RESEARCH ARTICLE

Efficiency Analysis of the Turbine using Calorific Value Parameters for a 10 Megawatt Gas Turbine

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Abstract: This research is based on the analyzed thermodynamics of a gas turbine energy plant. It considered the differing operating states, i.e. the ambient temperature, the compressor outlet temperature, pressure ratio, etc. and their affective modes on thermal efficiency, turbine work, compressor work, etc. which were derived and analyzed. The Gross (higher) calorific values at constant pressure ($\Delta h_{(liq)}$) heat of combustion in a flow process from state 1 to state 2 was considered and used to analyze our thermal efficiency. Ambient temperature cum air/ fuel ratio affected the turbine work, compressor work and system efficiency. In addition, the thermal efficiency nd power decreases linearly with increase of the ambient temperature. However, the efficiency analyzed when the calorific parameters were considered was higher than the efficiency when the basic thermodynamic theories (first and second law principles) were used. The first ranges between 31% to 33, while the second ranges between 28% to 32% under the same ambient temperature conditions.

Keywords: Efficiency, Calorific Parameters, Energy Analysis, Turbine work

1. Introduction

With the improvement in industrialization and people, there has been an extending interest for electrical energy in Nigeria. Power age in Nigeria is transcendently from three hydro-electric power stations, steam and gas warm stations. An enormous bit of these workplaces are being directed by the Nigerian Electricity Regulatory Commission (NERC), an organization guaranteed administration association that co-ordinates all activities of the power region be it creation, transmission, movement, or exhibiting and bargains. The free regulatory office, as given in the Electric Power Sector Reform Act of 2005 was depended with checking and coordinating the Nigerian power industry, with offering licenses to exhibit individuals, and with ensuring consistence with market oversees and working guidelines.

Oyedepo & Kalinko (2014) [1] did Thermodynamic examination of a gas turbine power plant exhibited with an evaporative cooler. in their dissemination the show improvement of gas turbine power plants by cooling the blower utilization air with an evaporative cooler is thought of. This assessment investigated the effect of bay air cooling structure on the introduction of a current gas turbine power plant in Nigeria. The results show that for each 5°C decay of delta air temperature, net yield power increases around 5 – 10% and warm usefulness increases around 2-5%. It is shown that the proportion of this development is higher when the squeezing factor extent is high and the turbine channel temperature is low. The results of this assessment shows that the retrofitting of the current gas turbine plant with channel air cooling structure gives a predominant system execution and may wind up being an engaging endeavor opportunity to the Nigerian government and accomplices of the plant.

Kotowicz et.al., (2016) [2], made on the system out of the gas turbine convenience evaluation. An evaluation thinking for isentropic effectiveness of a blower and turbine in a gas turbine establishment reliant upon polytropic sufficiency qualities was introduced. A gas turbine model was outlined into programming projects for power plant age. The evaluation depended upon assessments from iterative models for isentropic ampleness of the blower and the isentropic ability of the base on the turbine inlet temperature.

The isentropic value credits of the blower and the turbine and made through the as of late referred to assessments. The gas turbine improvement for the high blower degrees was the focal basic role for this

assessment. The got gas electric handiness qualities show that an improvement of pressing variable degree in excess of 50 isn't protected because of the slight expansion in the ability with an essential increase of turbine bay combustor outlet and temperature.

Rahman et.al., (2011) survey thermodynamic execution examination of gas-turbine power plant. The variety of working conditions (pressure proportion, turbine bay and fumes temperature, air to fuel proportion, isentropic blower and turbine proficiency and encompassing temperature) on the exhibition of gas turbine (warm productivity, blower work, power, explicit fuel utilization, heat rate) were researched. The scientific recipe for the particular work and proficiency were inferred and dissected [3].

The outcomes showed that the pressure proportions, encompassing temperature, air to fuel proportion just as the isentropic efficiencies are firmly impacted on the warm productivity. What's more, the warm proficiency and force yield diminishes directly with increment of the encompassing temperature and air to fuel proportion. In any case, the particular fuel utilization and warmth rate increments straightly with expansion in both encompassing temperature and air to fuel proportion. Hence, the thermodynamic boundaries on cycle execution are monetarily possible and gainful for the gas turbine tasks.

The poor performance of these thermal plants has contributed immensely to incessant power outages and economic loss. [4] revealed that only 15 out of the 55 units were available for power generation as at 1994. The 'then Minister of Power and Steel in [5] put the record at 20 out of 55 units (25.6%) were functional before 1999 and by 2001 the percentage had moved to 43.8% with the additional 2 units added while 17 units were under rehabilitation.

For greatest economic benefit, the availability of the most efficient and modem plant must be high. Agbauduta (1998), reported that Nigeria's economic losses from unreliable power generation and supply was put at a staggering N66 billion (equivalent to $0.55 billion). He identified thy lack of qualified maintenance engineers, non-availability of required spare parts and the instability of the national grid among other factors as being responsible for the poor performance of these power stations [6].

Hart (1992) revealed that the improvement of force age and dissemination in Nigeria didn't at first include a public matrix framework and accordingly, the producing plants were not arranged as for such a framework. The foundation of the framework was made essential on the grounds that an enormous piece of the nation didn't have producing stations situated inside them [7].

Energy Analysis of Power Plants

Obodeb & Ugwuoke (2013) did Time-Dependent energy examination of a 120MW steam turbine unit of Sapele Power Plant. The work by and large showed that energy misfortune expanded with expanded activity time. It was seen that disintegration and oldness might be the serious issues and that plant restoration is a possible arrangement [8].

Kwak *et al.,* (2010), checked exergetic and thermo-economic analyses on a 500-MW Power plant running a combined cycle system. In the analysis, they contemplated the protection of mass and energy of every part in the framework [9]. They likewise consider the quantitative equilibrium of exergy and exergetic cost (exergoeconornics) related with every part, and for the framework all in all. In an exergoeconomic model, they addressed the useful construction of the framework to imagine the expense related with the arrangement cycle and the useful connection between parts. Kwak fostered a PC program to decide the creation expenses of force plants using gas or steam turbines, or a blend of the two (cogeneration plant). The prom could likewise be utilized to consider plant qualities, especially thermodynamic execution and framework affectability change. The kind of changes inferred here are in measure and additionally segment plan factors.

Vundela et al. (2010) inspected on energy investigation of nuclear energy stations. They surveyed a thermodynamic examination of a coal based nuclear energy station and gas-based cogeneration power plant as far as energy for the various segments of the force plants in sub-basic reach. They presumed that, the significant energy misfortune was found to happen in condenser. The energy investigation showed that ignition chamber in both steam and gas turbine nuclear energy stations is primary wellspring of irreversibility. The irreversibility in condenser is inconsequential as the bad quality energy is lost in the condenser [10].

Egware and Obanor (2013) investigated energy examination of Omotosho stage 1 gas nuclear energy station. In their examination, they saw that exergy effectiveness diminishes and exergy annihilation productivity increments as surrounding temperature expansions in the plant and its parts [11].

Ahmadi et al., (2011) did Performed Thermodynamic displaying, exergy and exergoeconornic investigation and advancement of a joined cycle power plant. Their examination

demonstrates wise. The exergy obliteration is attributed to the irreversibility's related with the burning response and the warmth move across huge temperature contrasts between the lighted gases and the functioning liquid. Besides, the exergoeconomic examination recognizes the ignition chamber as having the best expense related with exergy annihilation, everything being equal. It is additionally confirmed that expanding the Gas Turbine Inlet Temperature (GTIT), adequately diminishes the proficiency of the turbine [12].

2. Materials and Methods

Calorific Value Analysis

Ordinarily stated, the calorific value of a fuel is the amount of heat liberated when a kilogram mass of fuel is assuredly burnt (synonymous to heat of combustion). However, it is defined with reference to the state (temperature, pressure and volume).

Four types of calorific values are defined

For non-flow process:

(i)   Higher (Gross) calorific value at constant volume, ΔU (liq)

This is $\approx \Delta U_0$ with the H₂O in the product in the liquid phase

$$-\Delta U_0 = -\Delta H_0 + \Delta(PV_0)$$

$$(\because 4 = u + p) \tag{1}$$

If the fuel is a gas

$$-\Delta U_0 = -\Delta H_0 + \Delta(R_0 T_0) \tag{2}$$

(ii)   Lower (Net) calorific value at constant volume, ΔU (vap)

This is $\approx \Delta U_0$ with the water of the product in the vapour phase

$$-\Delta U_0 = -\Delta H_0 + \Delta(PV_0)$$

Note:

$$\Delta U(vap) - \Delta U(Liq) = \eta H_2 0 \left(U'_{fg}\right)$$

$U'_{fg} = U_{fg}$ (latent heat of vapourization of water at constant volume at 25°C).

For flow process:

(iii)   Higher (Gross) calorific value at constant pressure ($\Delta h_{(liq)}$)

This is $\approx -\Delta H_0$ with the H₂O of the product in the liquid phase.

(iv)   Lower (Net) calorific value at constant pressure ($\Delta h_{(Vap)}$)

This is $\approx -\Delta H_0$ with the H₂O of the product in the liquid phase.

Note:

$$\Delta h_{(vap)} - \Delta h_{(lip)} = nH_2 Oh''_{fg} \tag{4}$$

Where $h'_{fg} = h_{fg}$ of H2O at reference temp of (25°C)

Heat of Combustion

Applying first law of thermodynamics to combustion results in heat of combustion.

Two types of heat of combustion are defined.

(i)   Constant volume heat of combustion or internal energy of combustion (ΔU_0), and

(ii)   Constant pressure heat of combustion or enthalpy of combustion (ΔH_0)

(a)   Internal energy of (constant volume heat of) combustion is the heat of combustion generated in the none-flow process.

$$Q- = U_{p2} - U_{R1} \tag{3}$$

State 1	State 0
(reference)	State 2
Reactants (R)	Combustion Product (P)

$$R(P_1, V_1, T_1)$$

$$R(P_0, V_0, T_0)P$$

$$(P_2, V_2, T_2)P$$

$$T_1 \xrightarrow{\text{some therm addition of heat}} T_0 \xrightarrow{\text{Combustion}}$$

$$T_0 \xrightarrow{\text{some therm process cooling}}$$

Some thermal process (addition of heat) some thermal process cooling

The total change of internal energy between state 1 and 2 may be written as:

$$U_{P2} - U_{R1} = (U_{P2} - U_{P0}) + (U_{P0} - U_{R0}) + (U_{R0} - U_{R1}) \qquad (5)$$

Where

$$U_{R0} - U_{R1} = \sum_R \eta_i (\overline{U}_{i0} - \overline{U}_{i1}) = \sum_R m_i (u_{i0} - u_{i1})$$

$$U_{P2} - U_{P0} = \sum_P n_i (\overline{U}_{i2} - \overline{U}_{i0}) = \sum_P m_i (u_{i2} - u_{i0})$$

Where η_i = the amount of substance (mol) of the constituents

m_i = mass of the constituents

$\overline{u}_i$ = Tabulated values of internal energy of the constituents at the required temp T_0 or T_i per mole.

U_i = Internal energy per unit mass

$U_{P0} - U_{R0} = \Delta U_0$ = Heat generated in combustion called the internal energy of combustion at T_0

The reference T_0 is usually taken as 25ºC (298K)

For gaseous reactants and products

$$U_{P2} - U_{P0} = \frac{\sum_P m_i Cv_i (T_2 - T_0) =}{(T_2 - T_0) \sum_P m_i Cv_i} \qquad (6)$$

$$U_{R0} - U_{R1} = \frac{\sum_P m_i Cv_i (T_0 - T_1) =}{(T_0 - T_1) \sum_R m_i Cv_i} \qquad (7)$$

Where m_i is the mass of the various constituents and c_{vi} is the mean specific heat at constant volume for the appropriate temperature ranges T_0 to T_2 and T_0 to T_1.

(b) Enthalpy of (constant pressure heat of) combustion is the heat of combustion generated in a flow process.

Considering combustion in a flow process from state 1 to state 2, from

$$Q - W = H_{P2} - H_{R1} \qquad (8)$$

Total change of enthalpy between states 1 and 2 may be written as:

$$H_{P2} - H_{R1} = (H_{P2} - H_{P0}) + (H_{P0} - H_{R0}) + (H_{R0} - H_{R1})$$

$$H_{R0} - H_{R1} = \sum_R \eta_i (\overline{h}_{i0} - \overline{h}_{i1}) = \sum_R m_i (h_{i0} - h_{i1})$$
Where

$$H_{P2} - H_{P0} = \sum_P n_i (\overline{h}_{i2} - \overline{h}_{i0}) = \sum_P m_i (h_{i2} - h_{i0})$$

Where $\overline{h}$ = tabulated enthalpy value per mole

h = enthalpy per unit mass

$$H_{P0} - H_{R0} = \Delta H_0$$

= heat generated during combustion and it is called the enthalpy of combustion or constant pressure heat of combustion at T_0

For gaseous reactants and products

$$H_{P2} - H_{P0} = \sum_P m_i Cp_i (T_2 - T_0) = (T_2 - T_0) \sum_p m_i Cp_i \qquad (9)$$

$$H_{R0} - H_{R1} =$$

$$\sum_R m_i Cp_i (T_0 - T_1) = (T_0 - T_1) \sum_R m_i Cp_i \qquad (10)$$

Where CP_i = mean specific heat at constant pressure for the ranges T_0 to T_2; T_0 to T_1

ΔH_0 and ΔH_0 are measured when both the combustion reactants and products are both at T_0.

Value of ΔU_0 and ΔH_0 are usually quoted for 25ºC and for 1kg or 1mol.

Also values of ΔU_0 and ΔH_0 are quoted with the phases of reactants and products specified (solid, liquid, gas) e.g.

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O \text{ (liquid)}$$

Or

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O \text{ (vap)}$$
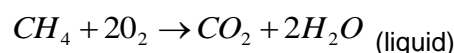
Other reactants and products CH_4, O_2, CO_2 are known to be gases.

If any constituent undergoes a change of phase the appropriate latent heat of vaporization (h_{fg}) must be included.

The difference between the ΔH_0 for a fuel in the liquid phase and ΔH_0 for same fuel in the gaseous phase is the latent heat of vaporization.

Generally for gaseous fuels

$$\Delta H_0 = \Delta U_0 + R_0 T_0 (nP - nR)$$
$$(\because h = u + pv = u + nRT)$$

(11)

Where $R_0 =$ Universal gas constant

= RM; where M = molar mass; R = specific gas constant

NP and nR are number of moles of products and reactants respectively.

Property tables for gases are used to calculate the terms in the equations that relate enthalpy changes.

The mean specific heat for a range of temperature t_1 to t_2 is equal to that for the average temperature $\dfrac{t_1 + t_2}{2}$ only if the specific heat-temperature relation is linear, otherwise accurate calculation should be done with values from the gas tables.

Hence, e.g.

$$H_{P1} - H_{P0} = \sum_P n_i (h_{t1} - h_{t0})$$

(12)

Where h_{t1} and h_{t0} are obtained from gas tables at temperature t_1 and t_0 respectively and n_i = mole of each constituent.

Enthalpy of Formation

Enthalpy of formation of a compound is the heat-transfer obtained as change in enthalpy between the product and reactants when the enthalpy of the all the elements (reactants) are made zero at standard condition of 25°C and 1 bar.

E.g. CO_2 is formed at 25°C and 1 bar from complete combustion of C in O_2

$$C + O_2 \rightarrow CO_2 + (-393,522kj)$$
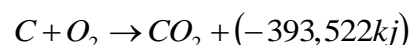
Applying 1st law of thermodynamics

$$Q = H_P - H_R = \sum_P ni\ hi - \sum_R ni\ hi$$

(13)

If the enthalpy of O_2 and C (reactants) = zero (being elements in their normal states) at 25°C and 1bar.

$$Q = H_P = \sum_p ni\ hi = -393,522 kj/mol$$

Then

= the enthalpy of formation of CO2 at 25oC and 1 bar, denoted by h^0_f

Thus (h^0_f) CO$_2$ = 393522 kj/mol

The enthalpy of CO_2 at any other state (any other temperature and pressure) relative to 25°C and 1bar is given by

$$h_{T,P} = h_f^0 \Delta h = h_f^0 + (h_T - h_{298})$$

(14)

= h at any other pressure and temperature.

$$Q = H_P - H_R = \sum_P n_i (h_f^0 + \Delta h)_i - \sum_R n_i (h_f^0 + \Delta h)_i$$

Where $\Delta h =$ change in enthalpy between 25°C(298K), 1bar and the required temperature and pressure $= h_T - h_{298}$

$$h_{T,P} = h_f^0 + (h_T - h_{298})$$

(15)

3. Results and Discussion

Following the procedurals and equations adopted, the outcome of the performance improvement of thermal power systems through energy efficiency analysis was presented. The plant energy efficiency of the gas turbine components has been evaluated.

Presentation of Input Data

Appropriate assumptions have been made for the power plant and stated in table 1 such that the results obtained will closely approximate actual data from the gas turbines.

Table 1. Energy Performance Analysis

S/N	Parameters	Dave Energy
1.	CH$_4$ calorific parameter (kJ/kg)	55208
2.	Net power of turbine	10650
3.	Gas flow rate (kg/hr)	170
4.	Air fuel ratio	63:1
5.	Pressure ratio	15:1
6.	Turbine efficiency	85
7.	Compressor efficiency	80
8.	Ambient temp (TI)	30
9.	Thermal efficiency	31.5

Table 1 presents the results of the energy analysis of the gas turbine. The result shows the energy efficiency

using the calorific parameters with ambient temperature at 30°C. The value for the calorific value was derived using the parameters in table 4.2 and equation 13 to 14 we were able to derive the calorific value for methane.

Table 2. Energy Performance Analysis

S/N	Parameters	Specific heat capacity at 500k (cp kJ/kg K)
1	CH_4	2.293
2	O_2	0.972
3	CO_2	1.014
4	$H_2O_{(g)}$	1.954

The data in table 2 was derived from standard table of specific heat capacity. The parameters are the relevant specific heat capacity (cp) at 500K. These parameters were used in deriving the calorific value of methane CH_4. Using equation 13to 14 in the previous chapter we were able to derive the calorific value for methane.

Table 3. Effect of Compressor Outlet Temperature on Turbine load considmering its Ambient Temperature

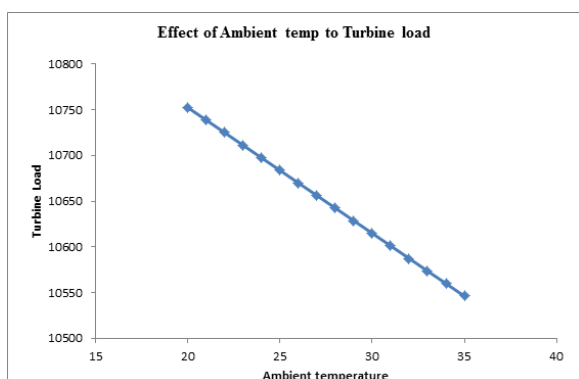| S/N	Ambient temperature (°C)	Turbine load (kW)
1	20	10752.57
2	22	10725.04
3	24	10697.51
4	25	10683.74
5	27	10656.2
6	30	10614.9
7	35	10546.07

Figure 1. Effect of ambient temp to turbine load

From table 3 and figure 1, the results show a significant decrease in turbine load as inlet temperature

increases just as its efficiency is affected when ambient temperature increases. This is due to the density of air reducing at increase in ambient temperature.

Table 4. Comparison between efficiencies of Thermal efficiency principles with that of calorific efficiency with respect to their ambient temperature

S/N	Ambient temperature (°C)	Calorific efficiency	Thermal efficiency
1	20	0.319317	0.317526
2	22	0.318569	0.313509
3	24	0.317821	0.309491
4	25	0.317447	0.307482
5	27	0.316699	0.303465
6	30	0.315577	0.297439
7	35	0.313706	0.287395

The table 4 shows us the varying results of both thermal efficiency and calorific efficiency at different inlet temperatures. Although its efficiencies are decreasing with increase in inlet temperature but their values differ at the same temperature value. The calorific efficiency values are higher than the thermal efficiency values.
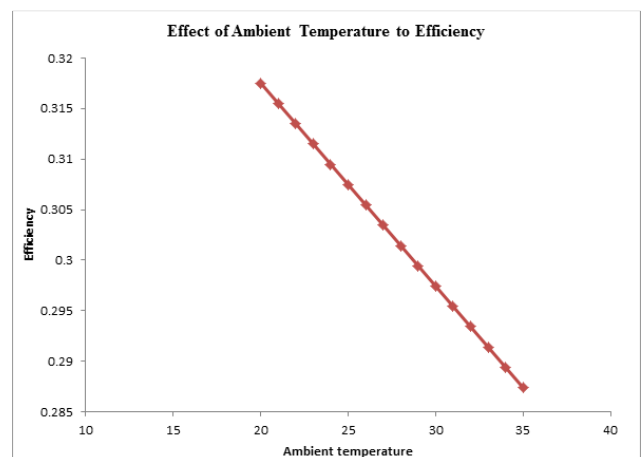

Figure 2. Effect of ambient temperature to efficiency

The efficiency of the gas turbine plant depends on the temperature of the inlet air. At high ambient temperatures there's drop in turbine efficiency. Fig 4 and table 4 considering just the parameters under the thermal efficiency shows the results of the thermal efficiency of the gas turbine with respect to ambient temperature which is one of the objectives of this project. We could see the linear decrease in efficiency

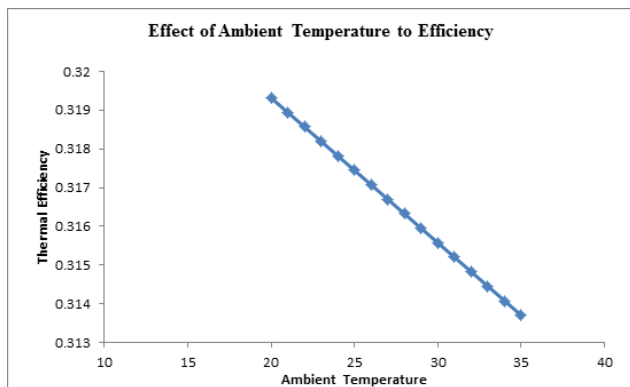as the ambient temperature increases from 20°C to 35°C. There's about 3% to 4% loss in thermal efficiency.

Figure 3. Effect of ambient temperature to efficiency

Increased ambient temperature lowers the density of the inlet air, thus reducing the mass flow through the turbine and therefore reduces the efficiency. At constant speed, where the volume flow remains approximately constant, the mass flow will increase with decreasing temperature and will increase with increasing temperature. Fig 5 and table 4 considering just the parameters under the calorific efficiency shows the results of the efficiency of the gas turbine with respect to its temperature. There's as well a decrease in efficiency of about 2% as the ambient temperature rises.

The figures 2 and 3 compare the efficiency of the turbine. The comparison is between efficiencies when calorific parameters are considered and when they are not at different ambient temperatures. We could see from the above figures that when the calorific parameters are used the efficiency ranges between 31% and 33%, while when calorific parameters are not considered the efficiency ranges between 28% to 32%. Hence, at higher ambient temperatures the efficiencies are lower when the nature of heat of combustion is not put into consideration. This could also be due to some forms of energy losses in the different components.

Comparison with Work of Past Researchers

In the work of Rahman and Thamir (2011) carried out analysis of the effect of ambient temperature, compression ratio and air to fuel ratio, etc. on the thermal efficiency. The present research having analyzed various operational conditions of Dave Energy Power Plant exposed the shortcomings of energy analysis when the Gross Ethalpy of (Constant pressure heat of) combustion is not put into consideration.

However, both analyses are relevant in the performance analysis of a power plant.

While in the work of Kotowicz (2016) carried out analysis on the methodology of the gas turbine efficiency calculation even when the Gross efficiency is considered. The efficiency of the turbine was around 22% while this research ranges from 31%. However, both analysis are still relevant in the performance analysis of gas turbine.

4. Conclusion

The result from the modeling of the influence of parameter showed that ambient temperature, air fuel ratio and turbine outlet temperature effect on performance of gas turbine power plant. The results were summarized as follows:

(i) The thermal efficiency and power output decreases linearly with increased of ambient temperature as well as the air to fuel ratio.

(ii) The variation of thermal efficiency at higher compression ratio, turbine outlet temperature and ambient temperature are very important.

(iii) The nature of the heat of combustion in the turbine with its effect on the turbine efficiency is important and necessary to consider while analyzing.

(iv) The thermal efficiency and power output decreases linearly with increase in ambient temperature.

(v) The peak efficiency and power occur at lower ambient temperature.

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Conflict of interest

The authors have no conflicts of interest to declare that they are relevant to the content of this article.

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