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# Performance Study of an Aero-Derivative Gas Turbine System with a Coalescer Filter at Varying Operating Conditions

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**Abstract**— The performance of an aero-derivative industrial gas turbine (ADIGT) with a coalescer filter at varying operating conditions is presented. The study determines the aptness of an ADIGT for power generation in two extreme locations. The ADIGT was modeled to operate at conditions of Usan offshore oilfield and Maiduguri desert in Nigeria. For all operating conditions of decrease TET, corresponding decrease in AFR was observed. For ambient temperatures between 15 and 38$^{O}$C in both environments the AFR, OPR, and IPD declined to above 0.5 %. Equally, for conditions below ISO and 1.8 % drop in TET, a 2.4% loss in AFR and 1.6% gain in OPR were attained. The estimated IPD and AFR values exist at about 90% validity with the experimental. However, for ADIGT to be suitable for power generation in these locations, it will require system modification and the use of inlet air pre-cooling system to bring the ambient air close to ISO.

**Index Terms**— Filter, Gas turbine, Performance, Pressure drop.

## 1 Introduction

The principal function of the inlet filter system in a filter housing of a gas turbine (GT) is to prevent ingestion of contaminants. The aerodynamic design of the air inlet filter and the duct system of the filter housing plays a significant role in allowing the unconstrained flow of air to the axial compressor. However, this invariably impacts the volumetric flow rate as well as the pressure drop across the filters. Modern aero-derivative industrial gas turbines (ADIGT) are designed for space reduction and high power output. Hence, the role of the air filter becomes noteworthy. Adequate inlet air filtering system enhances the GT performance, promotes the GT longevity and reduces maintenance cost, as well as environmental pollution [6]. Furthermore, high air swallowing capability and large contaminants retention are likewise the attributes of a good filter system.

Contaminants have the tendency to get covered on the aerofoil of the compressor causing erosion on the compressor blades, thus leading to high-pressure drop [19]. This pressure drop can be controlled by using filters that are capable of upholding the power developed by the GT over a long term service [6], [17]. The latter could be determined by evaluating the behaviors of filters when applied to industrial GTs operating at different locations at some engine conditions [8]. Also, a coalescer filter is occasionally used as a first stage filter or a second stage filter and can capture particles greater than 0.01 micron.

Other high efficient filters based on classification include EPA, ULPA and HEPA with efficiencies between 85 and 99 % (DEN, 2009).

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In determining the performance of GT for a particular environment, the theoretical analysis may require a combination of different levels of filters for best performance which is normally accompanied by high-level simulation with environmental conditions as the reference. Studies have revealed that as the filter performance or efficiency improves, the rate of dust penetration reduces, resulting in a decrease in the air flow pathway. Consequently, this results to increase in pressure loss due to the air restriction caused by high dust retention in the filter [7]. Moreover, if the compressor is to function effectively, it must overcome losses due to filter pressure drop in the system. The consequence of these losses is low turbine output and increase in the heat rate. Therefore, it is imperative that filter engineers must ascertain the needed pressure loss and efficiency for their application [13].

In Nigeria, for instance, where this study is domiciled. The total gas turbine installed capacity in the energy utility sector in 2012 stood at 6000 MW, with an actual generating capacity of about 4000 MW [2], [3], [6]. Several, technical discrepancies exist in the sector ranging from poor maintenance, low plant performance, and high losses. Some of these technical problems are ascribed to inadequate viability study afore plant installation as well as miscalculations based on system design specifications. Similarly, issues connected to plant design, economic estimation, efficient energy use and energy conversion methods have been viewed significantly in this period of scarce fossils fuel availability [2]. Therefore, adequate information on environmental conditions favorable for GT performance improvement is apt. As this will actuate effective measures to reduce the cost of maintenance, improve efficiency, and remove undue downtime. This has to be so since appropriate operating and design parameters will be upheld.

The primary goal of this study is to model the performance of an ADIGT with a coalescer filter, required to operate in two extreme environments, Maiduguri desert and Usan offshore oilfield in Nigeria. The specific goals include:

- To monitor the performance of the ADIGT at varying temperature conditions with a coalescer filter.

- To evaluate the effect of TET on indicators like thermal efficiency, OPR, power output, AFR and fuel flow rate at varying temperature conditions.
- To evaluate the effect of IPD on inlet AFR at different TET settings

## 2 METHODOLOGY

The GT simulation was carried out with TURBOMATCH, simulation software. The created engine model for an industrial ADIGT was similar to GE LM2500+ GT depicted in Fig. 1. Furthermore, the procedures involved iterative steps associated with accurate predictions to ensure consistency of the variables with the matching constraint [18]. The performance data at ISO condition for the two locations (Table 1) were used to create the engine model [6]. The design point model for off-design performance scenario was adopted. The model engine produced is a single shaft ADIGT with an integrated free power turbine of output 30.2 MW and efficiency of 39 %. Also, the engine comprises an intake system, a compressor, double (Premas) that bleeds 10% of compressed air used for turbine blade cooling. It also has a burner where combustion takes place, (Mixees) that absorbs the bled air used for cooling and a convergent nozzle.

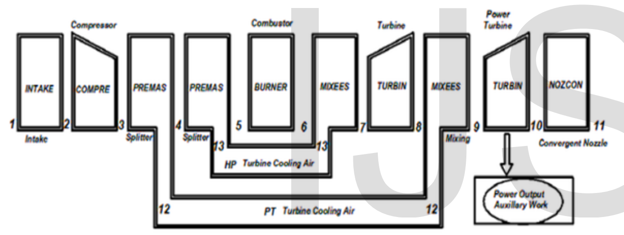


Fig. 1. TURBOMATCH GT Engine Model Schematics for GE LM2500+.

### 2.3 GT Model Formulation

The following assumptions were made following the steady energy equation to allow for thermodynamic balances in the control volumes. Compression and expansion processes are isentropic. The working fluid is considered to have ideal characteristics and with constant composition throughout the control volume. Constant-pressure specific heat data for the constituent gases occur as a polynomial function of temperature [14].

$$\sum \dot{m}_{in} - \sum \dot{m}_{out} = 0 \quad (1)$$

$$\sum (\dot{m}_{in} h_{in} - \dot{m}_{out} h_{out}) + Q + W = 0 \quad (2)$$

$$\bar{C}_{p,k}(T) = E + FT^{a} + GT^{b} + HT^{c} \quad (3)$$

Where, E, F, G and H are constants. The thermodynamic balances for the different control volumes are considered as follows:

INTAKE

The inlet pressure recovered (Figure 1) from the intake duct for use in the compressor is given by the following expression [6].

$$P_{rc} = \frac{\Delta P}{\ } = \frac{(P_{amb} - P_{f\,out})}{\ } = \frac{(P_1 - P_2)}{\ } \quad (4)$$

Where, $P_{amb} = P_1$ is the ambient pressure; $P_{f\,out} = P_2$ is the outlet pressure of the filter given by

TABLE 1
SUMMARY OF INPUT DATA FOR OFF-DESIGN PERFORMANCE [6]

| Condition | Usan Offshore oilfield | Maiduguri desert |
|---|---|---|
| Minimum ambient temperature (°C) | 22 | 17.7 |
| Average ambient temperature (°C) | 26.3 | 25 |
| Maximum ambient temperature (°C) | 30 | 38 |
| Altitude above sea level (m) | 30.48 | 353.8 |
| Total initial pressure loss (Pa) | 582.3 | 442.3 |
| Total final pressure loss (Pa) | 1661.82 | 1341.82 |
| Initial Pressure recovery (%) | 99.43 | 99.56 |
| Final pressure recovery (%) | 98.36 | 98.67 |

$$P_2 = P_{f\,out} = P_{rc} \times P_{amb} \quad (5)$$

"COMPRE" is a brick performance code that represents the compressor. The exit pressure of the compressor is given by [6].

$$P_3 = PR_c\, P_2 \quad (6)$$

The compressor outlet temperature ($T_3$) can be evaluated using the isentropic relationship.

$$T_{2s} = T_2 PR_c^{\left(\frac{\ }{\gamma_{air}}\right)} \quad (7)$$

Equally, the compressor isentropic efficiency ($\cap_c$) is defined by the expression

$$\cap_c = \left(\frac{\text{isentropic work of the compressor}}{\ }\right) = \left(\frac{h_1 - h_{2s}}{\ }\right) = \left(\frac{T_1 - T_{2s}}{\ }\right) \quad (8)$$

Based on expressions (7) and (8), $T_2$ can be evaluated as,

$$T_2 = T_1 \left| \left( \frac{(PR_c)^{\gamma_{air}} - 1}{\ } \right) + 1 \right| \quad (9)$$

Applying expression (9) to the GT model in Figure 1 and using the GT notations, the compressor outlet conditions are given by:

$$T_3 = T_{c\,out} = T_{f\,out} \left| \left( \frac{(PR_c)^{\gamma_{air}}}{\ } \right) + 1 \right| \quad (10)$$

$$P_3 = P_{c\,out} = P_{f\,out} \times PR_c \quad (11)$$

In the compressor the heat input $Q$ is zero, thus the compressor work ($CW$) can be obtained from expression (2) as:

$$CW = m_{air}\, C_{p_{air}} (T_{c\,out} - T_{f\,out}) \quad (12)$$

PREMAS

Assumption: 10 % of the compressed air flowing into the combustor was used to cool the turbine blades. PREMAS 3-4, (Figure 1) represents 2 % bled air used to cool the free power turbine blades while PREMAS 4-5 represents 8 % bled air used to cool the high-pressure turbine blades [4], [11]. Hence the airflow is balanced applying station notations as follows:

$$m_3 = m_{air} \quad (13)$$

$$m_4 = 0.98 \times m_{air} \quad (14)$$

$$m_{12} = m_{cool1} = 0.02 \times m_{air} \quad (15)$$

$$m_5 = m_{comb\,in} = 0.92 \times m_4 \quad (16)$$

$$m_{13} = m_{cool2} = 0.08 \times m_4 \quad (17)$$

BURNER

The expression for the combustor outlet temperature (COT) follows as:

$$COT = T_6 = \left| FAR \times FCV \times \frac{\eta_{comb}}{\ } \right| + T_{c\,out} \quad (18)$$

$T_{c\,out}$ is the fraction of compressed air exiting at compressor

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maximum temperature. The parameters $FAR$, $FCV$ and $T_{c\ out}$ accounts for the total COT. However, 7.5 % pressure loss is assumed in the combustion chamber [4]. The expressions for the combustion outlet pressure (COP) and combustor heat input ($Q_{in}$) are given by:

$$COP = P_6 = 0.925 \times P_5 \tag{19}$$

$$Q_{in} = m_{gas}(T_6 - T_5) \tag{20}$$

MIXEES 1

At the MIXEES 1 (Figure 1), we assumed 2 % bled air [6], with outlet conditions specified as:

$$T_7 = T_{mix1} = T_6 - T_{11} \tag{21}$$

$$T_{11} = T_{cool1} = 0.08 \times T_{c\ out} \tag{22}$$

$$P_7 = P_{mix1} = P_6 \tag{23}$$

TURBIN 1

At the TURBIN 1, the same isentropic correlation is applied and with steady state energy flow assumptions, the outlet conditions are expressed,

$$T_8 = T_{t1\ out} = T_{mix1} - \left(\frac{CW}{\;}\right) \tag{24}$$

$$P_8 = P_{t1\ out} = P_{mix1}\left|1 - \left|\frac{\left(T_{mix1}\right)}{\;}\right|\right| \tag{25}$$

MIXEES 2

Outlet Conditions:

$$T_9 = T_{mix2} = T_8 - T_{12} \tag{26}$$

$$T_{12} = T_{cool2} = 0.02 \times T_{c\ out} \tag{27}$$

$$P_9 = P_{mix1} = P_8 \tag{28}$$

TURBIN 2

Outlet Conditions:

Assumption: 2 % pressure loss assumed in the exhaust nozzle.

$$P_{10} = P_{t2\ out} = \frac{P_{amb}}{\;} \tag{29}$$

$$T_{10} = EGT = T_{mix2}\left|1 - \left|\eta_{t2}\left(1 - \left(\frac{P_{t2\ out}}{\;}\right)^{\overline{\gamma_{gas}}}\right)\right|\right| \tag{30}$$

At TURBIN 2, the heat input is zero, and the burnt gases are assumed perfect gas while expansion process is reversible adiabatic. Thus from the steady state energy flow assumption and applying this to Figure 1, the auxiliary work produced (useful work) by the GT and the overall thermal efficiency are:

$$PTW = AW = m_{gas} C_{pgas} (T_{mix2} - EGT) \tag{31}$$

$$\eta_{th} = \frac{m_{gas} C_{pgas} (T_{mix2} - EGT)}{\;} = \frac{AW}{\;} \tag{32}$$

## 2.3 GT Off Design Performance and Filter Pressure Drop

In evaluating the GT performance, off design data were used for both Usan offshore oilfield and Maiduguri desert. The inlet AFR was obtained first using the off design engine data at different power settings. The data for the coalescer filter contained in [9] were applied in expression (33) to estimate the inlet AFR. While expression (34) was used to evaluate the initial pressure drop (IPD) at ambient conditions of 15ºC ≤ $T_{amb}$ ≤ 30ºC.

$$AFR = \rho_{air} . A . V_{air} \tag{33}$$

$$IPD = \frac{F_d}{\;} \tag{34}$$

Where, $\rho_{air}$, $V_{air}$ and $F_d$ are air density, velocity of air and drag force determined as:

$$M = \frac{V_{air}}{\;} \tag{35}$$

$$F_d = \frac{C_d\, m_{air} . V_{air}}{\;} \tag{36}$$

Where, M is the Mach number and $V_s$ is the speed of sound [16].

TABLE 2
COMPARISON OF REAL PERFORMANCE DATA WITH SIMULATED RESULTS FOR GE LM2500+ [9]

| Parameter | GE LM2500+ | | |
|---|---|---|---|
| | ER | SR | D % |
| Power Output (MW) | 30.2 | 30.2 | 0 |
| Thermal Efficiency (%) | 39.0 | 38.65 | 0.8 |
| Compressor Pressure Ratio | 23.1 | 23.5 | 1.7 |
| Exhaust Gas Flow (kg/s) | 85 | 82.81 | 2.5 |
| Exhaust Gas Temperature (K) | 791.48 | 782.02 | 1.2 |

$$V_s = (331.1 + 0.606 T_{amb}) \tag{37}$$

## 3 RESULTS AND DISCUSSION

Table 2 depicts the simulated results with the experimental data obtained from the literature for the aero-derivative engine model. The results show a close match with that at ISO conditions. The deviations in the power output and thermal efficiency were between 0 and 0.8 %, while the deviations for other parameters were not greater than 3 %. The model was thus adopted for off design calculations.

Fig. 2 presents the sensitivity results for different operating conditions at temperatures between 15ºC ≤ $T_{amb}$ ≤ 38ºC. The results show that for 1.8 % reduction in TET (Fig. 2a) from design point (DP), leads to 2.3 % decrease in the inlet AFR. For the temperature range of 15ºC ≤ $T_{amb}$ ≤ 38ºC in both environments and 1.8 % decrease in TET, the inlet AFR decreases and varies from 2.7 % to 10.8 %. Furthermore, for same engine conditions and ambient temperature range of 15ºC ≤ $T_{amb}$ ≤ 38ºC (Fig. 2 b). The OPR decreases and lies between 2.2 ≤ OPR ≤ 6 %. At conditions below ISO ($T_{amb}$ ≤ 12ºC), the GT witnessed a reduction in AFR by 2.4 % with a 1.6 % increase in OPR.

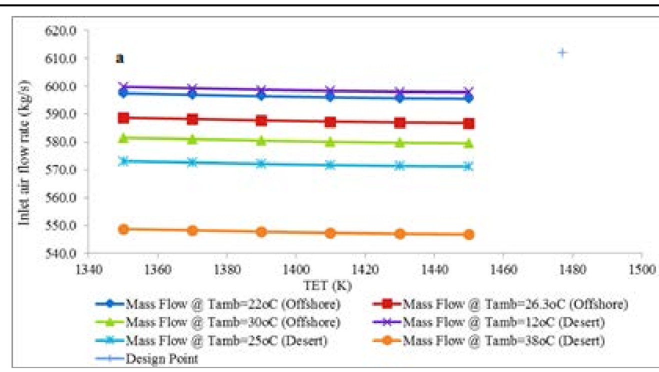


Fig. 2a. . Effect of TET and ambient temperature on inlet air flow rate

Fig. 2(c) to (e) presents the effect of TET oFigure.2(c) to (e) presents the effect of TET on fuel flow rate, power output, and thermal efficiency. For 8.6 % reduction in TET a corresponding decrease in fuel flow rate; shaft power output; and thermal efficiency exist. The values ranged between 22.6 ≤ $m_{fuel}$ ≤ 32.3

%, 33 ≤ Power ≤ 46.3%, and 14% ≤ $\eta_{th}$ ≤ 20.8 respectively. Similarly, at temperature conditions slightly below ISO 15°C (ISO) to $T_{min}$ ≤ 12°C lead to a decrease in fuel flow rate and an increase in OPR. The latter accounts for the reduction in shaft power and thermal efficiency observed in (Fig. 2d and 2e).

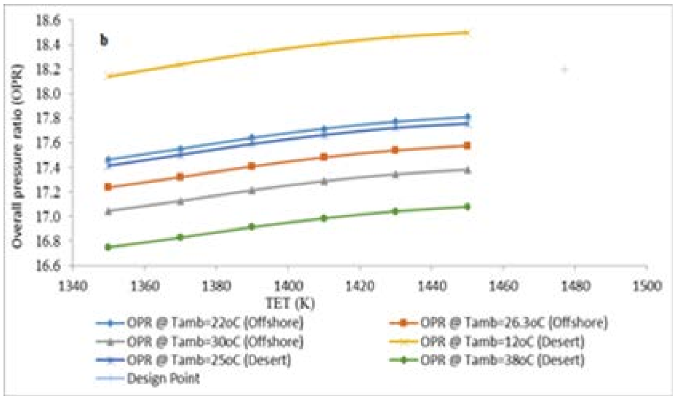


Fig. 2b. . Effect of TET and ambient temperature on overall pressure ratio

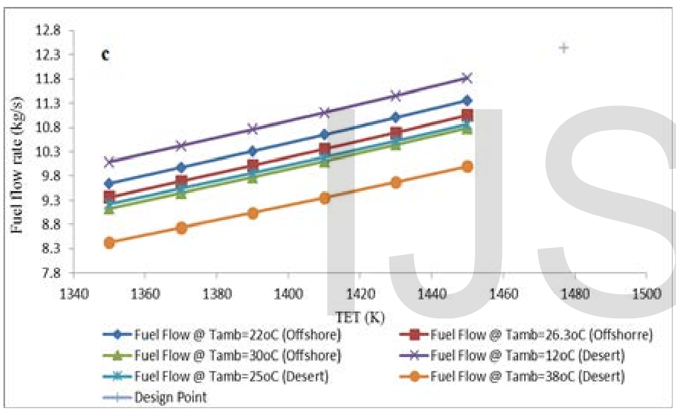


Fig. 2c. . Effect of TET and ambient temperature on Fuel flow rate

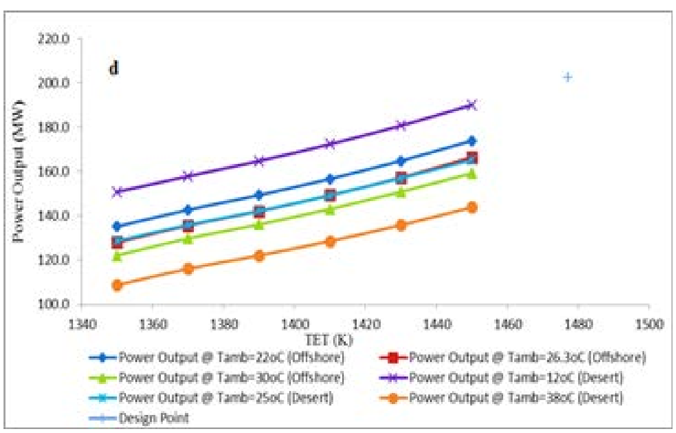


Fig. 2d. . Effect of TET and ambient temperature on Power output

Fig. 2f shows the result of the initial pressure drop on the coalescer filter at different power settings and temperatures. It also depicts the validation of the simulated initial filter pressure drop and inlet AFR with the experimental, which exist at 99.91 % validity. Also, from (Fig. 2f) a 1.8 % reduction in power setting from ISO at temperature conditions of 15°C ≤ $T_{amb}$ ≤ 38°C resulted in a decline in filter IPD and AFR. The values stood at 3.1 % ≤ IPD ≤ 5.7 % and 2.3 % respectively. Conversely, the results indicated the IPD and AFR are affected by the inlet air density whereas the ambient temperature of air remains the dominant factor.

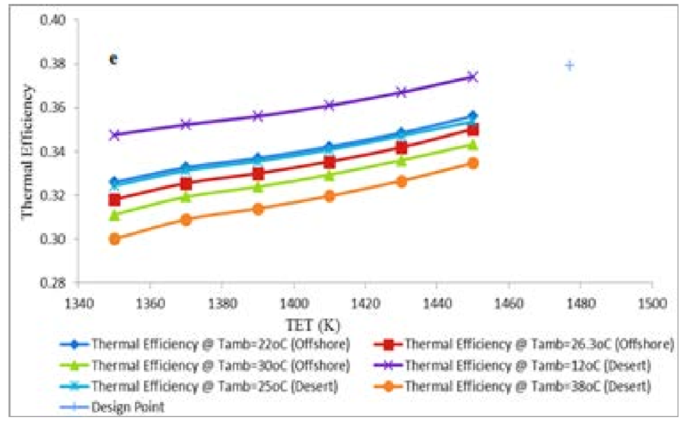


Fig. 2e. . Effect of TET and ambient temperature on Thermal efficiency

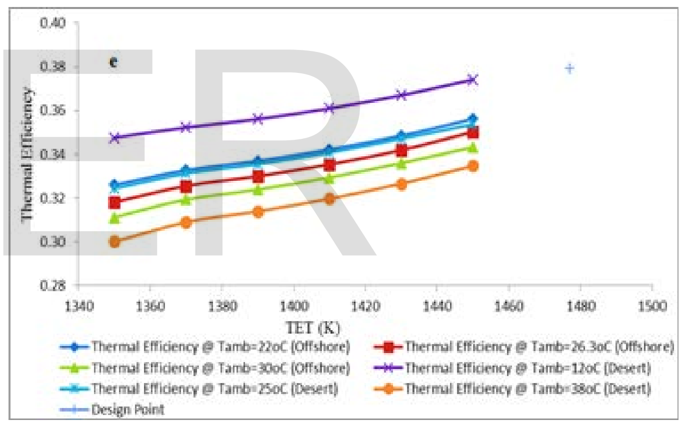


Fig. 2f. . Effect of TET and ambient temperature on Initial pressure drop

The reduction therefore in AFR might be compensated by modifying the aerodynamic design of the GT filter housing and a combination of high-efficiency filters with the coalescer. Consequently, the adequate design of filter housing and proper positioning of the filters during design and installation will improve the AFR and thus GT performance. Also, since, for all Tamb > 15°C led to a reduction in GT shaft power, the aero-derivative GT can best operate with an inlet air pre-cooling system in these locations. . The pre-cooling system will bring the condition of air close to ISO before compression thereby reducing compressor work and an increase in shaft power [1], [10], [15].

## 4 Conclusion

The performance modeling of an aero-derivative GT system with a coalescer filter at varying operating conditions is pre-

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sented. The GT was modeled to function in an offshore and desert environments in Nigeria. The performance of the GT shows that 1.8 % reduction in TET, led to a 2.3 % loss in inlet AFR. Moreover, for temperatures of $15^oC \le T_{amb} \le 38^oC$ in both environments, the inlet AFR, OPR and IPD declined significantly. The values ranged between 2.7 % $\le m_{air} \le$ 10.8 % , 2.2 $\le$ OPR $\le$ 6 % and 3.1 % $\le$ IPD $\le$ 5.7 %, respectively. At low ambient conditions ($12^oC$) from ISO for same 1.8 % reduction in TET, the GT witnessed a decrease in air flow rate by 2.4 % and a 1.6 % rise in OPR. The estimated initial pressure drop falls close to the experimental values having a validity of about 90 %. Indicating, the suitability of the modeled aero-derivative industrial GT system for power generation in the locations concerned.

Nevertheless, to adapt the ADIGT for power generation in the studied locations, the following were suggested: the use of combined filters of high efficiency with the coalescer filter will increase filter lifespan, boost GT performance and increase compressor longevity. Modification of the aerodynamic design of the GT filter housing may be necessary to enhance air flow and also sustain engine power for reasonable operating time. Since the ADGT was susceptible to large pressure drops, the use of GORE filters will be crucial. Adequate positioning of filters in the filter housing will equally enhance inlet AFR and reduce large pressure drops. Pre-cooling the inlet air close to ISO before compression will improve the volumetric flow rate, reduce compressor work and enhance output.

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