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Performance modeling of industrial gas turbines with inlet air filtration system

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ABSTRACT

The effect of inlet air filtration on the performance of two industrial gas turbines (GT) is presented. Two GTs were modeled similar to GE LM2500+ and Alstom GT13 E2-2012, using TURBOMATCH and chosen to operate at environmental conditions of Usan offshore oilfield and Maiduguri dessert in Nigeria. The inlet pressure recovered (P_{recov}) from the selected filters used in Usan offshore, and Maiduguri ranged between $98.36 \le P_{recov} \le 99.51\,\%$ and $98.67 \le P_{recov} \le 99.56\,\%$ respectively. At reduced inlet P_{recov} by 98.36% (1.66 kPa) and, at a temperature above 15 °C (ISA), a reduction of 16.9%, and 7.3% of power output and efficiency was obtained using GT13 E2-2012, while a decrease of 14.8% and 4.7% exist for power output and efficiency with GE LM2500+. In addition, a reduction in mass flow rate of air and fuel under the same condition was between $4.3 \le m_{air} \le 10.6\%$ and $10.4 \le m_{fuel} \le 11.5\%$ for GT13 E2-2012 and GE LM2500+, correspondingly. However, the GE LM2500+ was more predisposed to intake pressure drops since it functioned at a higher overall pressure ratio. The results obtained were found worthwhile and could be the basis for filter selection and efficient compressor housing design in the locations concerned.

1. Introduction

Gas turbines consume a large volume of ambient air during in-service condition. For this reason the quality of ambient air incoming the system is essential to the overall performance and the longevity of the gas turbine. A filtration mechanism is usually employed to regulate the quality of ambient air by removing foreign bodies or contaminants present in the moving air. Additionally, the choice of filtration system can be tasking, due to several factors considered during the selection of filtration system [1]. Nonetheless, the latter is predicated on the operational environment of the turbine, which includes contaminants from ambient air, surroundings emission, and seasonal changes. Inadequate filtration system leads to inlet pressure drop, reduction in power output and the overall engine efficiency [2–4]. The parameters for defining filters are average arrestance, dust holding capacity, the average efficiency and initial pressure drops [5]. The main high-efficiency filters comprise EPA, HEPA, and ULPA. HEPA and EPA filters have minimum efficiencies defined at 99.95 and 85%, respectively for particle size larger than or equivalent to 0.3 μm. In addition, the ULPA filters are described equally to have 99.995% minimum efficiency for particle sizes of 0.12 μm [6].

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Nomenclature

COP	compressor outlet pressure (kPa)
c_p	specific heat capacity (kJ kg^{-1} K^{-1})
COT	compressor outlet temperature (°C)
DP	design point
EGT	exhaust gas temperature (°C)
EPA	efficient particulate air filter
FAR	fuel air ratio
FCV	fuel calorific value (kJ kg^{-1})
FF	fuel flow rate (kg s^{-1})
g	acceleration due to gravity (m s^{-2})
GE	general electric
GT	gas turbine
h	enthalpy (kJ kg^{-1})
HEPA	higher efficiency particulate air filter
ISA	international standard atmosphere
m	mass flow rate (kg s^{-1})
OPR	overall pressure ratio
P	pressure (kPa)
PR_C	compressor pressure ratio
Q	heat input (kW)
T	temperature (°C)
TIT	turbine inlet temperature (°C)
ULPA	ultra low particulate air filter
W	work input (kW)
z	elevation (m)

Greek symbols

ι_{oval}	overall efficiency (%)
γ	specific heat ratio
Δ	change
$\cap$	isentropic efficiency (%)

Subscripts

air	pertaining to air
amb	ambient
c	compressor
comb	combustion chamber
cool	cooling
cout	compressor outlet
fout	filter outlet
gas	pertaining to burnt gases
in	input
mix	mixture of air and burnt gasses
recov	recovery
t	turbine

Studies have shown that for large power generating plants, a slight enhancement in the system efficiency can result in high system performance. For example, improvement in the global efficiency of the installed 2500 GW capacity gas turbines by 1%, would lead to a reduction of approximately 300 million tons of CO_2 per year. Moreover, about 100 t of fossil fuel can be saved from entering the atmosphere [5].

In Nigeria, the total installed capacity of gas turbine plants was estimated at 5976 MW in 2012 with actual average generating capacity of 3200 MW [7]. The plants are characterized by low performances, caused by varied technical problems leading to significant losses. However, issues related to system design, economic costing, efficient energy utilization and energy conversion process were viewed important in this period of limited conventional energy resources [8]. Adequate understanding of environmental conditions in which gas turbine operates will help reduce maintenance cost and unnecessary downtime, as appropriate design and operating parameters are upheld.

In this study, the initial and final pressure drops associated with the filtration system applied to two industrial gas turbines (heavy duty and small aero-derivative gas turbines) are investigated. The two gas turbines (GTs) are modeled similar to GE LM2500+ and Alstom GT13 E2-2012, chosen to operate at environmental conditions of Usan offshore oilfield and Maiduguri dessert in Nigeria. The specific aim is to develop working data that will assist in the selection of filtration system and further enhance the design of compressor housing in these locations.

2. Methodology and analysis

Two industrial gas turbines selected, comprise a heavy duty gas turbine (Alstom GT13 E2-2012) and an aero-derivative gas turbine (GE LM2500+). Performance data at ISA condition acquired from public domain is presented in Table 1. The data were used to create engine models similar to the selected engines using the in-house software developed by Cranfield University (TURBOMATCH) [9] as depicted in Figs. 1 and 2. Comparison between the simulated performances at ISA condition with that obtained from the literature for the two models is shown in Table 2. The Alstom GT13 E2-2012 is a single shaft system that produces 202.7 MW with efficiency of 38%. While GE LM2500+ is a single-shaft aero-derivative GT incorporated with free power turbine that produces 30.2 MW with efficiency of 39%. In addition, this study adopted the multistage filtration layout for both locations. A total of 520 filters, for the heavy duty industrial gas turbine are assumed with an air mass flow rate of 624 kg/s. In the same vein, 70 filters were considered for the aero-derivative gas turbine, having air mass flow rate of 85 kg/s. The initial and final pressure drops in the filtration system layout as specified by the manufacturers was adopted (Table 1) [10].

Table 1
Summary of input data for off-design performance [10].

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

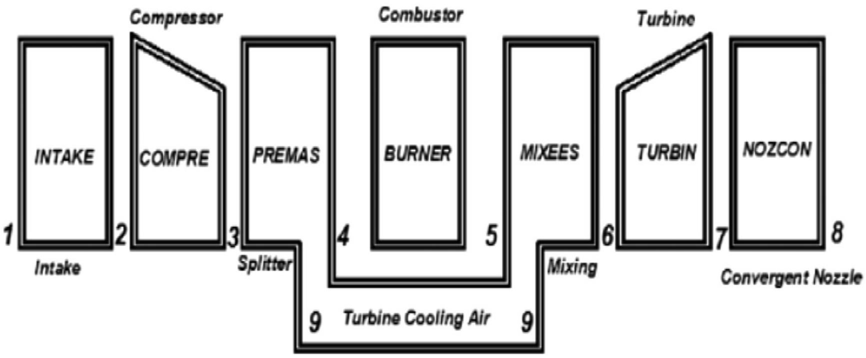

Fig. 1. Turbomatch engine model schematics for ALSTOM GT13 E2-2012.

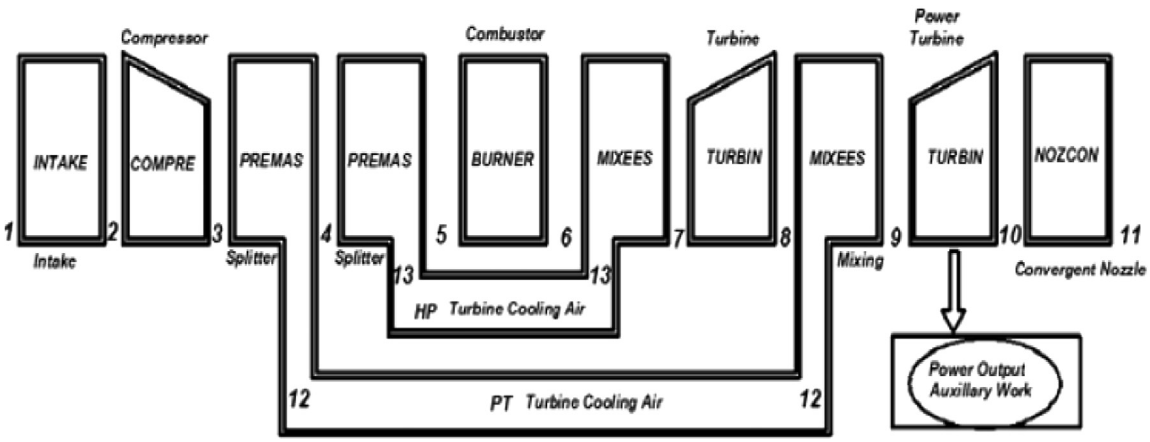

Fig. 2. Turbomatch engine model schematics for GE LM2500+.

2.1. Governing equations

Eqs. (1)–(27) depicts the major models for GT13 E2-2012 turbine (Fig. 1). The models for GE LM 2500+ (Fig. 2) are not presented since the underlying principles for derivation are the same with GT13 E2-2012. The general expression for the pressure recovered from the filter intake duct to the compressor inlet of a GT system is obtained using Eq. (1) [3]:

$$P_{recov} = \frac{\Delta P}{P_{amb}} = \frac{\left(P_{amb} - P_{f\ out}\right)}{P_{amb}} = \frac{(P_1 - P_2)}{P_1} \tag{1}$$

where $P_{amb} = P_1$ is the ambient pressure and $P_{f\ out} = P_2$ is the filter outlet pressure. The inlet pressure recovered (P_{recov}) is

Table 2
Comparison of real performance data with simulate results for GT 13E-2012 and GE LM2500+ [9].

Parameter	GT 13E-2012			GE LM2500+		
	Real	SR	*D* (%)	Real	SR	*D* (%)
Power output (MW)	202.7	202.7	0	30.2	30.2	0
Thermal efficiency (%)	38.0	37.81	0.5	39.0	38.65	0.9
Compressor pressure ratio	18.2	18.2	0	23.1	23.5	1.7
Exhaust gas flow (kg/s)	624	624.43	0.07	85	82.81	2.5
Exhaust gas temperature (K)	774	765.64	1.08	791.48	782.02	1.2

SR=Simulated results, D=difference between real and simulated results.

occasionally specified by some manufactures of different filter types [11]. Subsequently, the filter outlet pressure (P_2) can be calculated directly for known value of P_{recov} as shown in Eq. (2).

$$P_2 = P_{f\ out} = P_{recov} \times P_{amb} \tag{2}$$

The compressor outlet pressure is defined [3, 8] by:

$$P_2 = PR_C\, P_1 \tag{3}$$

Where PR_C is the compressor pressure ratio.

Applying Eq. (3) to the GT13 E2-2012 model (Fig. 1) and using the engine notations or the station points, the exit pressure can be defined by the following expression:

$$P_3 = P_{c\ out} = P_{f\ out} \times PR_C \tag{4}$$

Similarly, the ideal compressor outlet temperature can be determined from the isentropic correlation as:

$$T_{2S} = T_1 PR_C^{\left(\frac{\gamma_{air}-1}{\gamma_{air}}\right)} \tag{5}$$

The isentropic efficiency of the compressor ($\cap_C$) is defined as the ratio of isentropic work of compressor to the actual work of the compressor. The thermodynamic expression for $\cap_C$ is given by [3].

$$\cap_c = \left(\frac{\text{Isentropic work of compressor}}{\text{Actual work of compressor}}\right) = \left(\frac{h_1 - h_{2s}}{h_1 - h_2}\right) = \left(\frac{T_1 - T_{2s}}{T_1 - T_2}\right) \tag{6}$$

The actual discharge temperature T_2 can be obtained by substituting Eq. (5) into Eq. (6) and rearranging terms.

$$T_2 = T_1\left[\left(\frac{(r)^{\frac{\gamma_{air}-1}{\gamma_{air}}} - 1}{\cap_C}\right) + 1\right] \tag{7}$$

Applying Eq. (7) to the engine model (Fig. 1) and using engine notions, the compressor exit temperature is determined as below:

$$T_3 = T_{c\ out} = T_{f\ out}\left[\left(\frac{(PR_c)^{\frac{\gamma_{air}-1}{\gamma_{air}}} - 1}{\cap_C}\right) + 1\right] \tag{8}$$

Furthermore, the air from the compressor is splitted into primary and secondary air. The primary air is used for combustion while the secondary air (bled air) is used to lower the burnt gas temperature before entry into the expansion turbine. The latter is to cool the turbine blades in order to reduce the creep life. In this study, 4% of the compressed air was used for cooling while 96% was utilized for combustion Eqs. (9) and (10) [12,13].

$$m_{4air} = 0.96 \times m_{3air} \tag{9}$$

$$m_{9air} = m_{air\ cool} = 0.04 \times m_{3air} \tag{10}$$

Where, m_{3air} is the flow rate of air from the compressor, m_{4air} is the mass flow rate of air into the combustion chamber and m_{9air} is the mass flow rate of turbine blade cooling air. In addition, the combustor outlet temperature (COT) and the combustor outlet pressure (COP) are presented in Eqs. (11) and (12), respectively with 7% pressure loss assumed in the combustion chamber [3, 14, 15]

$$COT = T_5 = \left[FAR \times FCV \times \frac{\iota_{comb}}{C_{Pgas}} \right] + T_{c\ out} \tag{11}$$

$$COP = P_5 = 0.93 \times P_{c\ out} \tag{12}$$

Where, *FAR* is the fuel air ratio defined as the ratio of mass of fuel to the mass of air. The parameters in the square bracket, Eq. (11) are responsible for the flame temperature (temperature of the burnt gases), while $T_{c\ out}$ is a fraction of the compressed air exiting at the maximum compressor temperature but lower than the temperature of the burnt gases. The two temperatures accounts for the COT.

Moreover, in determining the energy balances within the components of the GT, the relevant steady state energy equation is considered [3].

$$Q - W = \Delta h + \frac{1}{2}\Delta v^2 + g\Delta z \tag{13}$$

Where Q and W are the heat and work done per unit mass flow of the working fluid. Thus, in applying Eq. (13) to the GT system in Fig. 1, the following assumptions were made: (i) Compression and expansion process are isentropic (reversible and adiabatic) (ii) the change in the kinetic energy $\frac{1}{2}\Delta v^2$ and potential energy $g\Delta z$ are negligible (iii) the composition of the working fluid throughout the entire control volume is the same an assumed a perfect gas and (iv) fuel flow rate is constant [3]. Consequently, from these assumptions, the compressor work (W_C), heat input in the combustion chamber (Q_{in}), the rate of fuel flow into the combustor (FF) and the flow rate of burnt gases (m_{gas}) are determined by the expressions in Eqs. (14)–(17), respectively.

$$W_C = m_{3air} C_{pair} (T_{c\ out} - T_{f\ out}) \tag{14}$$

$$Q_{in} = m_{gas}\ C_{pgas} (T_5 - T_4) \tag{15}$$

$$FF = \frac{Q_{in}}{FCV} \tag{16}$$

$$m_{gas} = \sum (m_{3air}, FF, m_{4air}) \tag{17}$$

Where *FCV* is the fuel calorific value.

The mixing of the burnt gases and the cool air before entry the expansion turbine occur at a temperature T_{mix} and pressure P_{mix} (Fig. 1). Hence the outlet conditions are:

$$T_6 = T_{mix} = T_5 - T_9 \tag{18}$$

$$P_6 = P_{mix} = P_5 \tag{19}$$

From Eq. (10) 4% of the compressed air is utilized for cooling, thus T_9 can be obtained as:

$$T_9 = T_{cool} = 0.04 \times T_{c\ out} \tag{20}$$

Substituting Eqs. (11) and (20) into Eq. (18), we have

$$T_6 = T_{mix} = \left[FAR \times FCV \times \frac{\iota_{comb}}{C_{Pgas}} \right] + T_{c\ out} - 0.04T_{c\ out} \tag{21}$$

Applying the same isentropic correlation to the turbine unit, and assuming a 2% pressure loss in the exhaust nozzle the outlet conditions are expressed [3, 12] by:

$$P_7 = P_{t\ out} = \frac{P_{amb}}{0.98} \tag{22}$$

$$T_7 = EGT = T_{mix}\left[1 - \left[\cap_t \left(1 - \left(\frac{P_{t\ out}}{P_{mix}} \right)^{\frac{\gamma_{gas}-1}{\gamma_{gas}}} \right) \right] \right] \tag{23}$$

In the turbine unit, the heat input (Q) is zero and the terms $\frac{1}{2}\Delta v^2$ and $g\Delta z$ are negligible. Hence, the steady state equation, Eq. (13) reduces as:

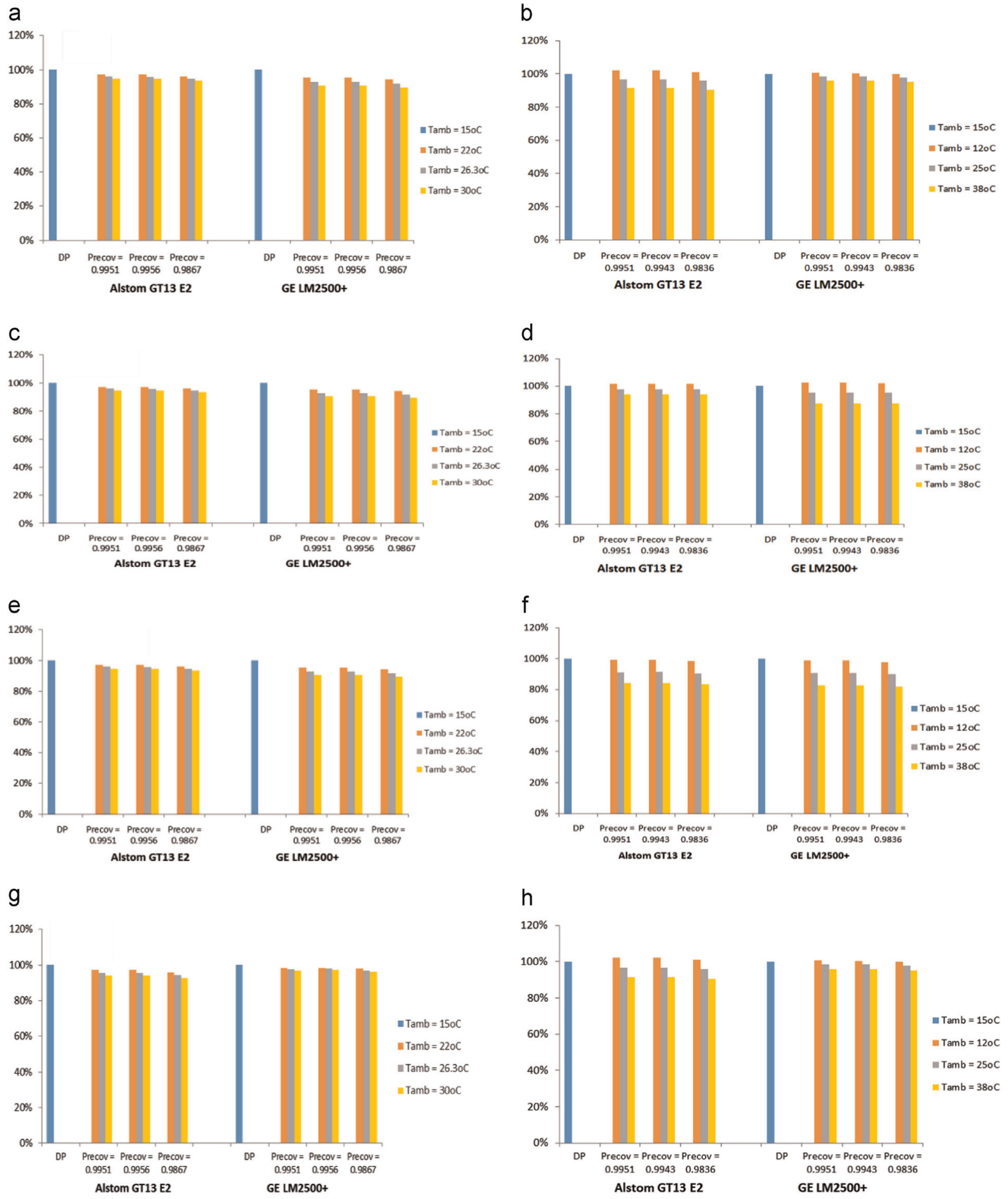

Fig.. 3. Sensitivity results of effect of pressure drop on (a,b) inlet air flow rate, (c,d) overall pressure ratio (e,f) fuel flow rate and (g,h) thermal efficiency for Usan oilfield and Maiduguri desert respectively.

$$W_t = \Delta h \tag{24}$$

The enthalpy change Δh of an ideal gas during a thermodynamic cycle involving two states is obtained from [16]:

$$\Delta h = \int_1^2 c_p(T)dT = c_p(T_2 - T_1) \quad (25)$$

Applying Eq. (25) to the GT model (Fig. 1) for a particular mass flow rate of burnt gases expanding in the turbine, with specific heat (C_{pgas}), the final turbine work output and the overall efficiency are determined.

$$W_t = m_{gas} C_{pgas} (T_{mix} - EGT) \quad (26)$$

$$\eta_{oval} = \frac{W_t}{Q_{in}} \quad (27)$$

3. Results and discussion

Table 2 presents a comparison between the simulated results with real values obtained from the literature for GT13 E2-2012 and GE LM2500+ engine models. The results show a close match with that at ISA conditions. The deviations for the efficiencies were 0.5% and 0.9% for GT13 E2-2012 and GE LM2500+ respectively. Fig. 3 depicts the sensitivity results obtained for various operating conditions. In Fig. 3(a) and (b), the inlet filter pressure drop (IFPP) decreases the air flow rate and ranged between $2.8 \le m_{air} \le 6.4\%$ and $5.9 \le m_{air} \le 10.6\%$ in Usan oilfield, for ambient temperatures (T) of $22\,^\circ C \le T \le 30\,^\circ C$ for GT13 E2-2012 and GE LM2500+, respectively. While Maiduguri desert with yearly minimum and maximum T determined at $T_{min} \le 12\,^\circ C$ and $T_{max} \le 38\,^\circ C$, decrease in IFPP for temperatures from 15 °C (ISA) to $T_{min} \le 12\,^\circ C$, results in a reduction of air flow rate ranged between $2.3 \le m_{air} \le 6.6$ % with GT13 E2-2012. Also, for T conditions between 15 °C (ISA) and 25 °C, a decrease of 1.7% and 16% exist for GE LM2500+ model respectively. At constant turbine inlet temperature (TIT), pressure drops have little effect on the overall pressure ratio (OPR), for both GTs, but at an increasing ambient temperature, more compression work is required by the GTs and thus a fall in OPR Fig. 3(c) and (d).

Fig. 3(e) and (f) presents the effect of pressure drop in fuel flow rate for ranges of T between $15\,^\circ C \le T \le 30\,^\circ C$ and $15\,^\circ C \le T \le 38\,^\circ C$ in Usan and Maiduguri respectively. The fuel flow rate varies from 4.7% to 10.4% for GT13 E2-2012 and 6.2% to 11.4% for GE LM2500+ in Usan. Maiduguri environment recorded a reduction between 0.8% and 16%. However, the GE LM 25000+ was more predisposed to intake pressure drops since it functioned at a higher overall pressure ratios. Equally, in Fig. 3(g) and (h) the pressure drops and variations in ambient temperature was found to have affected the thermal efficiency due to output power reduction. Furthermore, to improve system performance an air precooling system will be required to bring the ambient air close to ISA condition before compression in the two environments [17].

4. Conclusions

The effect of inlet air filtration with respect to the initial and final pressure recovered on the performance of Alstom GT13 E2-2012 and GE LM2500+ gas turbines was studied. Two extreme locations (Usan offshore oilfield and Maiduguri dessert) in Nigeria were considered. The study was structured to generate technical information for the selection of suitable filtration system. In addition, provide options for compressor filter housing design under the considered locations. The findings are thus summarized:

- At reduced inlet P_{recov} by 98.36% (pressure drop of 1.66 kPa) and increased ambient temperature from 15 °C (ISA) to 38 °C, lead to a decrease in air flow rate ranging between $2.8 \le m_{air} \le 6.4\%$ and $5.9 \le m_{air} \le 10.6\%$ for GT13 E2-2012 and GE LM2500+ in Usan oilfield, respectively. Whereas Maiduguri desert with yearly minimum and maximum T determined at $T_{min} \le 12\,^\circ C$ and $T_{max} \le 38\,^\circ C$, decrease in pressure drops from 15 °C (ISA) to $T_{max} \le 38\,^\circ C$, lead to a reduction in air flow rate in the bandwidth $2.3 \le m_{air} \le 6.6$ % and $1.7 \le m_{air} \le 16$ % for GT13 E2-2012 and GE LM2500+ respectively. Equally, the thermal efficiency and the OPR decreases at same conditions for the two GTs.
- The GE LM2500+ engine model was more susceptible to higher intake pressure drops in both environments since it functioned at a higher overall pressure ratios. Thus, replacing the static pre-filters regularly, will improve system performance. Especially, in the desert locations where filters are susceptible to high dust build-ups. Likewise, increase in the active area of the pre-filters will increase volumetric flow rate, improve filter performance and operational life span.
- The HEPA filters used for the two environment of Usan oilfield, and Maiduguri desert have demonstrated some high level of performance. However, comparing the two operational locations, high percentage differential increase and decrease from the design point (DP) for the different performance parameters were observed. For the installation of gas turbines in the two locations, an inlet air pre-cooling system will be necessary to bring the air condition close to ISA before compression. In addition, increase in the size of the gas turbine inlet filter housing or the use of GORE ® turbine filters can be of immense benefits.

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