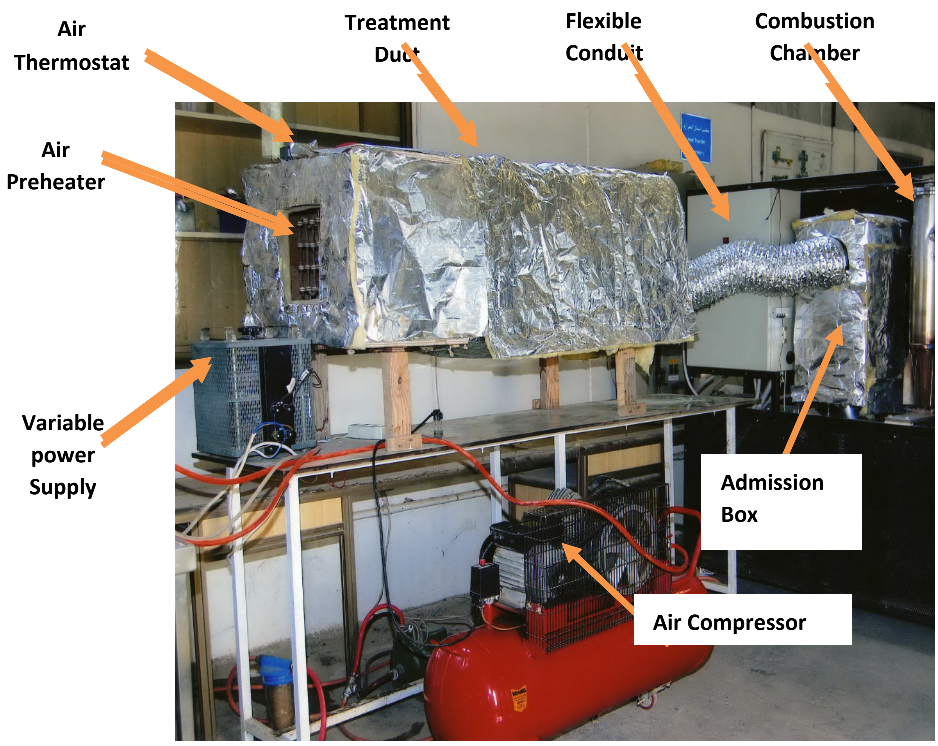


Fig. 4. Set-up of the Ethanol Fuelled Mini-Gas Turbine.

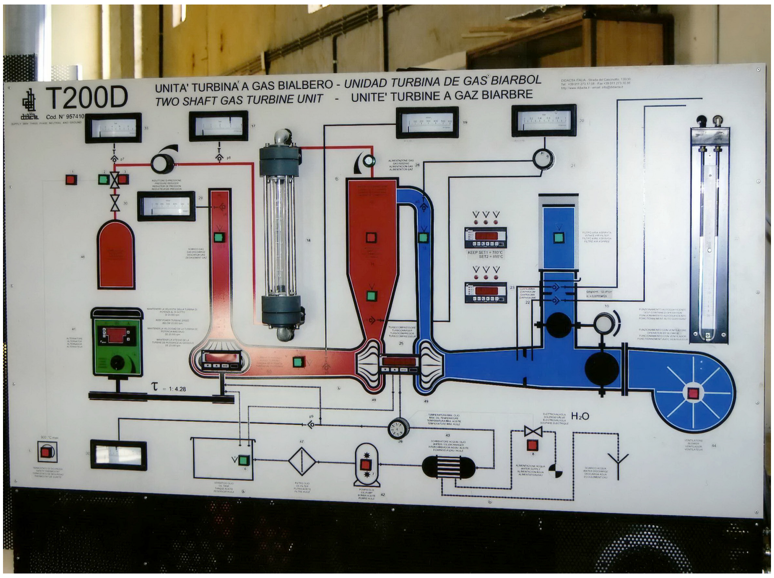


Fig. 5. Synoptic Panel of the T200D Mini-Gas Turbine.

A 6000W air electric heater has been installed at the entrance of the spray chamber to control the induced air temperature. The electric heater is controlled by a thermostat with operating range of (30 to 120^{o}C). The thermostat is activated by a sensing bulb placed prior to the ethanol atomiser. A variable power supply unit *Ogawa Seiki* model SAT-2030 with maximum current of 30 Amp is used to supply the exact value of electric current needed to establish the intended inlet air temperature. This temperature is measured by an intelligent *LUTRON* model YK-2001 TM professional, type K, thermometer.

The study comprises the performing of experiments on the gas turbine unit combined with the ethanol injection system in purpose of tracking the changes in the operational parameters of the gas turbine cycle as ethanol being sprayed in the intake air of the unit.

5.2. Testing Procedure

The tests are initiated with running the machine on the conventional LPG fuel in the sequence described in reference [12]. Each test is targeted to examine the engine performance under the intended operating condition. Switching to dual fuel run needs firstly to reduce the LPG flow rate in accordance to the prescribed volume ratio. Secondly, the ethanol fuel is to be supplied into the inlet air in the required volume ratio and brought the ethanol vapour into the combustion chamber. Nevertheless, this process is very difficult since the two fuels are come from separate streams such that the combustion chamber acts as a mixing box for them. A good approximation is to deliver the two fuels in amounts determined on basis of the heating value to compensate the reduction in the heat liberated due to lower ethanol heating value. However, practically this is not enough because of the unexpected losses accompany the real engine run. Slight changes in both fuels flow rates must be done very slowly to keep the volume mixing ratio while attaining the intended condition with the adequate total mass flow of the dual fuel.

Due to this prolonged complicated process, some deviations from the nominated volume ratios are to be accepted, although, these deviations were actually marginal. The maximum deviation recorded in the worst cases was on the lower limit of nominal 10% ethanol which gave a ratio of 9.74% representing a deviation of – 2.6%. On the other hand, the upper limit of nominal 20% ethanol in dual fuel gave a ratio of 21.2% representing a deviation of + 6.1%. However, all other tests were accomplishing volume ratios with deviations never exceeded $\pm$ 1.5%.

6. Results and Discussion

Experiments were first conducted at five different inlet air temperatures namely, 23, 27, 35, 40 and 45^{o}C in order to demonstrate the impact of the air ambient temperature on the cycle performance. These tests were carried out under a constant load of 65% of the engine full load.

Increasing inlet air temperature lowers the air mass flow due to the density reduction as shown in Figure (6). The air mass flow has an average decrease of about 11% as the air temperature increased to 45°C for all gas generator speeds.

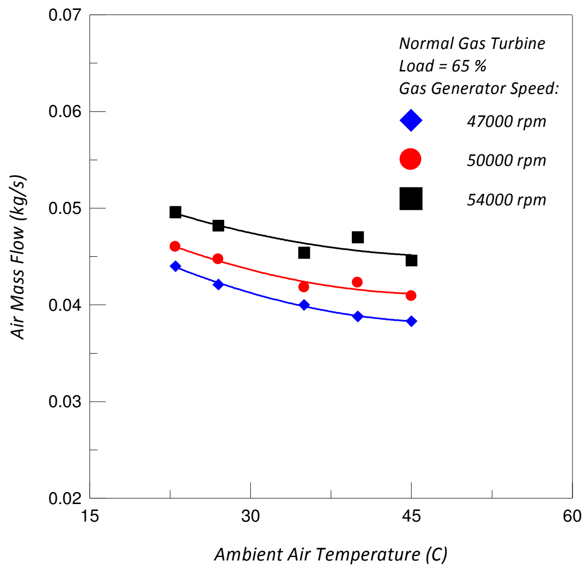


Fig. 6. Effect of Ambient Air Temperature on the Induced Air Mass Flow.

The effect of the ambient temperature on the power output is clearly described in Figure (7). The output degrades with increasing ambient temperature due to the increase in compression work caused by higher delivery pressure and temperature. The maximum degradation in power was 30.2% of the power developed at 23°C for corresponding ambient temperature of 45°C measured at a gas generator speed of 47000 rpm. When the gas generator speed rose to 54000 rpm; the maximum reduction in power became 32.8% for the same corresponding ambient temperatures.

Figure (8) indicates the decrease in overall efficiency with higher ambient temperature resulted from the reduction in power output. The maximum reduction of 25% was recorded at ambient temperature of 45°C with 47000 rpm, nevertheless, when gas generator speed reached 54000 rpm; the reduction became 20% as the heat rate was reduced avoiding the limit of inlet turbine temperature.

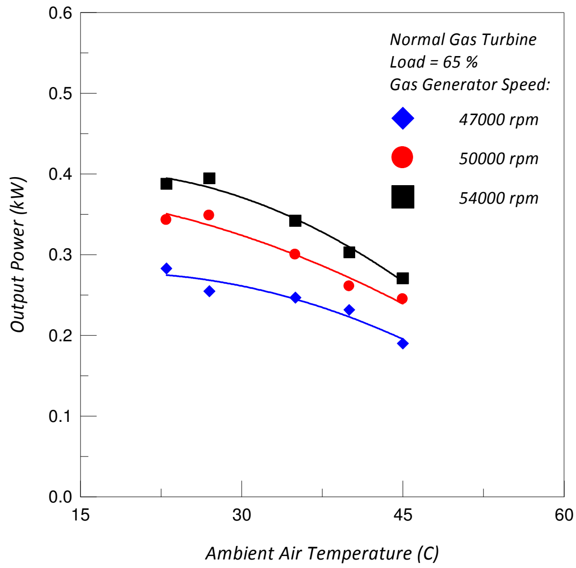


Fig. 7. Effect of Ambient Air Temperature on the Engine Power Output.

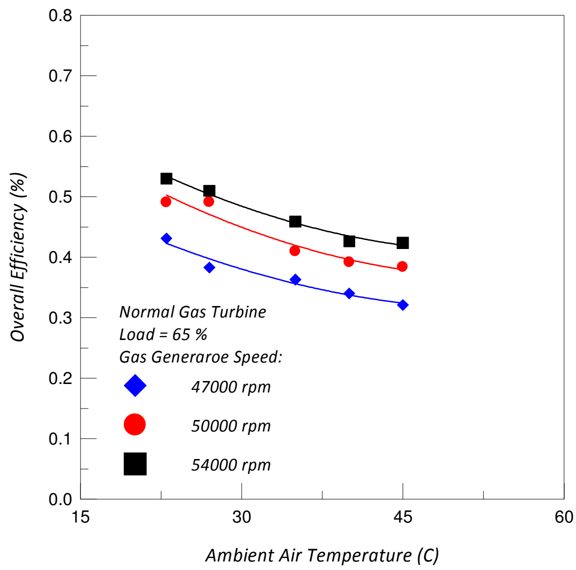


Fig. 8. Effect of Ambient Air Temperature on the Cycle Overall Efficiency.

When detecting the effect of inlet ethanol injection on the cycle performance, experiments were conducted at two Eth/LPG fuel ratios of 10 and 20% by volume with air under 45°C. The tests carried out under the same loading level of 65% on the engine as explored in Figures (9) to (11).

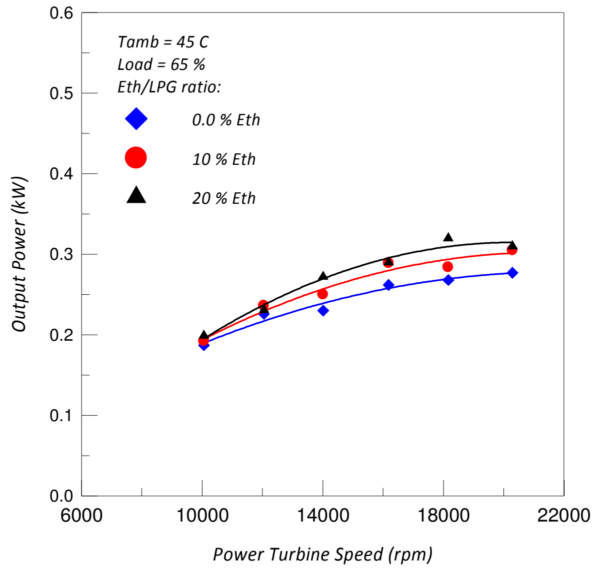


Fig. 9. Relationship between Engine Power Output and Power Turbine Speed at Different Eth/LPG Ratios.

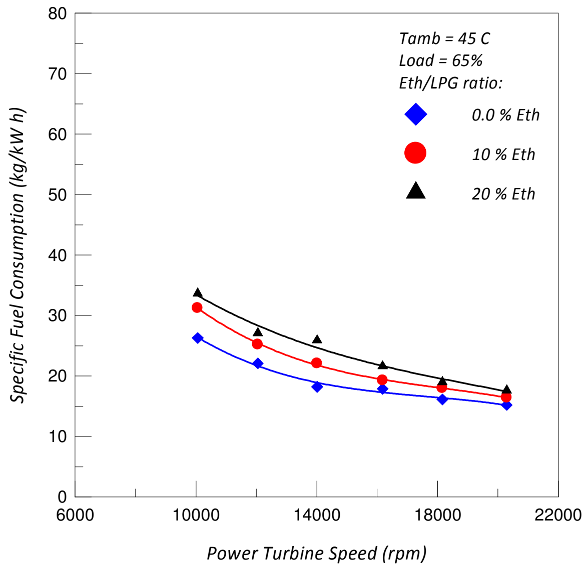


Fig. 10. Relationship between Specific Fuel Consumption and Power Turbine Speed at Different Eth/LPG Ratios.

The evaporation of injected ethanol brings the inlet air temperature to be reduced, hence, increasing the density at the compression commencement. This effect will certainly decrease the delivery air temperature, as well as, the work needed for the compression. Therefore, increasing ethanol injection rate is coupled with increasing the power output of the engine as a result of the higher mass flow, see Figure (9). At 45^{o}C ambient air temperature the ultimate power enhancement recorded was 6.4% with Eth/LPG ratio of 10%. But when the ratio has increased to 20%, the enhancement reached 19.2% compared to that of normal engine run on LPG alone.

With the increased ethanol content in the compressed air, the SFC worsens as more fuel should be supplied to maintain the same power output, due to the lower heating value of ethanol in the fuel mixture. The deterioration in the SFC ranked a level of 13.4 and 22% for Eth/LPG ratio of 10 and 20% respectively, compared to the pure LPG consumption, as clarified in Figure (10).

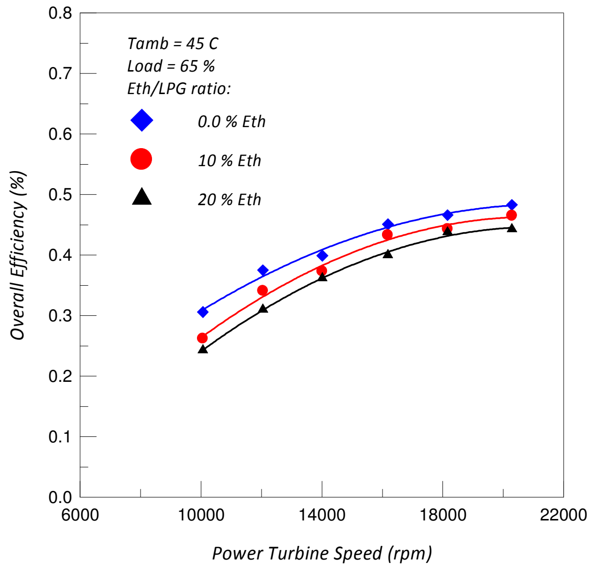


Fig. 11. Relationship between Overall Efficiency and Power Turbine Speed at Different Eth/LPG Ratios.

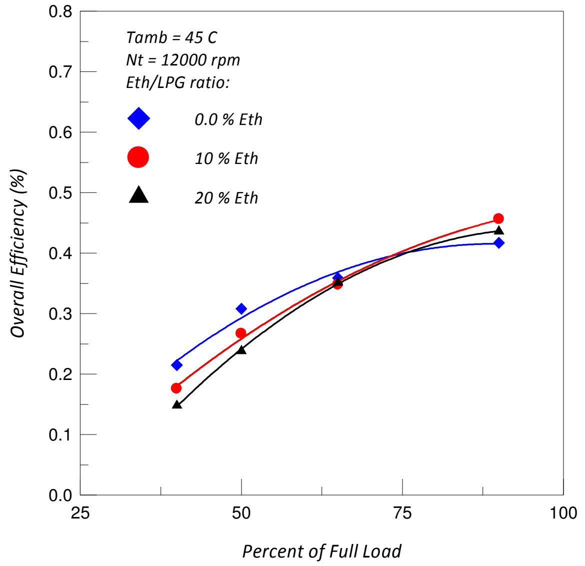


Fig. 12. Effect of Engine Loading on Overall Efficiency at Different Eth/LPG Ratios.

The lower heating value of ethanol requires more fuel to be burned, thus, more energy to input and that reflects on the overall efficiency as shown in Figure (11). The average reduction in efficiency was about 9% for the Eth/LPG ratio of 10%, whilst, for the ratio of 20%; the reduction reached 14.4% with respect to pure LPG run.

Understanding the impact of engine loading on the dual-fuel gas turbine, requires tests to be conducted at four steps of loading namely, 40, 50, 65 and 90% of the engine full load. Those were executed to maintain a constant power turbine speed of 12000 rpm.

Figure (12) shows the variation of overall efficiency with engine loading at air ambient temperature of 45°C. Like before, at light and medium loading range, the efficiency has reduced with increasing Eth/LPG ratio. But, when going towered the higher engine loading with more fuel consumption, the overall efficiency was slightly improved due to the extra ethanol in compressed air, whose contains oxygen atom within its chemical structure that help aiming complete fuel combustion. The improvement in efficiency was 6% with Eth/LPG ratio of 10%, nevertheless, at a ratio of 20% the improvement is only reached 3.1% when going beyond 70% of the full engine load.

7. Conclusions

- The air mass flow has an average decrease of about 11% as the air temperature increased from 23 to 45°C for all gas generator speeds.
- With the ambient temperature increased from 23 to 45°C, the maximum degradation in power output was 32.8% when speed reached 54000 rpm. At that speed, the overall efficiency suffered a maximum reduction of 20% with respect to normal LPG run.
- Adding ethanol with Eth/LPG ratio of 20% by volume made an enhancement in the power output of 19.2%, while overall efficiency went down by 14.4%. This ethanol concentration brings an increase in SFC of 22% compared to that of LPG normal run, all under 45°C air temperature.
- Raising the engine loading beyond 70% led the dual-fuel engine to excel the normal LPG engine by 6 and 3.1% at Eth/LPG ratios of 10 and 20% respectively.

Nomenclature

Symbols

C	Specific heat capacity	kJ/kg K
f	Correction factor	-
h	Manometer reading	mm H_2O
H	Lower Heating Value of fuel	kJ/kg
I	Electric current	Amp
$\dot{m}$	Mass flow rate	kg/s
N	Rotational speed	rev/min
p	pressure	kN/m^2
$\dot{Q}$	Heat power	kJ/s
s	Specific entropy	kJ/kg K
T	Absolute temperature	K
v	specific volume	m^3/kg
V	Volume	m^3
W	Specific work done	kJ/kg
γ	Ratio of specific heats	-
η	Efficiency	-
θ	Time	s
ρ	Density	kg/m^3
Φ	Electric Voltage	Volt

Subscripts

a	Air
amb	Ambient condition
c	Compressor
C	Corrected value
E	Electric
f	Fuel
g	Gas generator turbine
ov	Overall
p	Constant pressure process
t	Power turbine
0	Stagnation condition
1	Compressor inlet
2	Compressor outlet

Abbreviations

BMEP	Brake mean effective pressure
Eth	Ethanol
GE	General Electric Company
LPG	Liquefied Petroleum Gas
NACA	National Advisory Committee for Aeronautics
SFC	Specific fuel consumption

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Appendix

Specifications of T200D Twin-Shaft Gas Turbine Unit [12]

Compressor	Centrifugal
Turbines	Radial inflow
Power output	1.5 kW @ N_g/N_t : (55000/21000) rpm
Compressor Speed	60000 rpm max.
Free turbine Speed	23000 rpm max.
Overall Pressure ratio	1.3 / 1 max.
Firing Temperature	700 – 750^{o}C max.
Fuel	LPG, 3 g/s max. @ 2 bar max.

تعزيز القدرة الخارجة لوحدة توربين غازي مصغرة بأضافة الأيثانول الى الهواء الداخل للضاغطة

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الخلاصة

تم إجراء دراسة تجريبية حول إستغلال تقنية حقن الأيثانول في هواء الدخول بهدف تقييم أثرها على إداء ماكنة توربين غازي ثنائي المحور T200D . أقصى تردي للقدرة الخارجة سجل بحدود 32.8% عند درجة حرارة محيط قدرها $45^{o}C$. أضافة الأيثانول عند درجة الحرارة تلك بنسبة حجمية أيثانول\غاز قدرها 20% يجلب زيادة للقدرة الخارجة بحدود 19.2% مقارنة مع الماكنة التقليدية. إستهلاك الوقود النوعي لماكنة الوقود- الثنائي حققت مستوى 22% أعلى من إستهلاك الغاز الصرف. الكفاءة الأجمالية تعاني أنخفاضاً أقصى قدره 14.4% عند نسبة أيثانول\غاز 20% ، لكن مع زيادة التحميل لمافوق 70% من حمل الماكنة الكلي؛ فأن ماكنة الوقود- الثنائي تتفوق على الماكنة التقليدية بتحسن كفائتها بمقدار 3.1% لنسبة خلط الوقود تلك للحفاظ على سرعة ثابتة لتوربين القدرة مقدارها 12000 دورة\دقيقة.