Exergo-Economic Analysis of a Gas Turbine Plant with Inlet Air Fogging System – A Case Study

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Abstract

This study investigates the exergoeconomic analysis of a gas turbine power plant with compressor inlet air fogging system. The objective of the study is to determine the degree of exergy destruction and economic implications which accrues due to the fogging arrangement for the proposed case study. First, a comprehensive exergy analysis was presented based on plant component modelling. Exergoeconomic cost rates were developed using standard purchase and equipment cost as functions of plant operating variables. The developed cost matrix was solved using a written program in engineering equation solver EES. The results show thatthe cost stream of the combustion chamber reduced by 1.52 %, while that after the turbine reduced by 2.22 % for the fogged system. Additionally, the exergy streams related to compressor work for the fogged system had total reduction of 3.76 % leading to a power increase of about 5.20 MW. Moreover, a sensitivity breakdown indicates that for consecutive 10 MW rise, the exergy stream of the compressor improved by 142 $/hr. at 110 MW, 138 $/hr. at 120 MW, 135 $/hr. at 130 MW and 130 $/hr. at 140 MW. Similarly, the inlet stream to the expander (turbine) exist at 1560 $/hour, 1487 $/hour, 1429 $/hour, 1381 $/hour, 1341 $/hour, and 1307 $/hour for 270MW, 290MW, 310MW, 330MW, 350MW and 370MW of turbine output respectively. The results show that the cost rates are smaller at the exit of the turbine due to small temperature difference in the burnt gasses. The results show reduced cost of exergy destruction due to fogging justifies the retrofitting of the examined gas turbine plant.

Keywords: Exergy, exergo-economics, exergetic destruction, gas turbine, fogging system.

1. Introduction

The general concern today about engineering systems especially energy conversion devices is waste reduction and efficiency improvement. The latter, however, is achieved by ensuring thermodynamic processes involving energy conversion are improved. Also, the dwindling conventional energy resourceshas prompted scientists and engineers to look for alternative energy sources as well as improve and optimize existing systems for low energy consumption (Levenda et al., 2021; Frangopoulos, 2018). Gas turbines are identified as heavy energy conversion machines. Currently, in most parts of the world, they are applied for peak loads generation because of their quick start ability. In Nigeria, for instance, gas turbines are applied in base load applications and dominates the power generating sector.

Thermodynamically, the power output of gas turbines is known to decrease with increased ambient temperatures (Homji-Meyer et al., 2002). This factor is more obvious in Nigeria where the ambient temperature condition is higher than the recommended ISO standard (15°C). Thus, installed turbines in Nigeria generate below design capacity. Incidentally, at such periods of elevated ambient temperature, electrical energy demand usually increases owing to the power needed for cooling and ventilation systems. However, one of the identified methods of increasing power output in GT systems is by cooling the inlet air prior to the compressor.

The two common methods for compressor inlet air cooling are the evaporative and refrigeration cooling systems. One of the methods of the evaporative cooling system is the high pressure fogging of the inlet air. This method has been known for relatively low cost of installation and operations as well as effective turbine output power augmentation (Ehyaei, et al., 2011; Jassim, et al., 2010). Although, gas turbine inlet air cooling has been an established engineering process and the technologies involved common, especially in the developed economies of the world where the use of the gas turbine has been since in the late 1930s, manufacturers and power plant operators in recent times are concerned in designing improved turbine systems with relatively low impact to the environment in terms of environmental pollution and greenhouse emissions. This is achieved by matching the operating conditions in real situation and also understanding off-design conditions.
Current researchers have presented numerous works concerning performance assessments of gas turbines employing the exergy concept (second law of thermodynamics) (Kotas, 1995; Ebadi et al., 2005). The first law of thermodynamics treats all energy forms as equivalent and takes no account of internal losses or irreversibilities in the thermal processes. The exergy method (a consequence of the second law of thermodynamics) identifies, locates and gives magnitude of thermodynamic inefficiencies or losses in thermal systems (Abam, et al., 2012). This offers the opportunity for improvements, either by redesigning or by retrofitting the existing systems.

Related studies in literature indicate that an evaporative cooler can increase the relative humidity of air by 90 %, boost the turbine output power by 5 – 10 % and increase efficiency by 1.5 - 2.5 % (Jonsson and Yan, 2005). The same studies show that cooling the inlet air by chillers can boost turbine output power and efficiency by 15 – 20 % and 1 – 2 % respectively. Jaber *et al.,* 2007 considered the influence of intake air cooling on the combustion turbine performance. They observed the performance characteristics of a turbine with respect to operational parameters such as ambient temperature, relative humidity, turbine inlet temperature, and pressure ratio. Their results showed that the evaporative cooling system is capable of boosting the power and enhancing the efficiency of the studied turbine unit in a way much cheaper than the cooling coil system.

Marzouk and Hanafi, (2013) also compared the chiller cooling and evaporative cooling for a 264 MW. They concluded that the evaporative cooler is more economical as it required a low maintenance cost, low electricity consumption, and low capital cost. Similarly, Oyedepo and Kilanko, (2014) studied the performance enhancement of a turbine with an evaporative cooler. Their result shows that for each 5ºC decrease of inlet air temperature, the net output power and thermal efficiency ranged from 5 to 10 % and 2 - 5 % respectively. Indeed, there are different methods for gas turbine inlet air cooling, each with its limitations and degree of success bothering on economy, power requirement and maintenance cost. In this research work, the exergoeconomics of air fogging system is applied to an in-service power plant (Afam power plant, Nigeria) to show that reduced cost of exergy destruction due to fogging justifies the retrofitting of the examined gas turbine plant.

2. Objectives

- To determine the degree of exergy destruction in the fogging system
- To determine the economic implications which accrues due to the fogging arrangement for the proposed case study.

3. Methods

3.1 Description of gas turbine with fogged cooling system

Fig.1 shows the gas turbine with the fogging system. Raw water is drawn into the demineralized water plant at state (5) where certain mineral deposits are removed, maintaining the pH of the water at 7.5

maximum. The demineralized water enters the storage facility at (6) and exit at (7), from where the water is pumped to the position nozzles at state (8). The nozzles atomize the demineralized water into tiny water droplets (fog). Each droplet, by the nozzles design, is recommended to be less than 50 microns in size. The water is made to develop such high pressure as 70 bars. Critical parameters of the fogging system include droplets mean size, their distribution pattern and their extent of penetration into the air duct from the points of their production.

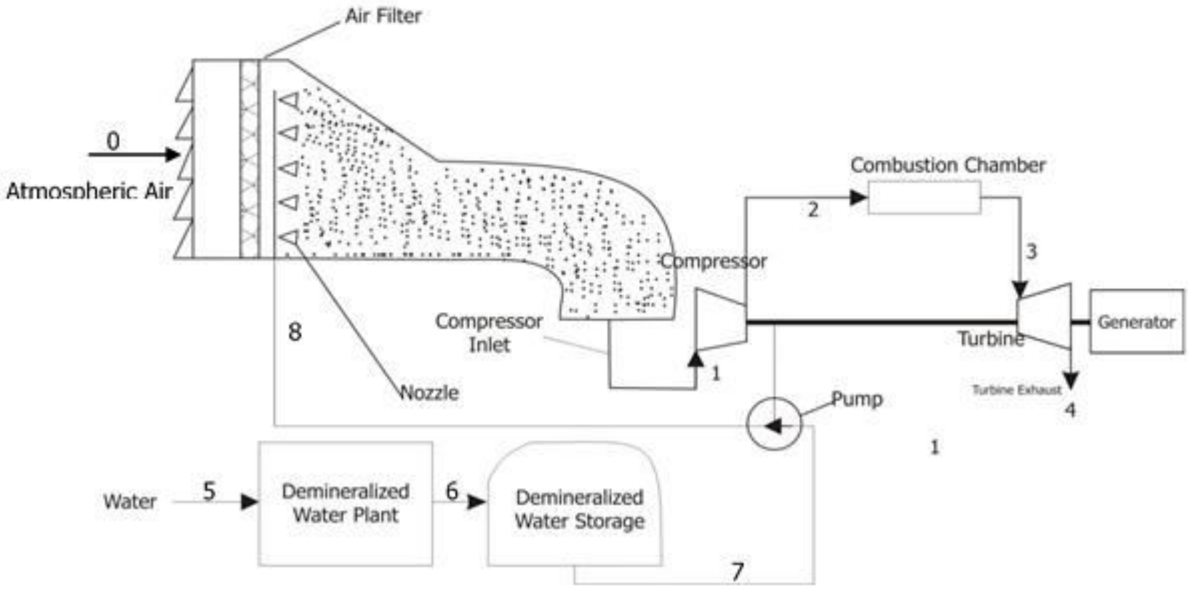

Figure. 1. Schematic diagram of the gas turbine modelled with inlet fogging system

3.2 Exergoeconomic analysis

Exergoeconomic analysis of thermal systems involves first principle balancing of cost rates of exergy streams at inlet and outlet conditions as well as the cost rate of the component under consideration. Cost rates of GT components based on exergoeconomic modelling for both the fogged and the unfogged systems are similar except for varying values of the magnitude of exergy streams at the points; where the variation is due to the fogging arrangement. Purchase cost of plant components are represented as functions of plant operating parameters. In addition, auxiliary equations are developed to aid computation of cost variables. The thermoeconomic analysis in terms of cost balances is presented.

3.2.1 Exergoeconomic balance

For any system receiving heat Q, and performing some work, W with exergy influx, E_i and exergy efflux, E_e, a general cost equation can be written as, (Lazzaretto and Tsatsaronis, 2006)

$$c_{q,k}E_{q,K} + \sum_i\left(c_i\dot{E}_i\right)_k + \dot{Z}_K = \sum_e\left(c_e\dot{E}_e\right)_k + c_{w,k}\dot{W}_k \qquad 1$$

Where:
$\sum_e\left(c_e\dot{E}_e\right)_k$ = Cost rates associated with exit exergy streams of the k^{th} component.
$c_{w,k}\dot{W}_k$ = Cost rates of power generation.
$c_{q,k}\dot{E}_{q,k}$ = cost rates associated with heat transfer.
$\sum_i\left(c_i\dot{E}_i\right)_k$= cost rates associated with entering exergy streams.
$\dot{Z}_k$ = capital investment cost rate.
c_e, c_w, c_q, c_i are the average costs per unit of exergy.
In line with the nomenclature of the schematic diagram (Fig. 1), the following cost balances are presented:

Air compressor:

$$C_1 + C_{W_{AC}} + \dot{Z}_{AC} = C_2 \quad 2$$

Combustion chamber:

$$C_2 + C_{Fuel} + \dot{Z}_{CC} = C_3 \quad 3$$

Gas turbine:

$$C_3 + \dot{Z}_{GT} = C_4 + C_{W_T} + C_{W_{AC}} \quad 4$$

The variables in equations 2, 3, and 4 are five (i.e. $c_2, c_3, c_4, c_{W_T}, c_{W_{AC}}$). This is because $c_1 = 0$ *and* c_F can be determined. With five variables and three equations the cost equations are not susceptible to valid solution values. Consequently, auxiliary equations are developed on the turbine using specific exergy costing principles as follows.

$$\frac{C_3}{E_3} = \frac{C_4}{E_4} \Rightarrow C_3E_4 - C_4E_3 = 0 \quad 5$$

$$\frac{C_{W_T}}{W_{NET}} = \frac{C_{W_{AC}}}{W_{AC}} \Rightarrow C_{W_T} * W_{AC} - C_{W_{AC}} * W_{NET} = 0 \quad 6$$

In matrix form, the cost balance is here represented below:

$$\begin{bmatrix} -1 & 0 & 0 & 0 & 1 \\ 1 & -1 & 0 & 0 & 0 \\ 0 & 1 & -1 & -1 & -1 \\ 0 & E_4 & -E_3 & 0 & 0 \\ 0 & 0 & 0 & W_{AC} & -W_{NET} \end{bmatrix} \begin{bmatrix} C_2 \\ C_3 \\ C_4 \\ C_{W_T} \\ C_{W_{AC}} \end{bmatrix} = \begin{bmatrix} -C_1 - Z_{AC} \\ -Z_{CC} - C_{fuel} \\ -Z_{GT} \\ 0 \\ 0 \end{bmatrix} \quad 7$$

3.2.2 Cost rates of GT plant components

The cost rates of GT plant components are modelled with respect to operating parameters of the system in monetary units. All costs due to owning and operating a GT plant depends on its expected life, required capital and on the type of financing (Gorji-Bandpy *et al*, 2010). The cost rates of each plant component are modelled as obtained in (Balli, *et al.,* 2008; Sayyaadi and Sabzaligol, 2009)

$$Z_k(\$/h) = \frac{\Phi_k C_k}{N} \quad 8$$

Where Φ_k is the maintenance factor for each plant component whose expected life is *'N'* years. N is the annual number of hours of plant operation; C_k is the annual levelised cost. In line with values for capital recovery factor (CRF) and present work factor (PWF), C_k is expressed as (Gorji-Bandpy et al., 2010).

$$C_k(\$/year) = [PEC - (SV)PWF(i,n)]CRF(i,n) \quad 9$$

Substituting for values,

$$C_k(\$/year) = \left(PEC - (SV)(1+n)^{-n}\left[\frac{i(1+i)^n}{[(1+i)^n - 1]}\right]\right) \quad 10$$

Where CRF is determined from the relation (Balli et al., 2008)

$$CRF = \left[\frac{i(1+i)^n}{[(1+i)^n - 1]}\right] \quad 11$$

$$PWF(i,n) = (1+i)^{-n} \quad 12$$

From equation (10), the purchase and equipment cost PEC for each GT component is obtained as in (Bejan and Tsatsaronis, 1995; Frangopoulos, 1991):

Air compressor

$$PEC = \left(\frac{71.1 * m_a}{0.9 - \eta_{AC}}\right)\left(\frac{P_2}{P_1}\right) ln\left[\frac{P_2}{P_1}\right] \quad 13$$

Combustion chamber

$$PEC \;\; = \left[\frac{46.08\,(m_a+m_f)}{\left(0.995-\frac{p_3}{p_2}\right)}\right] * (1+exp[0.018T_3-26.4]) \qquad 14$$

Gas turbine

$$PEC \;\; = \left(\frac{479.34m_g}{0.92-\eta_T}\right)\ln\left(\frac{P_3}{P_4}\right)(1+exp[0.036T_4-54.1]) \qquad 15$$

3.3 Energy analysis of the system

Thermodynamic models for computation of state properties, as well as exergy models for the components are presented in line with the schematic diagram of Fig. 1.

The compressor work is obtained as a function of the ambient temperature, pressure ratio, and the index of compression as follows:

Compressor:

$$W_{AC} = C_{pair}\left\{T1\left[1+\frac{1}{\eta_{comp.}}\left(\left[\frac{P_2}{P_1}\right]^{\left(\frac{\alpha-1}{\alpha}\right)}-1\right)\right]-T_1\right\} \qquad 16$$

With air fogging arrangement, compressor work requirements can be obtained as,

$$\text{Or } W_{AC} = \dot{m}_a\left\{\frac{C_{pair}.T_1}{\eta_{comp.}}\left(\left[\frac{p_2}{p_1}\right]^{\left(\frac{\propto-1}{\propto}\right)}-1\right)+w\left(i_{g2}-i_{g2}\right)\right\} \qquad 17$$

Where i_{g2} and i_{g1} are the enthalpies of saturated water vapour at compressor exit and inlet respectively.

The combustion chambers

The energy balance for the combustion chamber for the base case is expressed in (Rajput, 2009) as:

$$\dot{m}_f C_v \eta_{comb.} + \dot{m}_a C_{pair} T_{2a} = \left(m_a+m_f\right)C_{pg}.T_3 \qquad 18$$

Rearranging equation (18) and substituting $Q_{in} = \dot{m}_f C_v.\eta_{comb.}$ the following is obtained,

$$Q_{in} = \left(m_a+m_f\right)C_{p.g}T_3 - \dot{m}_a C_{pair}\left\{T_1\left[1+\frac{1}{\eta_{comp}}\left(\left[\frac{p_2}{p_1}\right]^{\left(\frac{\propto-1}{\propto}\right)}-1\right)\right]\right\} \qquad 19$$

Hence, the energy balance for the combustion chamber for the fogged system is expressed as,

$$\dot{m}_f C_v.\eta_{comb} + \dot{m}_a C_{pair}.T_{2a} + \dot{m}_v hv_2 = \left(\dot{m}_a+\dot{m}_f\right)C_{pg}.T_3 + \dot{m}_v hv_3 \qquad 20$$

$$\text{Or } Q_{in} = \left(\dot{m}_a+\dot{m}_f\right)C_{pg}T_3 - \dot{m}_a C_{pair}.T_{2a} + \dot{m}_v(hv_3-hv_2) \qquad 21$$

The enthalpies of water vapour h_{v_2} and h_{v_3} expressed in equation (21) are estimated in (Dossat, 1997) as:

$$h_{v_{1_i}} = 2501.3 + 1.8723T_i \qquad 22$$

The term f is the ratio of air mass and fuel approximated according to the relationship in (Alhazmy *et al.*, 2006):

$$\mathrm{f} = \frac{\dot{m}_f}{m_a} = \frac{C_{pg}(T_3-298)-C_{pair}(T_2-298)+w(hv_3-hv_2)}{C_v.\eta_{comb.}-C_{pg}(T_3-298)} \qquad 23$$

The turbine

Energy balance for the turbine is presented with the relationship (Dossat, 1997):

$$C_{pg}.m_t.T_3 = W_T + C_{p.g}m_t.T_{4a} \quad 24$$

Where $m_t = \dot{m}_a + \dot{m}_v + m_f = m_a(1 + w + f)$ 25

Employing the isentropic expansion in the turbine, the exit temperature is related to the isentropic efficiency with the expression,

$$T_{4a} = T_3\left(1 - \eta_T\left[1 - \frac{1}{(r_p)^{\frac{\alpha-1}{\alpha}}}\right]\right) \quad 26$$

The turbine work can further be presented according to (Alhazmy et al. 2006) as,

$$W_T = \dot{m}_a(1 + w + f)C_{pg}.\eta_T.T_3 1 - \frac{1}{(r_p)^{\frac{\alpha-1}{\alpha}}} \quad 27$$

3.4 Exergy analysis of the system

The general exergy models for a control volume comprising exergy influx ex_{in}, efflux ex_{out}, heat input Q_{in} and work output W_{out} can be expressed under steady state conditions as (Bejan and Tsatsaronis, 1995):

$$\sum ex_{in} + ex_Q = \sum ex_{out} + ex_W + E_D \quad 28$$

Where the specific exergy is expressed at a temperature $\boldsymbol{T}$ and pressure $\boldsymbol{P}$ all referenced at the ambient temperature $\boldsymbol{T_0}$ and pressure $\boldsymbol{P_0}$. From (28), general component exergy balances are presented as follows:

Air compressor

$$Ex_1 + Ex_w = Ex_2 + E_D \quad 29$$

The exergy at state point 1 and 2 are written with respects to the combustion turbine base case configuration and that of the fogged system.

$$Ex_1 = (h_1 - h_0) - T_0(S_1 - S_0) \quad 30$$

Expressing the terms to reflect temperature and pressure components

$$Ex_1 = C_{pa}(T_1 - T_0) - T_0\left(C_{pair}ln\left(\frac{T_1}{T_0}\right) - Rln\left(\frac{P_1}{P_0}\right)\right) \quad 31$$

However, the exergy at point 1 for the base case is zero since the temperature at point 1 is ambient exergy at point 2 is expressed as,

$$Ex_2 = C_{pa}(T_2 - T_1) - T_0\left(C_{pair}ln\left[\frac{T_2}{T_0}\right] - Rln\left[\frac{P_2}{P_0}\right]\right) \quad 32$$

The exergy at point 1 for the fogged case is obtained per kg of air flow as in (Jassim *et al.*, 2009),

$$Ex_{1if} = \left(C_{pa} + wC_{pv}\right)T_0\left[\frac{T_1}{T_0} - 1 - In\left(\frac{T_1}{T_0}\right)\right] + (1 + 1.607w)R_aT_0In\left(\frac{P_1}{P_0}\right) \quad 33$$

Incorporating the mass flow, it becomes obvious that the mass flow at point 1 includes air and water vapour. Thus, equation (33) is changed to,

$$Ex_{1if} = (\dot{m}_a + \dot{m}_v)\left\{\left(C_{pa} + wC_{pv}\right)T_0\left[{}^{T_1}/_{T_0} - 1 - In\left({}^{T_1}/_{T_0}\right)\right] + (1 + 1.607w)R_aT_0In\left({}^{P_1}/_{p_0}\right)\right\} \quad 34$$

Which simplifies as:

$$Ex_{1if} = \dot{m}_a(1 + w)\left\{\left(C_{pa} + wC_{pv}\right)T_0\left[\frac{T_1}{T_0} - 1 - In\left(\frac{T_1}{T_0}\right)\right] + (1 + 1.607w)R_aT_0In\left(\frac{P_1}{P_0}\right)\right\} \quad 35$$

Similarly, the exergy at point 2 is expressed as,

$$Ex_{2if} = \dot{m}_a(1+w)\left\{(C_{pa}+wC_{pv})T_0\left[\frac{T_2}{T_0}-1-In\left[\frac{T_2}{T_0}\right]\right]+(1+1.607w)R_aT_0In\left[\frac{P_2}{P_0}\right]\right\}$$ 36

The combustion chambers

For the combustion chamber, the exergy balance is expressed as:

$$\mathrm{E_{x2}+E_{x,fuel}=E_{x3}+E_D}$$ 37

The exergy of fuel is composed of the physical and chemical parts.

$$\mathrm{E_{xf}=E_{xf,physical}+E_{xf,chemical}}$$ 38

The chemical component of fuel exergy is obtained for the base and fogged cases with the following relationships (Bejan and Tsatsaronis, 1995): for the fogged case,

$$E_{xf,chem.} = \sum_{i=1}^{n} x_i e_{ex1} + RT_0\sum x_i In(x_i)RT_0\left[(1+1.609w)In\{(1+1.607w_o)/(1+1.607w)\}+1.607wIn\left({}^{w}/_{W_o}\right)\right]$$ 39

And for the base case (Bejan and Tsatsaronis, 1995):

$$\mathrm{E_{xf,chem}=\sum_{i=1}^{n}x_iE_{chi}+RT_0\sum_{i=1}^{n}x_iIn(x_i)}$$ 40

The physical exergy component of fuel is calculated with the relationship (Bejan and Tsatsaronis, 1995),

$$\mathrm{E_{f,physical}=C_{pf.}(T_f-T_0)-T_0\left\{C_{pf}In\left[\frac{T_f}{T_0}\right]-R_{fuel}In\left(\frac{P_f}{P_0}\right)\right\}}$$ 41

The exergy of gas stream after combustion at state point 3 is obtained as

$$E_{x3}=C_{p,g}(T_3-T_0)-T_0\left\{C_{pg}ln\left(\frac{T_3}{T_0}\right)-R_gln\left(\frac{P_3}{P_0}\right)\right\}$$ 42

The exergy at point 3 for the fogged system is obtained per kilogram of fluid as;

$$E_{x3}=(C_{pa}+wC_{pv})T_0\left[\frac{T_3}{T_0}-1-ln\left(\frac{T_3}{T_0}\right)\right]+(1+1.607w)R_aT_0ln\left(\frac{P_3}{P_0}\right)$$ 43

And for a given mass equation (43) takes the form

$$E_{x3}=m_a(1+w+f)\left\{(C_{pa}+wC_{pv})T_0\left[\frac{T_3}{T_0}-1-In\left(\frac{T_3}{T_0}\right)\right]+(1+1.607w)R_aT_0In\left(\frac{P_3}{P_0}\right)\right\}$$ 44

The gas turbine

$$\mathrm{E_{x3}=E_{x4}+E_W+E_D}$$ 45

The exergy at point 4 is computed with the relationship below using the temperature and pressure values of this point as well as its mass flow rate as,

$$E_{x4}=C_{pg}(T_4-T_0)-T_0\left\{C_{pg}ln\left(\frac{T_4}{T_0}\right)-Rln\left(\frac{P_4}{P_0}\right)\right\}$$ 46

The expression for the exergy of the fogged system is presented per kg of gas as,

$$Ex_4=(C_{pg}+C_{pv}w)T_0\left(\frac{T_4}{T_0}-1-ln\frac{T_4}{T_0}\right)+(1+1.607w)R_aT_oln\left(\frac{P_4}{P_0}\right)$$ 47

For the mass stream equation (47) is written as,

$$E_{x4}=\dot{m}_a(1+w+f)(C_{pg}+C_{pv}w)T_0\left(\frac{T_4}{T_0}-1-ln\frac{T_4}{T_0}\right)+(1+1.607w)R_aT_oln\left(\frac{P_4}{P_0}\right)$$ 48

4 Results and discussion

4.1 Exergoeconomic analysis of GT with and without fogging

An exergoeconomic analysis of the GT plant is presented below for the two cases. The exergy streams for the products and fuel as well as the cost of exergy destruction is computed and compared for both cases in Tables 1 and 2.

Table 1. Summary of exergy values at different state points

State point	Exergy values (MW)	
	Fogged system	Unfogged system
1	0	0
2	130.0	132.0
3	424.0	423.0
4	109.5	110.0
Fuel	350.7	347.6
W_C	142.8	145.0
W_T	292.0	289.0

Table 2 Summary of exergy in product, fuel and destruction for the fogged/un-fogged system

Component	Fogged plant [MW]			Unfogged plant[MW]		
	E_P	E_F	E_D	E_P	E_F	E_D
air compressor	130	142.8	10.83	132	145.0	13.30
combustion chamber	424	774.7	56.70	423	770.7	59.00
gas turbine	292	314.5	23.00	289	313.0	24.00

The cost of exergy streams for the two cases is shown in Fig. 2. The cost of exergy streams for the state points and that for turbine and compressor work show significant difference in magnitude. The cost of exergy stream just after compression from the air compressor is 3276$/hr and 3189$/hr for the fogged and unfogged systems respectively, with about 2.73 % reduction (87$/hr) in the cost of exergy stream due to fogging. Similarly, reduction in exergetic cost stream was observed in the exergy streams encompassing the combustion chamber and the turbine. The cost stream after the combustion chamber reduced by 1.52%, while that after the turbine reduce by 2.22 % all in favour of the system with inlet fogging arrangement.The exergy streams related to compressor work is also significantly reduced at the fogged case with a total reduction of 3.76 % since the compressor work is less with the introduction of fogging. However, the cost stream associated with turbine work rather increase comparatively after fogging with about 1.34 % to cater for the cost which accrues due to the 3 MW augmentation

in the turbine output brought about by fogging. Furthermore, the total cost of exergy streamsis significantly reduced as a result of the fogging arrangement.

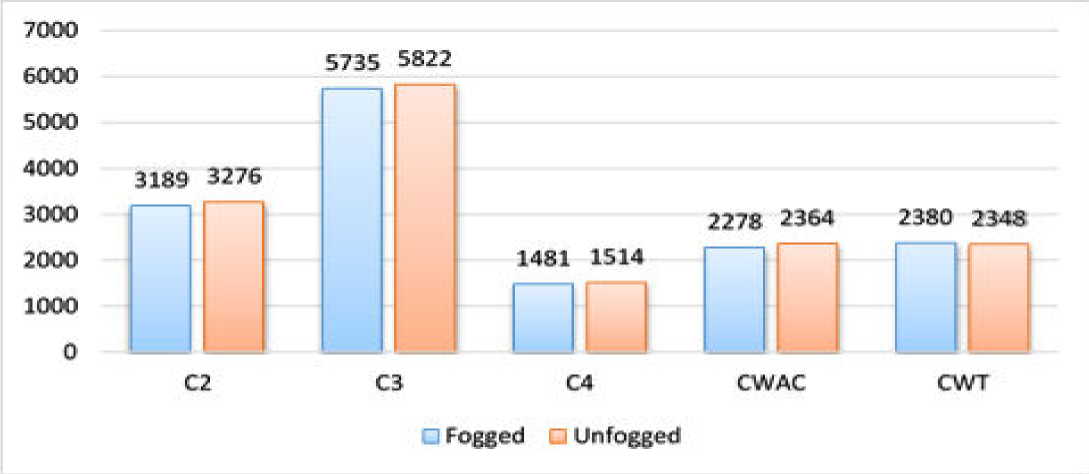

Figure 2. Cost of exergy streams for the fogged and unfogged turbine (in $/hr.)

The variation in the exergetic cost streams as observed in Fig. 2, is partly due to variation in temperature of the streams but mostly on the compressor and turbine work. Consequently, a study on the sensitivity analysis of the effect of compressor work on the cost of these streams is worthwhile. This analysis is presented in Fig. 3 where variation in the compressor work is studied on the cost streams while other parameters are kept constant.

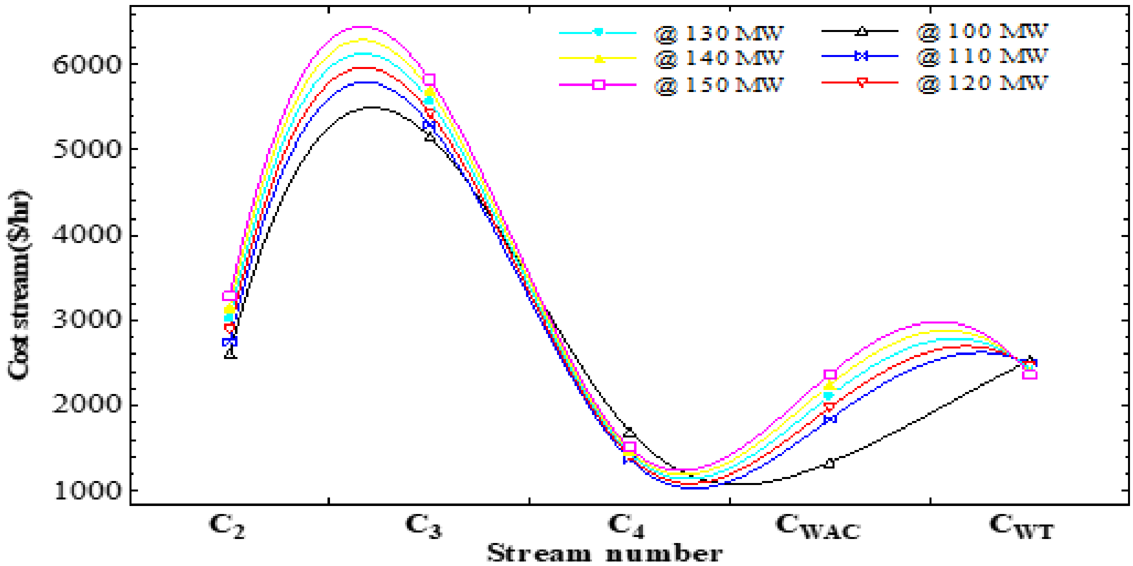

Figure 3: Variation of compressor work on exergetic cost streams

Furthermore, increasing the compressor work results in either an increase or decrease in exergetic cost stream according to the state point. For instance, an increase in the compressor work results in increase of the exergy streams at points 2, 3, 4, that due to compressor work, but with a decrease in the exergetic cost stream of turbine work. In fact, for successive 10 MW increase, the exergy stream at point 2 increased by 142 $/hr. at 110 MW from 100 MW, 138 $/hr. at 120 MW from 110 MW, 135 $/hr. at 130 MW from 120

MW, 130 $/hr. at 140 MW from 130 MW and 128 $/hr. at 150 MW from 140 MW. The cost of the streams was as expensive as the temperature of the state points at any compressor power input. As the compressor work increased, the cost streams before and after the combustion chamber increased resultantly as these hot gasses expands in the turbine. However, these cost rates are rather smaller at the exit of the turbine due to comparatively small approximations in exhaust gasses temperature.

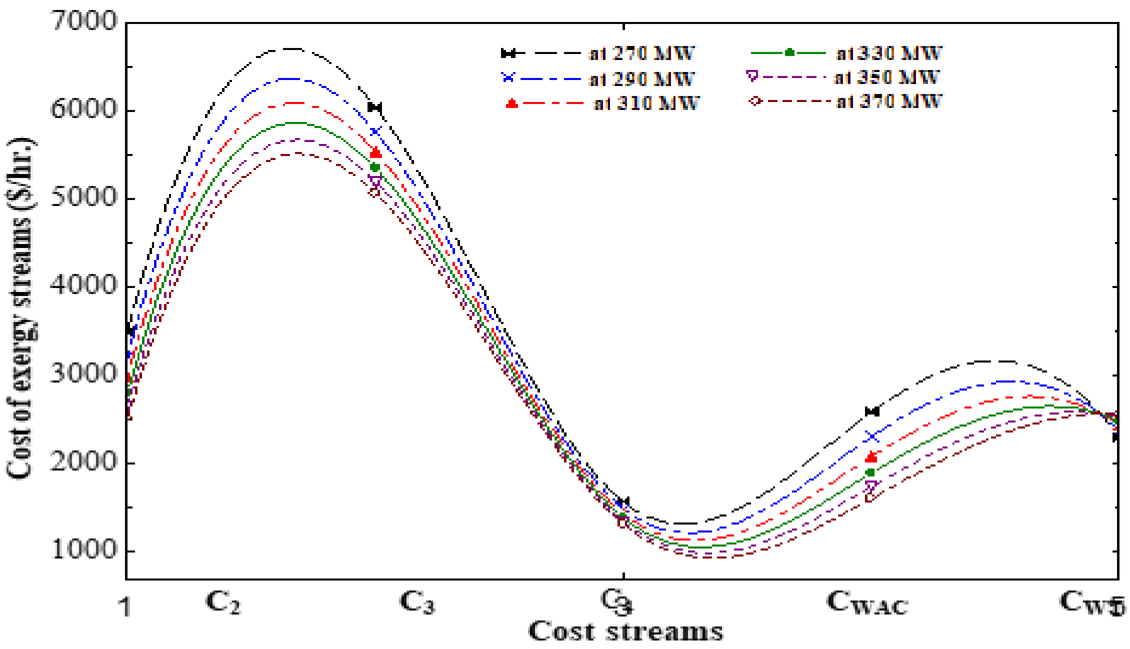

Figure 4: Effect of turbine output on exergetic cost streams

The effect of the turbine output on the cost streams is seen to vary significantly from that of the compressor (Fig. 4). The cost stream at entry to the turbine is the most expensive for the considered case. The stream at entry to the turbine relates well with the turbine output and is estimated at 1560 $/hour, 1487 $/hour, 1429 $/hour, 1381 $/hour, 1341 $/hour, and 1307 $/hour for 270MW, 290MW, 310MW, 330MW, 350MW and 370MW of turbine output respectively. The trend of the cost streams at entry to the turbine rather decreases as the work output increases, since the work potential at higher turbine output is relatively high. For a 20 MW increase in turbine output (from 270 MW to 290 MW), the cost streams reduced by 73 $/hour.

5. Conclusion

Exergoeconomic analysis of a gas turbine plant with compressor inlet air fogging was carried out. A case study of an in-service 150 MW gas turbine plant was used for the thermodynamic considerations. The obtained results for the unfogged and fogged gas turbine plants are detailed as follows: The exergetic efficiency of the two plants stood at 41.14 % for the unfogged and 42.5 % for the fogged plant with efficiency improvement of 1.36 %.

References

1. Abam, F. I., Ugot, G.U, Igbong, D. I., (2012). Performance analysis and components irreversibilities of a (25 MW) gas turbine power plant modelled with a spray cooler, *American Journal of Engineering and Applied Sciences* 5(1),35-41.
2. Alhazmy, M. M., Jassin R. K, Zaki, G. M. (2006) Performance enhancement of gas turbine by inlet air-cooling in hot and humid climates. *International Journal of energy Resources* 30:777-797.
3. Balli O., Aras H., Hepbasli A. (2008) Exergoeconomic analysis of a combined heat and power (CHP) system. *International Journal of Energy Research*. 32:273-89.
4. Ebabi, M. J, Gorj, M. H. (2005) Exergetic analysis of gas turbine plants. *International Journal of Exergy*. 2(1): 285-90.
5. Ehyaei, M. A., Mozafari, A., Alibiglou, M. H. (2011). Exergy, economic and environmental (3E) analysis of inlet fogging for gas turbine power plant, *Energy* 36:6851-61.
6. Frangopoulos, C. A. (2018). Recent developments and trends in optimization of energy systems. Energy.
7. Frangopoulos, C. C. (1991) In analysis of thermal and energy systems – proceedings of the international conference ATHENS'91, pp. 305-318, Kouremenos D. A., Tsatsaronis G., and Rakopoulos C. D., eds., ASME, New York.
8. Gorji-Bandpy, M., Goodarian, H., Biglari, M. (2010) Cost effective analysis of a gas turbine power plant. *Energy sources*, part B: Economics, planning, and policy, 5(4): 348-358.
9. Homji-Meher, B. C., Mee, T., Thomas, R. (2002) Inlet fogging of gas turbine engines, Part B: Droplet sizing analysis, nozzle types, measurement and testing, Proceedings of the ASME Turbo Expo 2002, Amsterdam, Netherlands, June 2002 Paper No: GT-30563.
10. Jaber, Q. M, Jaber, J.O, Khawaldah, M. A. (2007), Assessment of power augmentation from gas turbine power plants using different inlet air cooling system, *Jordan Journal of Mechanical and Industrial Engineering*, Vol.1, No.1, 7-15.
11. Jassim, R. K, Zaki, G.M and Alhazmy, M.M. (2009) Energy and exergy analysis of reverse brayton refrigerator for gas turbine power boosting, *International Journal of Energy*, Vol. 6, No. 2, 143-165.
12. Jassim, R. K., Zaki, G. M., Allazng, M. M. (2010) Thermo-economics analysis of gas turbines power plants with cooled air intake, *Yanbul Journal of Engineering and Science*, October 2010, Vol. 1:26-41.
13. Jonsson, M., Yan, J. (2005) Humidified gas turbines – A review of proposed and implemented cycle, *Energy* 30:1013-78.
14. Kotas, J. J. (1995). The exergy method in thermal power plant analysis. Reprint edition, Malabar: Krieger.
15. Lazzaretto, A., Tsatsaronis, G. (2006) SPECO: A Systematic and general methodology for calculating efficiencies and costs in thermal systems. *Energy* 31: pp. 1257-1289.
16. Levenda, A. M., Behrsin, I., & Disano, F. (2021). Renewable energy for whom? A global systematic review of the environmental justice implications of renewable energy technologies. Energy Research & Social Science, 71, 101837.
17. Marzouk, A., Hanafi, A. (2013) Thermo-economic analysis of inlet air cooling in gas turbine plants, *Journal of Energy Technology* 93 (2), pp: 90-99.
18. Oyedepo, S. O., Kilanko O. (2014) Thermodynamic analysis of a gas turbine power plant modeled with an evaporative cooler, *InternationalJournal of Thermodynamics,* Vol. 17 (No.1), pp: 14- 20.
19. Sayyaadi, H., Sabzaligol, T. (2009) Exergoeconomic optimization of a 1000MW light water reactor generation system. *International Journal of Energy Research*. 33(4):378-95.