



An Equilibrium Model of NO_x Emission in Gas Turbine Combustors

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Abstract

In this study, an equilibrium model for NO_x emission in gas turbine combustors was developed. The model is based on the governing thermodynamic, materials, and energy balance equations. Results obtained from the model simulation were compared with operational data from a local gas turbine power plant. There was about a 3 to 6 percent deviation for the periods considered. Consequently, the model is reliable in the design of new combustors and in assessing the pollutant emission during operations. It was established that factors that affect NO_x formation include flame temperature and quantity of excess air used. Furthermore, the results revealed that the most important parameter that governs the formation of NO_x is flame temperature, as the formation of NO_x rises exponentially with an increase in the combustor flame temperature.

Keywords: Nitrogen oxide, Combustors, Emissions, Modelling, PEMS

1.0 INTRODUCTION

Oxides of nitrogen (NO_x), carbon monoxide, and other products of combustion have long been identified as harmful to humans and the environment [1]. There are three methods of monitoring NO_x emissions industrially. The first is by periodic measurements, via sampling; the second is by the installation of continuous emissions measuring systems (CEMS), and third by predicting the emissions through algorithm-based software utilizing parameters that affect the pollutant formation. This method is called parametric emissions monitoring system (PEMS). Purchase and maintenance costs of PEMS are cheaper than continuous emission monitoring systems, which require the installation of on-site emission monitoring equipment [1]. Combustion in a gas turbine is essentially clean due to a large amount of excess air and the steady nature of the process. As a result, there have not been many problems associated with gas turbine emissions apart from the need to eradicate smoke from the exhaust [2]. Nevertheless, control of emissions has recently become probably the most important factor in the design of industrial gas turbines, as there has been greater awareness about the causes and effects of pollution as the

number of installed gas turbines increase. Gas turbine designers are continually faced with the challenge of improving the efficiency of the turbine while reducing emissions to meet regulations. This challenge is particularly difficult because the two performance indicators are conflicting: To improve cycle efficiency, a very high maximum temperature is needed. However, higher temperatures favour the formation of major pollutants, especially oxides of nitrogen (NO_x). Nonetheless, reducing available air to lower maximum cycle temperature and hence reduce NO_x formation will lead to incomplete combustion, and therefore increased amount of carbon monoxide (CO) and unburnt hydrocarbon (UHC) emissions produced. Virtually all gas turbine manufacturers use some form of lean-premixed gas turbine combustor; also called dry low NO_x (DLN) combustor, for their gas turbine engines [3]. Thus, DLN is currently the most widely used combustor design method for reducing NO_x and CO emissions in gas turbines. In Lean-Premix combustion, the fuel and excess air are mixed upstream of the combustion zone. This helps to create a homogenous mixture, thereby leading to a reduction in NO_x and CO emissions.

Although it is expected that a stoichiometric mixture of fuel and air gives complete combustion, it is impossible in practice. This is because there is limited time for the combustion process. Consequently, more oxygen than the stoichiometric amount is needed to achieve a complete reaction between oxygen and the combustible

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components in the fuel [4]. Thus, gas turbines are operated with excess air. Using excess air means operating with a quantity of air that is more than the theoretical amount required for complete combustion. Other reasons why gas turbines operate lean mixtures are to reduce the flame temperature and not to exceed the metallurgical limits of the material of construction.

The amount of excess air used in industrial gas turbines typically ranges between 100% - 600% [5]. The maximum temperature achievable in the combustor is at the stoichiometric conditions, and as the amount of air deviates from the stoichiometric requirement, the temperature of the combustion system decreases. The NO_x formation in a lean-premixed combustion system at gas turbine operating conditions mainly depends on the Zeldovich mechanism, which is often also regarded as thermal NO_x. This is the predominant medium of NO_x formation, especially in natural gas-fired gas turbines [6]. Other mechanisms for NO_x formation include the prompt NO_x and the fuel NO_x mechanisms. The fuel NO_x mechanism describes how organically bound nitrogen in the fuel oxidises to NO. Due to small amounts of nitrogen in natural gas, this mechanism is of little significance in natural gas-fired combustion [3]. However, the mechanism becomes important when burning fuels that contain nitrogen compounds such as coal. Nitrogen oxides are produced in the central hot region of the combustion chamber by oxidation of atmospheric nitrogen, and most of the NO_x formed is nitric oxide [6].

There has long been a need to establish methods of evaluating pollutant emissions. The high cost of CEMS has made it difficult to determine the emissions impact of gas turbines, especially in relatively small units. Thus, a parametric emissions monitoring system (PEMS) will more readily inform stakeholders of the emission impact of gas turbines [7]. Several researchers have published some form of parametric models for NO_x emissions in gas turbine combustors. A considerable number of works on parametric emissions modelling of gas turbines was done by Hung [3]. In 1975, a description of an analytical model used to determine NO_x emissions for a specific model of conventional gas turbine combustors; the solar turbine was studied. Furthermore, amendments made to the model to include the effects of water injection, operation at low air/fuel ratio, primary air leaning, fuel with low calorific value, and fuel/air premixing were also studied. Since the distinctive feature in this model is how the diffusion was accounted for, it was ultimately called the diffusion-limited mixing model. The model showed a good agreement with laboratory and industrial data but was only used as a design tool for reducing NO_x in combustors.

Hung's published work on a PEMS where the diffusion-limited mixing model was referenced, but not

directly used. Instead, the PEMS was based on a performance emissions program that was established for various gas turbine models that utilizes field data on NO_x emissions [8]. The program was developed by gathering NO_x data, correcting it to ISO conditions, and establishing the ISO corrected NO_x as functions of fuel/air ratio and water/fuel ratio, where applicable. In 2003, CMC solutions published a pure statistical-based PEMS. This PEMS was developed from past CEMS data. The PEMS was installed on two GE frame 7 gas turbines.

The algorithm was based on an initial 40 to 60 hours of data acquired from a CEMS. CMC solutions didn't however provide the details of the algorithm. They gathered an additional 720 hours of CEMS data which formed the basis of generating a final algorithm. They didn't utilize any fundamental engineering principle in their approach and therefore is strikingly different from the method adopted in this research work. Several semi-analytical expressions have also been developed to determine NO_x emissions from gas turbines [9-12]. The semi-analytical approach usually combines statistical data with parameters derived from governing engineering laws, such as combustor temperature.

Lewis (1991), [11], presented an equation to show the amount of NO_x formed in lean, homogenous combustion. Lewis correlation is given in Equation 1:

$$NOx = 3.32 * 10^{-6} * \exp(0.008Tc) P^{0.5} [ppmv] \quad (1)$$

This equation assumes that NO_x formation is solely dependent on after-combustion temperature and pressure and is completely independent of the combustion gases' residence time. Lewis asserted that this is because the pertinent time is not the combustion product's residence time but the time required for the molecules involved to relax, that is primarily the Nitrogen molecules, and this remains the same for all combustion systems using air as the oxidant. Rokke et al. (1993) [13] developed a NO_x correlation given in Equation 2:

$$NOx = 18.1P^{1.42}m_a^{0.3}FA^{0.72} [ppmv] \quad (2)$$

Their correlation was applied to five natural gas-fired turbines with power output ranging from 1.5 MW to 34 MW and was found to satisfactorily predict the NO_x emissions from them. Although the combustion temperature dependence of NO_x is not apparent in their developed model, its influence is acknowledged by the fuel-air ratio term. Bakken and Skogly (1995) [7] published a correlation that was specifically developed for GE-LM 2500 gas turbines for NO_x prediction given in Equation 3:

$$NO_x = 62 * P_3^{0.5} * (FA)^{1.4} * \exp\left(-\frac{653}{T}\right) \quad (3)$$

P_3 is the compressor discharge pressure, FA is the fuel to air ratio, and T is the combustor discharge temperature. Their correlation was developed from field measurement and simulated data at atmospheric conditions ($T_0 = 15^\circ\text{C}$, $P_0 = 1\text{ bar}$, $\phi_0 = 60\%$). Marshall et al. (1997) [13] published a PEMS model to develop CO and NO_x emission algorithms for small stationary gas turbines in natural gas pipeline compression stations. The general forms of the NO_x and CO predictive algorithms were based on fundamental engineering principles, but the forms of these algorithms were said to be proprietary they presented no details of how their algorithms were developed. In addition, these studies [14-20] have also predicted NO_x emissions in a combustor using different models.

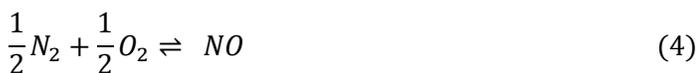
There is currently no universally accurate parametric model, and all available models have their strengths and limitations. Modern engineering is concerned about the environment. Thus, in the design of modern-day gas turbines, the environmental impact should be treated as of equal importance as efficiency and profitability. A reliable parametric model can help engineers to consider environmental impact in their design and simulations. In this study, an equilibrium model for NO_x emissions in natural gas turbines has been developed. The model is validated with actual plant data, and parametric studies are carried out to determine the effect of temperature, pressure, and concentration on NO_x emissions.

The model in this work has relied on fundamental engineering principles, as against the purely statistical or semi-analytical models commonly developed.

2.0 MATERIALS AND METHOD

2.1 Model Development

An equilibrium expression for the fixation of nitrogen and oxygen is given in Equation 4:



Based on Equation (4), an equivalent thermal equilibrium expression for the process is given in Equation 5:

$$K_{eq}^\theta = \frac{(p_{NO})}{(p_{O_2})^{\frac{1}{2}} * (p_{N_2})^{(1/2)}} \quad (5)$$

Where K_{eq}^θ is the thermal equilibrium constant or dissociation constant

p_{NO} is the partial pressure of NO, p_{N_2} is the partial pressure of N_2 and p_{O_2} is the partial pressure of O_2 .

It is noteworthy that nitric oxide (NO) is the primary oxide of nitrogen, and is the oxide formed by high-temperature fixation of nitrogen and oxygen [6]. After NO formation, however, some fraction may get converted to other oxides of nitrogen. Consequently, this work is accounting for oxides of nitrogen formation by considering the quantity of NO producible by thermodynamic equilibrium.

Equation 5 can be rewritten as shown in Equation 6:

$$p_{NO} = K_{eq}^\theta * (p_{O_2})^{\frac{1}{2}} * (p_{N_2})^{\frac{1}{2}} \quad (6)$$

From Dalton's law of partial pressure given in Equation 7:

$$p_i = \frac{n_i}{n_t} * P_t \quad (7)$$

Where: p_i is the partial pressure of a component, P_t is the system total pressure; n_i is the number of moles of a component, and n_t is the total number of moles of all components in the system.

Relating Equations 6 and 7, an equivalent expression becomes:

$$\frac{n_{NO}}{n_t} * P_t = K_{eq}^\theta * \left(\frac{n_{O_2}}{n_t}\right)^{\frac{1}{2}} * P_t^{\frac{1}{2}} * \left(\frac{n_{N_2}}{n_t}\right)^{\frac{1}{2}} * P_t^{\frac{1}{2}} \quad (8)$$

On simplification of Equation 8, a resulting equation is given by:

$$\frac{n_{NO}}{n_t} * P_t = K_{eq}^\theta * (n_{O_2})^{\frac{1}{2}} * (n_{N_2})^{\frac{1}{2}} * \frac{P_t}{n_t} \quad (9)$$

By further simplifying Equation 9; Equation 10 is derived and is given by:

$$\frac{n_{NO}}{n_t} = K_{eq}^\theta * \frac{(n_{O_2} * n_{N_2})^{\frac{1}{2}}}{n_t} \quad (10)$$

Since mole fraction of a component ' y_i ' is given in Equation 11:

$$y_i = \frac{n_i}{n_t} \quad (11)$$

Equation (10) can be rewritten as shown in Equation 12:

$$y_{NO} = K_{eq}^\theta * \frac{(n_{O2} * n_{N2})^{\frac{1}{2}}}{n_t} \tag{12}$$

To derive an expression for the equilibrium constant K_{eq}^θ , experimental data for the operating temperature range, extracted from thermodynamic and transport properties of fluids tables, compiled by [21] is presented in Table 1.

Table 1: Natural logarithm of equilibrium constant for NO

Temperature (K)	$\ln K_{eq}^\theta$
1000	-9.353
1200	-7.541
1400	-6.245
1600	-5.273

Analysis of the data on Table 1 with Microsoft Excel gives a good correlation for $\ln K_{eq}^\theta$ as depicted in Figure 1.

From the correlation Equation 13 in Figure 1,

$$\ln(K_{eq}^\theta) = 8.6957\ln(T) - 69.32 \tag{13}$$

Consequently,

$$k_{eq}^\theta = \exp^{(8.6957\ln(T)-69.32)} \tag{14}$$

Combining Equations 12 and 14, the expression developed for NO_x in this research work is given in Equation 15:

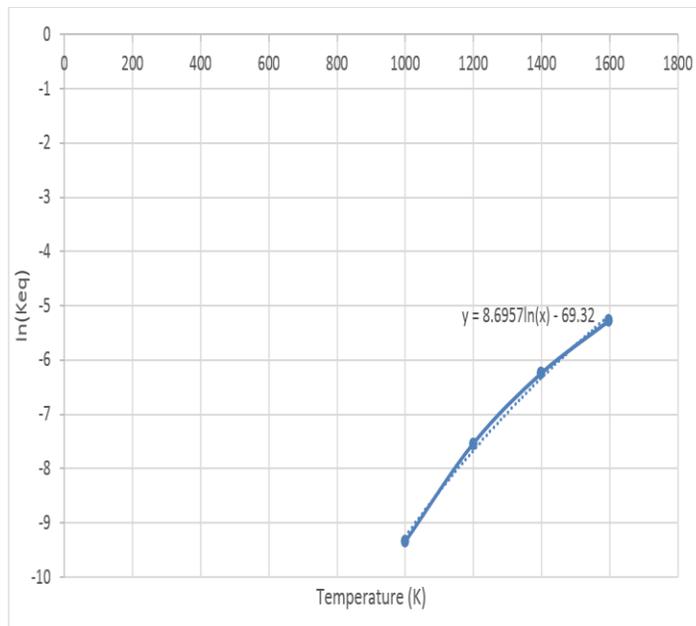


Figure 1: NO_x correlation of $\ln K_{eq}^\theta$ with temperature

$$y_{NOx} = \exp^{(8.6957\ln(T)-69.32)} * \frac{(n_{O2} * n_{N2})^{\frac{1}{2}}}{n_t} \tag{15}$$

Where y_{NOx} = mole fraction of NO_x in the exhaust, T = combustor temperature (K) n_{O2} = amount of oxygen in products stream (kmol/s), n_{N2} = amount of nitrogen in products stream (kmol/s), n_t = total amount of products (kmol/s).

The assumptions in the application of this model include: Firstly, the oxygen available for thermal fixation with nitrogen to form NO_x is the excess oxygen in the combustion chamber after the amount required by stoichiometry has reacted. Secondly, the NO_x production takes place after the main combustion has occurred and the adequate temperature has been achieved to enable nitrogen and oxygen fixation.

2.1 Plant Description

The gas turbine considered in this study is one unit of a 4*112.5 MW, simple cycle gas turbine power plant shown schematically in Figure 2. It was completed in 2013, and gas is supplied through the Excravos-Lagos Pipeline System. It is located in Edo State Nigeria. The plant operating condition is given in Table 2.

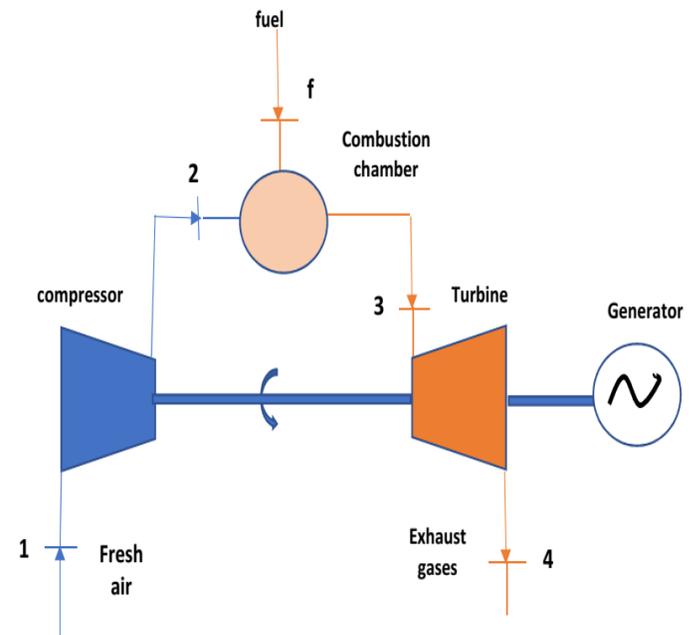


Figure 2: Schematic of the simple gas power plant

Table 2: Plant Operating Conditions:

S/N	Operating Parameters	Value	Unit
1	Air temperature into compressor	303	K
2	Air flowrate into compressor	376.75	Kg/s
3	Air pressure into compressor	101.33	Kpa
4	Air temperature out of compressor	631.5	K
5	Outlet pressure of air from compressor	973.7	Kpa
6	Fuel gas (natural gas mass flow rate)	6.7	Kg/s
7	Air/fuel ratio at full load	56/1	
8	Inlet pressure of fuel gas	22800	kPa
9	Inlet temperature of fuel gas	328	K
10	Outlet temperature from combustion chamber	1263.5	K
12	Maximum exhaust temperature of turbine Outlet	832	K
13	Lower heating value of fuel	45390	kJ/kg
14	Compressor isentropic efficiency	87.8	%
15	Turbine isentropic efficiency	89.4	%
16	Combustion efficiency	99	%

2.2 Fuel

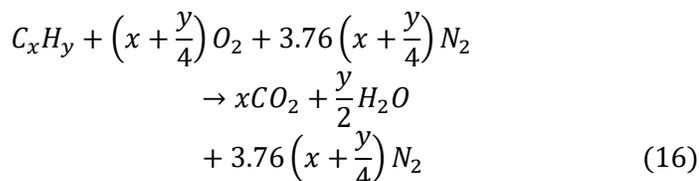
The plant under consideration in this study is fired by natural gas. The composition is given in Table 3.

Table 3: Fuel Composition

S/N	Components	Molar Composition (%)
1	Nitrogen	0.73
2	Methane	86.66
3	Carbon dioxide	3.12
4	Ethane	6.08
5	Propane	2.13
6	i-Butane	0.39
7	n-Butane	0.4
8	n-Pentane	0.26
9	n-Hexane	0.23
TOTAL		100

2.3 Materials and energy balance

The equation for complete combustion of hydrocarbon given in Equation (16):



This is used to determine the material balance of the

combustibles, assuming complete combustion.

Also, a constant pressure combustor is assumed for the energy balance and a reference temperature of 25 °C was chosen. The constant pressure combustion equation is given in Equation (17):

$$H_{P(T_2)} - H_{P(25)} + \Delta H^0_{RX(25)} + H_{R(25)} - H_{R(T_1)} = 0 \quad (17)$$

Where

$\Delta H^0_{RX(25)}$ is the enthalpy of reaction at 25 °C, $H_{P(T_2)}$ =Enthalpy of products at combustor exit, $H_{R(T_1)}$ =Enthalpy of reactants at combustor inlet, $H_{P(25)}$ = Enthalpy of products at 25 °C, $H_{R(25)}$ = Enthalpy of reactants at 25 °C. (18)

$$H = m * C_p * \Delta T \quad (19)$$

C_p = specific heat capacity at constant pressure, and ΔT = change in temperature

3.0. RESULTS AND DISCUSSION

This section presents the materials and energy balance streams. Also, the model developed as given by Equation (15) was solved, and appropriate charts were plotted with the MATLAB software. Tables 4 to 9 presents the materials and energy balance results for all the streams

i-Butane	-134600	58.123	0.07954519 8	0.14305008 3	$C_4H_{10} + 6.5O_2 = 4CO_2 + 5H_2O$	-2648580	3624.758	1.93 5	0.03691 5	- 0.000011402
n-Butane	-126150	58.123	0.08158481 8	0.14305008 3	$C_4H_{10} + 6.5O_2 = 4CO_2 + 5H_2O$	-2657030	3729.562	1.67 7	0.03785 3	- 0.000011945
n-Pentane	-146500	72.15	0.06582805 5	0.11523908 5	$C_5H_{12} + 8O_2 = 5CO_2 + 6H_2O$	-3272020	2985.318	2.46 4	0.04535 1	- 0.000014111
n-Hexane	-167300	86.177	0.06955374 9	0.09648166 0.18892297	$C_6H_{14} + 9.5O_2 = 6CO_2 + 7H_2O$	-3886560	3136.856	3.02 5	0.05372 2	- 0.000016791
CO ₂	-393520	44.01	0.48184493 7	0.18892297 2				5.45 7	0.00104 5	- 115700
Nitrogen	0	28.014	0.07176279 1	0.29679803				3.28 3	0.00059 3	4000
TOTAL										303202.1

Table 6: Energy Balance Stream: Fuel (Cont:d)

	m.C _p Coefficients				mC _p at t = 298.15k	C _p at t = 298.15K	m.C _p at t = 328K
	Constant	T	T ²	T ⁻²			
Methane	4.30344	0.022961	-5.47159E-06	0	10.66285839	2.185586683	11.24597558
Ethane	0.200634	0.00341	-9.86493E-07	0	1.129756507	1.760947899	1.213119314
Propane	0.075384	0.001789	-5.48381E-07	0	0.559993446	1.698998357	0.603142211
i-Butane	0.022018	0.00042	-1.29743E-07	0	0.135724036	1.706250533	0.145837676
n-Butane	0.019572	0.000442	-1.39407E-07	0	0.138893647	1.702444768	0.149474937
n-Pentane	0.018692	0.000344	-1.07046E-07	0	0.111749041	1.697589911	0.120017626
n-Hexane	0.0203	0.000361	-1.12679E-07	0	0.117769452	1.693215005	0.126424649
CO ₂	0.496759	9.51E-05	0	-10532.4	0.406638798	0.843920453	0.430062419
Nitrogen	0.069861	1.26E-05	0	85.19622	0.074585045	1.039327534	0.074795556
TOTAL	5.226659	0.029834	-7.49534E-06	-10447.2	13.33796836		
					H _{f(298.15)} =	3976.715268	
					H _{f(328k)} =		4627.702789

Notes:

H_{f (298.15)} = enthalpy of fuel at 298.15K

H_{f (328)} = enthalpy of fuel at 328K

Table 7: Energy Balance Stream: Air

Com p.	GV Comp	Act. Mass Flow (kg/s)	C _p =R(A + BT + CT ² + DT ⁻²)				ΔH _r (298)	R (kJ/kmolK)	m.C _p Coefficients			C _p at t = 298.15K	m.C _p at t = 631.5K
			A	B	C	D	8	olK	Constant	T	T ²	T ⁻²	5K

O ₂	0.233	87.78 275	3.63 9	0.0005 06	2270 0	0	0.25983 6	83.00 246	0.011 541	0	517767.4 808	80.61895	0.918 392	88.99 253
N ₂	0.767	288.9 673		0.0005 93		0	0.29679 8	281.3 089	0.050 859	0	343059.6 416	300.3316	1.039 328	314.2 864
TOT		376.7						364.3	0.062		174707.8			403.2
AL		5						114	4	0	392	380.9506		789
											H_{a(298.15)} =	113580.4	H_{a(631.5)} =	25467 0.6
											H_{r(298.15)} = H_f+H_a (298.15) =	117557.1		
											H_{r(Tin)} = H_{a(631.5)}+ H_{f(328)} =	259298.3188		

Notes:

H_{a(298.15)} = enthalpy of air at 298.15K

H_{a(631.5)} = enthalpy of air at 631.5K

H_{r(Tin)} = enthalpy of reactants at inlet conditions = enthalpy of fuel at inlet + enthalpy of air at inlet

Table 8: Energy balance Stream: Products

PRODU CTS	MOL AR MASS (kg/k mol)	MAS S FLO W (kg/s)	C _p =R(A + BT + CT ² + DT ⁻²)				ΔH _f (298).kJ/ kmole	Specific gas constan t (kJ/km ol.K)	m.C _p Coefficients				m.C _p at t = 298.1 5K	C _p at t = 298.1 5K	ΔH _f (298) .kJ
			A	B	C	D			Const ant	T	T ²	T ⁻²			
CO ₂	44.01	17.63 213	5.4 57	0.001 045	1157 00	-393520	0.18892 3	18.17 789	0.003 481	0	385409. 8725	14.88 011	0.843 92	1576 60	
H ₂ O	18.015	13.09 861	3.4 7	0.001 45	1210 0	-241820	0.46153 2	20.97 764	0.008 766	0	73149.7 0796	24.41 408	1.863 867	1758 26	
N ₂	28.014	289.0 39	3.2 8	0.000 593	4000	0	0.29679 8	281.3 788	0.050 871	0	343144. 8378	300.4 062	1.039 328	0	
O ₂	31.999	63.67 989	3.6 39	0.000 506	2270 0	0	0.25983 6	60.21 214	0.008 372	0	375601. 9781	58.48 308	0.918 392	0	
TOTAL		383.4 496						380.7 464	0.071 491	0	344717. 3048	398.1 835		3334 86	
											H_{p(298.15)} =	11871 8.4			

Note: H_{p(298.15)} = enthalpy of products at 298.15K

Table 9: Products Flow Stream

Component	Molar Flow (kmol/s)	Molar Mass (kg/kmol)	Mass Flow(kg/s)	Molar Flow (kmol/h)	Mass flow (kg/h)
CO ₂	0.400639106	44.01	17.632127	1442.300783	63475.65747
H ₂ O	0.727094884	18.015	13.098614	2617.541581	47155.01158
N ₂ (Fuel +Air)	10.31766305	28.014	289.03901	37143.58699	1040540.446
O ₂ (Excess)	1.99005865	31.999	63.679887	7164.211141	229247.5923
TOTAL	13.43545569		383.44964	48367.6405	1380418.707

Table 10: Model comparison with Plant data

Month	NO _x Conc. (ppm)	Developed model (ppm)	Deviation	Percentage Deviation
April 2018	135	127.779	-7.221	5.348
May 2018	132	127.779	-4.221	3.198
June 2018	136	127.779	-8.221	6.044

Energy balance Equation;

$$H_{P(T2)} - H_{P(25)} + H_{P(25)} - H_{r(25)} + H_{r(25)} - H_{r(Tin)} = Q + W = 0$$

$$H_{P(T2)} - H_{P(25)} + \Delta H_{r(25)} = H_{r(Tin)} - H_{r(25)}$$

$$H_{P(T2)} = H_{r(Tin)} - H_{r(25)} - \Delta H_{r(25)} + H_{p(25)}$$

$$390.746439 \times T + 0.071491T^2 - 344717.3048T^{-1} = 259298.319 - 117557.1274 - (-303202.1) + 118718 + 380.7464397T + 0.071491T^2 - 344717.3048T^{-1} = 563661.2916$$

On solving the non-linear equation, T = 1207.4 K

Assuming that there is a maximum 3% absolute temperature drop due to heat losses and dissociation, *Combustor temperature* = 0.97 × 1207.4.

Therefore, Combustor temperature = 1171.178 K

In this Microsoft Excel model, non-linear C_p equations taken from [22] have been utilized, and no approximation has been made in the enthalpies of reactions and other parameters.

3.1 Comparison with Plant Data

Table 10 presents the result of the solving Equation

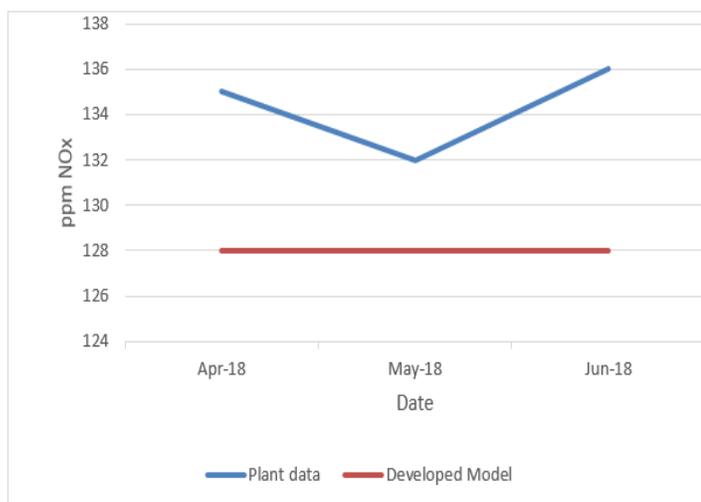


Figure 3: Developed model against plant data.

(15) and the deviations from a measured emission from the

plant. Figure 3 shows the NO_x concentration plot for three months. As observed, the concentration remains constant, for the model developed in this study, while for the plant data, NO_x concentration decreases up to 135-132 ppm, and then increases back to 136 ppm. boilers as reported by [1].

Table 10 shows that there is a close agreement between the equilibrium model developed for NO_x emission and actual plant data. The calculated value of NO_x gives 5.35%, 3.20%, and 6.04% deviation from actual plant data in April, May, and June 2018 respectively. In addition, this result confirms that thermal NO_x is predominant in nitrogen oxides formation mechanism, especially in natural gas-fired

3.2 Effect of temperature on NO_x emission

A graphical representation of the effect of temperature on NO_x emission is given in Figure 4. The graph was plotted with MATLAB; from the solution of Equation (15) at different combustor temperatures.

Figure 4 depicts the importance of combustor temperature to NO_x concentration in the flue gas. An increase in combustion temperature causes an exponential rise in NO_x emissions. At 1000 K for instance, the concentration of NO_x is only 32.34 ppm. However, when the combustion temperature becomes 1600 K, the

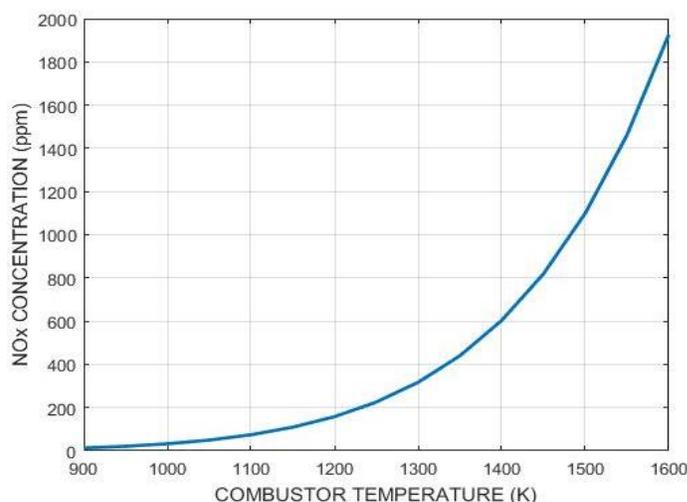


Figure 4: Effect of Temperature on NO_x Emission

concentration of NO_x becomes 1926 ppm. Consequently, the combustion temperature is a very important parameter

in the NO_x emissions control strategy. The flame temperature must be carefully monitored to ensure that NO_x concentration in the exhaust gases is kept within the required level.

3.3 Effect of N₂ Concentration on NO_x Emission

Figure 5 depicts the effect of increased N₂ concentration in the combustion chamber on NO_x emission. The graph is a result of solving Equation (15) with MATLAB, at different nitrogen concentrations in the combustion chamber.

It was observed that NO_x emission increases with an increase in N₂ molar concentration. At a constant temperature of 1171.18 K, when the N₂ concentration in the combustor is 76.8%, NO_x emission is 127.8 ppm. As the concentration of N₂ increases to 83.5%, NO_x emission increases to 133.2 ppm and finally, as the concentration of N₂ becomes 92.4%, NO_x emission becomes 140.2 ppm. The relationship between N₂ molar concentration in the combustion chamber and NO_x formation is nearly linear.

Thus, reducing the concentration of N₂ in the flue gas can help to decrease the NO_x produced. Additionally, it can be observed from Equation (15) that when N₂ concentration becomes zero, no NO_x will be formed at all. This means that using oxygen for combustion instead of air can eliminate NO_x formation. However, this will lead to increased operational expenses and reduced profit for the organization.

3.4 Effect of pressure NO_x emission

Although the importance of pressure on NO_x emission seems intangible, as there is no pressure component in Equation (15), higher compressor discharge pressure can increase the flame temperature and thereby increase the formation of NO_x.

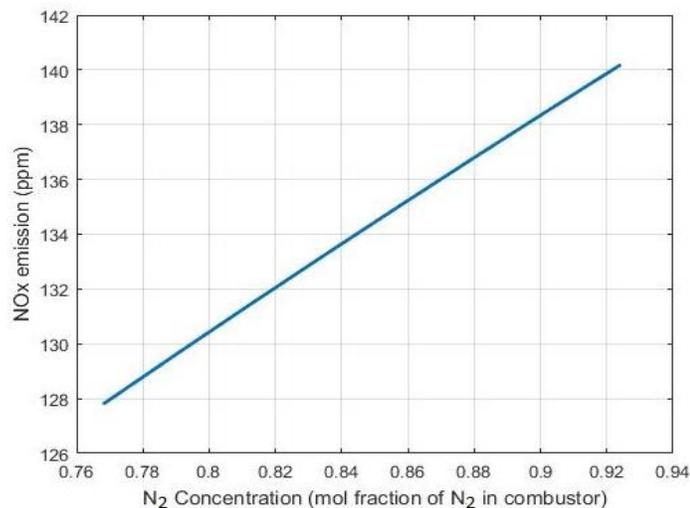


Figure 5: Effect of N₂ Concentration on NO_x emission

From the factors observed, it is deduced that the most important parameter affecting NO_x formation is the combustor temperature.

4.0 CONCLUSION

A thermal equilibrium model for NO_x formation in natural gas-fired gas turbines has successfully been developed in this work. The close agreement between local gas turbine emissions data and the developed model confirms that thermal NO_x is the predominant formation mechanism in natural gas-fired gas turbine combustors. The main parameter that affects NO_x formation is temperature. As the combustion temperature reduces, NO_x emission significantly declines. However, low temperature leads to reduced cycle efficiency and power output. Therefore, there has to be a compromise between cycle efficiency and combustor outlet temperature. The models developed in this research can be a valuable tool in the design of effective NO_x pollutant prevention and control strategies as gas turbine operating conditions become more stringent. A careful study and simulation of the equation developed in this research-work can help designers and engineers determine optimum conditions for minimum pollutant emission while maintaining suitable operational efficiency.

NOMENCLATURE

A/F	Air to fuel Ratio
AMW	Average molecular weight (kg/kmol)
CEMS	Continuous emissions monitoring system
C _p	Specific heat capacity at constant pressure (kJ/kgK)
CO	Carbon monoxide
DLN	Dry Low NO _x
F/A	Fuel to air ratio
H	Enthalpy (kJ)
K _{eq}	Equilibrium constant
m	Mass flow rate (kg/s)
MW	Molecular weight (kg/kmol)
N	Number of moles (kmol)
NIPP	National Independent Power Project
NO _x	Oxides of nitrogen
P	Pressure (bar)
PEMS	Parametric emissions monitoring system
ppm	Parts per million
ppmv	Parts per million, by volume
T	Temperature (K)
UHC	Unburnt hydrocarbon
V	Volume (m ³)
W	Workdone (kJ)
y	Molar concentration (kmol/kmol)
Φ	Equivalence ratio
Δ	Change
ρ	Density (kg/m ³)

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