Sudan University of Science and Technology

College of Graduate Studies

A Design of Water Injection System for Gas Turbine by Numerical Analysis at Garri Power Plant

تصميم نظام حقن مياه لتوربين غازي عن طريق التحليل العددي في محطة كهرباء قري

A thesis submitted in partial fulfillment of the requirements for the degree of M.Sc. in Mechanical Engineering (power)

Prepared by

Abd Elazim Awad Allah Elebad Mohammed

Supervised by

Dr. Abdullah Mokhtar Mohammed

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استهلال

قال تعالى:

(وَقُلِ اعْمَلُوا فَسَيَرَى اللَّهُ عَمَلَكُمْ وَرَسُولُهُ وَالْمُؤْمِنُونَ)

صدق الله العظيم

[سورة التوبة: الآية 105]

Dedication

TO MY FATHER' SPIRIT

TO MY MOTHER GAVE ME HER LIFE AND GAVE ME HER SATISFACTION

TO MY WIFE'S AND SMALL FAMILY

TO WHO HELPED ME FROM MY BROTHERS AND SISTERS?

TO ALL MY TEACHERS

TO COLLEAGUES AND EMPLOYEES AT GARRI POWER STATION

TO GRADUATE STUDENTS

TO THEM ALL I DEDICATE MY RESEARCH TO THEM

Acknowledgments

FIRST, I AM VERY GRATEFUL TO ALL THOSE WHO SHOWED THEIR SUPPORT DURING THE COURSE OF THIS PROJECT, ESPECIAL MY FAMILY.

I ALSO APPRECIATE ALL THE LESSONS I LEARN FROM MY TEACHERS AT THE SUDAN UNIVERSITY OF SCIENCE AND TECHNOLOGY, LAST BUT NOT LEAST, I WANT THANK **DR. ABDALLA MOKHTAR** FOR HIS SHARING HIS INVALUABLE KNOWLEDGE AND ALWAYS GUIDING ME TOWARDS THE PATH OF SUCCESS.

Abstract

The Thermal power plants play an important role in converting chemical energy to electrical and heat energy.Gas turbines are used for power electrical generation, operating airplanes and for several industrial applications.At garri electrical stationThe gas turbine designed for power output 42 MW but because of many reasons (as specified by the International Standardization Organization (ISO) Therefore, the air inlet conditions are: air temperature 15 °C, relative humidity 60%, absolute pressure 101.325 k Pa at sea level) the actual power decreased to 30 MW, and the Environmental pollution increased according to the type of fuel used. So we will try the possibility of using a water injection system to avoid the pollutions (NO_x, SO_x, CO) and also improving the over hall performance and efficiency.

The objectives of this research are To Increase output power & efficiency produced from gas turbine and To Decrease environmental impact.

Gas turbine power enhancement technologies such as water injection may have on the operational integrity of the gas turbine to reducing NO_x formation also to reduce the flame temperature by introducing a water or steam into the flame zone. Both water and steams are very effective at achieving this goal. Generally, the amount of water is limited to the amount required to meet the NO_x requirement in order to minimize operating cost. The results show that using water injection systems increased Net work output, efficiency and the heat rate. Without using water injection the total Network output was (30MW) and the efficiency about 29.3% but when we used the same flow rate of fuel and water .total Network output became 31.7MW an increase about (5.67%) also the efficiency become (31%)an increase about(5.8%)when we use water flow rate equal half of fuel flow rate total Net work output became 30.9MW an increase about (3%)also the efficiency become(31.2%)an increase about(6.47%) that's mean the total work and the efficiency directly affected with the percentage of the water injection.

المستخلص

تلعب محطات الطاقة الحرارية دورًا مهمًا في تحويل الطاقة الكيميائية إلى طاقة كهربائية وحرارية. تستخدم التوربينات الغازية لتوليد الطاقة الكهربائية وتشغيل الطائرات والعديد من التطبيقات الصناعية. صمم التوربين في محطة كهرباء قري لإنتاج الطاقة بقوة 42 ميجاوات ولكن لأسباب عديدة (كما حددتها المنظمة الدولية للتوحيد القياسي لذلك فإن ظروف مدخل الهواء هي: درجة حرارة الهواء 15 درجة مئوية، الرطوبة النسبية 60٪، الضغط المطلق 101.325 كيلو باسكال عند مستوى سطح البحر) انخفضتالطاقة الفعلية إلى 30 ميجاوات، وزاد التلوث البيئي وفقًا لنوع الوقود المستخدم.لذلك سنحاول إمكانية استخدام نظام حقن المياه لتجنب التلوث (أكاسيد النيتروجين،اكاسيد الكبريت، اكاسيد الكربون) وأيضًا تحسين الأداء الاجمالي والكفاءة. أهداف هذا البحث هي زيادة القدرة الخارجة والكفاءة الناتجة من التوربينات الغازية وتقليل التأثير البيئي. تقنيات تحسين قدرة التوربينات الغازية مثل حقن الماء ربما تضمن سلامة التشغيل للتوربينات الغازية لتقليل تكوين أكاسيد النيتروجين أيضًا لتقليل درجة حرارة اللهب عن طريق إدخالالماء او البخارفي منطقة اللهب. كل من المياه والبخار فعالة للغاية في تحقيق هذا الهدف. بشكل عام تقتصر كمية المياه على الكمية المطلوبة لتلبية متطلبات أكاسيد النيتروجين من أجل تقليل تكلفة التشغيل. أظهرت النتائج أن استخدام أنظمة حقن الماء زاد من ناتج الشغل الصافي والكفاءة ومعدل الحرارة. دون استخدام حقن الماء، كان إجمالي ناتج الشغل الصافي (30 ميجاوات) والكفاءة % 29.3ولكن عندما استخدمنا نفس معدل تدفق الوقود والماءأصبح إجمالي ناتج صافيالشغل 31.7 ميقا واط زيادة حوالي%5.67 ايضا الكفاءة اصبحت %31 زادت بحوالي %5.8.

عندما نستخدم معدل تدفق الماء يعادل نصف تدفق الوقود اصبح صافي انتاج الشغل30.9 ميقا واط زاد بحوالي%3 كما اصبحت الكفاءة %31.2 بزيادة حوالي%6.47، هذا يعني أن الشغل الكلي والكفاءة معدلات الحرارة تتأثر مباشرة بنسبة حقن الماء.

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List of Abbreviations

NO_x	Nitrogen oxides

GT	Gas turbine
GTIs	Gas turbine installations
CC	Combustion chamber
GTIWI	Gas turbine installation water injection
GTITWI	Gas turbine installation total water injection
GE	Generating electrical
HRSG	Heat recovery steam generator
FBN	Fuel bound nitrogen
GT1	Gas turbine number one
LHV	Lowering heating value
SFC	Specific fuel consumption

List of symbols

K	Kelvin

T	Temperature, (K)
T_1	Temperature at compressor inlet
T_2	Temperature at combustor inlet
T_3	Temperature at turbine inlet
T_4	Temperature at turbine outlet
r_p	Compressor ratio
P	Pressure, N/m^2
Q_{in}	Heat input rate, J/m^3
$ή_c$	Compressor efficiency
$ή_t$	Turbine efficiency
Cp_{air}	Specific heat of air at constant pressure/kg-K
Cp_f	Specific heat of fuel constant pressure, J/kg-K
T_a	Reference temperature
W_C	Compressor work per unit mass,W/kg
W_T	turbine work per unit mass,W/kg
W_{Net}	Net work of gas turbine, W
$\dot{m}_{air}$	Air mass flow rate,Kg/s
$\dot{m}_f$	fuel mass flow rate,Kg/s
$\dot{m}_{arc}$	Air rotor cooling mass flow rate,Kg/s
T_w	Temperature of injected water
LH_w	Latent heat of vaporization of water,J/m^3
T_f	Temperature of fuel
$ή_{th}$	Gas turbine thermal efficiency
γ_a	Specific heat of Air
γ_a	Specific heat of gas
P	Density,kg/m^3
W	Works,W

CHAPTER I

INTRODUCTION

CHAPTER I

INTRODUCTION

1.1 Introduction:

Gas turbines are used for power electric generation, operating airplanes and for several industrial applications, The gas turbine engine consist of a compressor to raise combustion air pressure, a combustion chamber where the fuel/air mixing is burned, and a turbine that through expansion extracts energy from the combustion gases, These cycles operates according to the open Brayton thermodynamic cycle and present low thermal efficiency and are referred as combustion turbines. Usually, the rated capacities of combustion turbines are based on standard ambient air, and zero inlet and exhaust pressure drops, as specified by the International Standardization Organization (ISO) Therefore, the air inlet conditions are: air temperature 15 °C, relative humidity 60%, absolute pressure 101.325 k Pa at sea level [2].

Due to environmental reasons resulting pollutant formation has been a part of interest in research. In a combustion with air however, Nitrogen oxides (NO_x) in the exhaust gas are not completely avoidable. Nitrogen oxides harm the environment by being responsible for the ozone depletion in the stratosphere, contribute to the greenhouse effect in the troposphere and are suspected of causing lung cancer [3].

On the one hand water injections influence the NO_x formation physically by lowering the flame temperature. On the other hand the combustion chemistry is affected by changes in radical concentrations. In order to predict NO_x emissions accurately and to investigate the combustion chemistry numerical simulation is useful for various applications [6].

1.2 Problem Statement:

The gas turbine designed for power output 42 MW but because of many reasons (type of fuel, ambient temperature…etc.) the actual power decreased to 30 MW, and the Environmental pollution increased according to the type of fuel used. So we will try the possibility of using a water injection system to avoid the pollutions (NO_x, NO_x, CO) and also improving the over hall performance and efficiency.

1.3 Research Objectives:

The objectives of this research are:

1. To Increase output power&efficiency produced from gas turbine
2. To Decrease environmental impact.

1.4 Scope:

This research concerns about studying Garri (1, 2) combined cycle Gas turbine.

1.5 Significance of this study:

Water injection one of the key elements for increasing the efficiency of the Gas turbine, to achieve the maximum energy in it and improving the over hall performance.

1.6 Thesis outline:

- General background, problem statement, research objectives, scope, and significance of the research (chapter one).
- Introduction about energy sources, power generation, conventional sources of energy, non-conventional sources of energy, thermal power plants, Garri (1,2) thermal power plant, fuel analysis, and current studies water injection systems in Gas turbine (chapter two).

- Calculations of power output, efficiency, fuel consumption and the maximum heat rate in the case of using water injection systems at different rates of water and fuel consumptions (chapter three).
- Results and discussions (chapter four).
- Conclusion and recommendation of the study (chapter five).

CHAPTERII

LITERATURE REVIEW

CHAPTER II

LITERATURE REVIEW

2.1 Preface:

Gas turbine power enhancement technologies, such as inlet fogging, inter stage water injection, saturation cooling, inlet chillers, and combustor injection, are being employed by end users without evaluating the potentially negative effects these devices may have on the operational integrity of the gas turbine [8].

2.2 Water and steam injections purposes:

2.2.1 Gas turbine installation with total water injection in the combustion chamber:

The authors present the results of their thermodynamic analysis, regarding gas turbine

Installations (GTIs) with total water injection in the combustion chamber (CC). Among existing GTIs, the majority are without water or steam injection in the working fluid, while some of them employ partial water/steam injection Gas turbine installation water injection (GTIWI). The technical solution proposed by the authors, with total water injection (GTITWI), is designed to realize the desired temperature in the CC by means of the injected water exclusively.

The introduced air flow is only the quantity strictly needed to produce combustion. As a result, for many cases, we have higher values for the thermal efficiency in a GTITWI compared with the GTIs without water/steam injection. Consequently, from this point of view, GTIWI are situated between GTI and GTITWI. Also, like for all GTIWI, for GTITWI too, using water injection results in much lower values for NO_xemissions [1].

2.2.2 Steam and Water Injection for Power Augmentation:

Injecting steam or water into the head end of the combustor for NO_x abatement increases mass flow and, therefore, output. Generally, the amount of water is limited to the amount required to meet the NO_x requirement in order to minimize operating cost and impact on inspection intervals.

Steam injection for power augmentation has been an available option onGenerating electricalcompany (GE) gas turbines for over 30 years. When steam is injected for power augmentation, it can be introduced into the compressor discharge casing of the gas turbine as well as the combustor. The effect on output and heat rate is the same as that shown in Figure14. GE gas turbines are designed to allow up to 5% of the compressor airflow for steam injection to the combustor and compressor discharge.

Steam must contain 50 F/28 C superheat and be at pressures comparable to fuel gas pressures. When either steam or water is used for power augmentation, the control system is normally designed to allow only the amount needed for NO_x abatement until the machine reaches base (full) load. At that point, additional steam or water can be admitted via the governor control. [2]

2.2.3 Total water injection application in the combustion chamber:

Fig. 2.1 represents a simplified scheme of a GTITWI.

In the main, the installation is composed of an air compressor—1, chamber or chambers of combustion—2, gas turbine—3, the directly or by a GB drawn machine—4, low pressure water Pump—5, chemical water treatment installation—6, high pressure water pump—7 and fuel feeding unit—8.

We define ‘‘the water addition coefficient’’ (injected in the CC) needed to obtain the desired.

Temperature in the CC as the quotient between the water quantity Gw injected in the CC and the fuel quantity Gf introduced in the CC.

The water injected is instantly transformed into steam because of the very high temperature (usually higher than 800 C), which is obtained in the conditions of stoichiometric combustion. In this way, the combustion gases evacuated from the CC are, in fact, a mixture of gases resulting from the fuel combustion and steam from the vaporization of the water injected in the CC.

Let us denote with this water addition coefficient$\delta = G_W/G_f$

To determine the value of $ we perform a thermal balance with reference to the CC (Fig2.2). Thus, if we refer to 1 kg of fuel, the thermal quantities introduced in the CC are the following

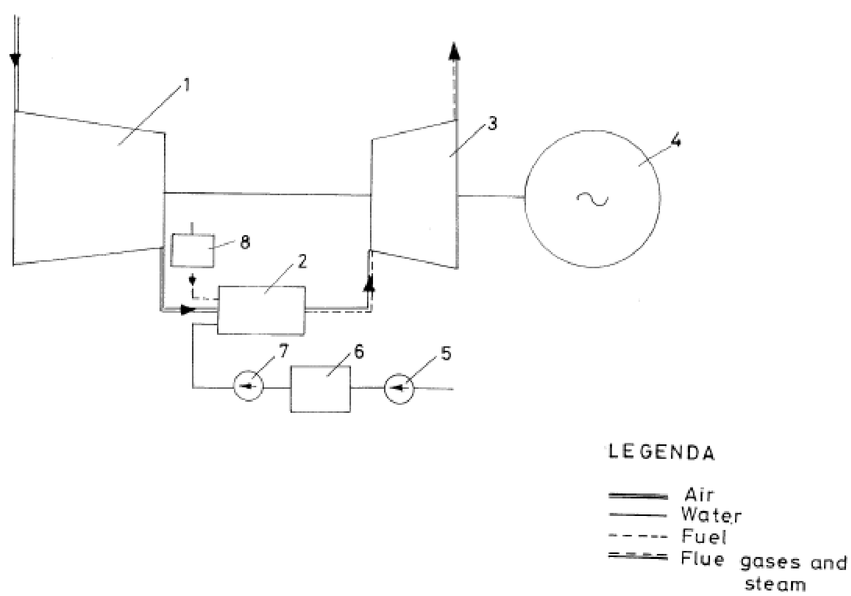


Figure 2.1 GTITWI—schematic representation. [1]

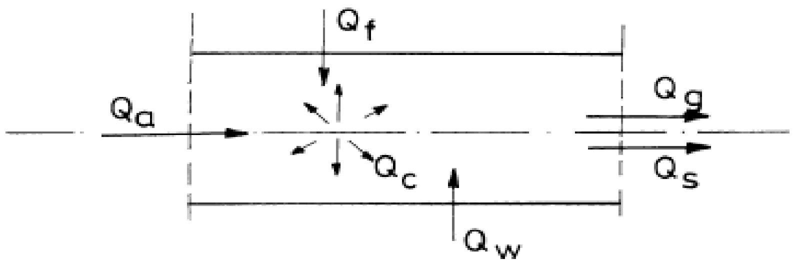


Figure 2.2 Heat flow in/out CC.**[1]**

2.2.4 Water- and Steam-Injected in Gas Turbines:

Water or steam can be injected into the gas turbine combustor to control NO_x emissions. However, although water injection reduces NO_x emissions, it exerts an adverse Impact on gas turbine efficiency. Steam provides both NO_x control and power output Enhancement in steam-injected gas turbine plants. The injected steam produces more power for a given turbine inlet temperature(TIT) value in all cases, but additional heat input is required to heat the steam to the turbine inlet temperature level. Therefore, an increase in efficiency is not always attained. The injection of steam or water can occur at various points along the gas path in the gas turbine. Owing to the application of cooling.

The firing temperature can be further increased, and thus the gas turbine efficiency can be enhanced.

Steam-injected reheat gas turbine power plant is shown schematically in Figure 2.3. Steam produced in the heat recovery steam generator (HRSG) is injected in the primary (main) combustor and in reheat combustor. An increased mass flow rate through the gas turbine resulting from steam injection enhances the power output of the gas turbine by 10–40% with practically the same air flow.

It is clear that water or steam injection reduces NO_x emissions because of the Lowered flame temperature. The effect of steam (water) injection on NO_x and CO.

Emissions as well as on power output and heat rate of the gas turbine plant is illustrated in Figure 2.4. The NO_x emissions reduction degree depends on the mass of water or steam injected per unit mass of fuel.

Water is a more efficient flame coolant

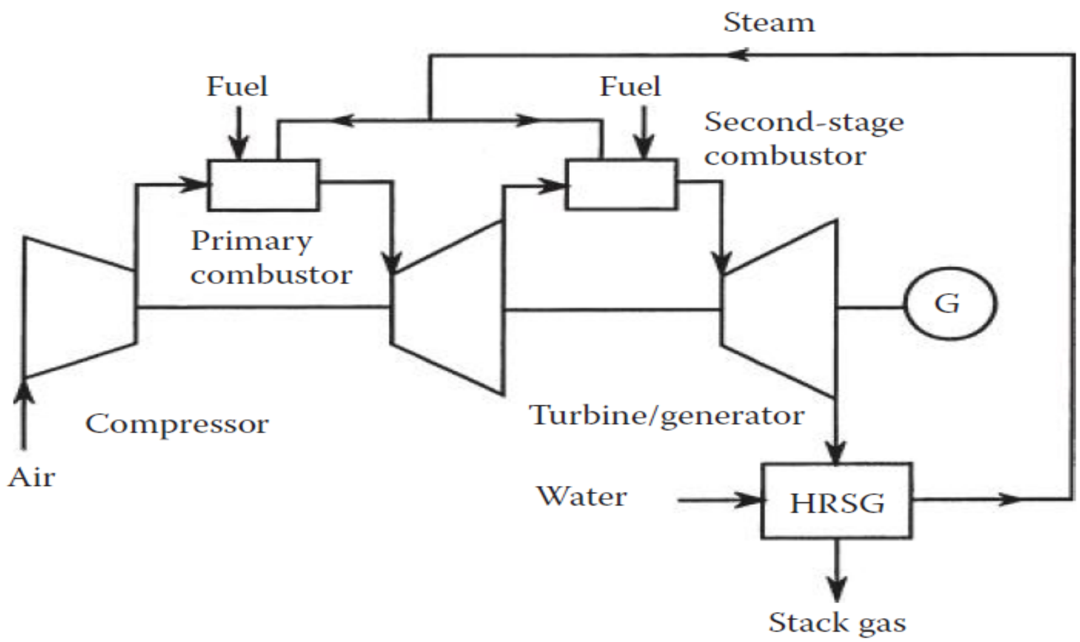


Figure 2.3Schematic diagram of a steam-injected reheat cycle gas turbine power plant. HRSG, heat recovery steam generator[3]

Than steam owing to evaporation inside the flame. However, this method of emissions reduction has the following drawbacks:

-it requires large quantities of treated water

And it lowers the gas turbine efficiency, especially when water is injected. [3]

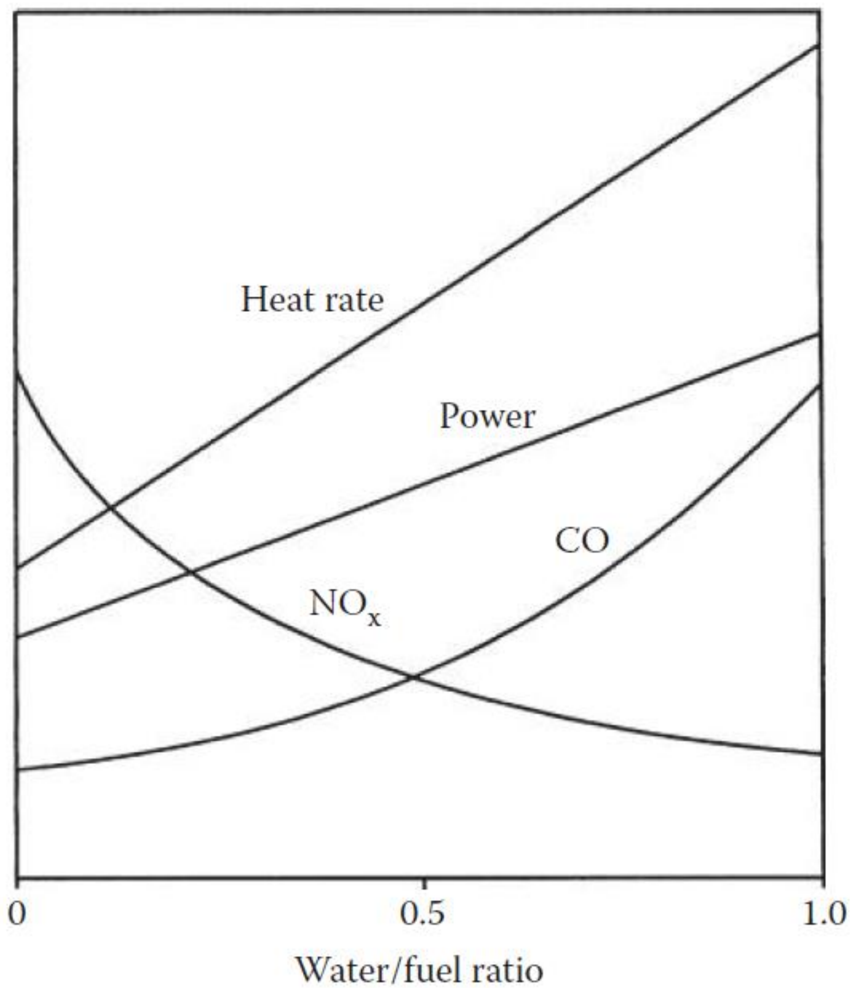


Figure 2.4Effect of water/fuel ratio on NO_x and CO emissions, power output, and heat rate of the gas turbine power plant[3].

2.3 Emissions Characteristics of Conventional Combustion Systems:

Typical exhaust emissions from a stationary gas turbine are listed in Table 1. There are two distinct categories. The major species (CO_2, N_2, H_2O, and O_2) are present in percent concentrations. The minor species (or pollutants) such as CO, UHC, NO_x, SO_x, and particulates are present in parts per million concentrations. In general, given the fuel composition and machine operating conditions, the major species compositions can be calculated. The minor species, with the exception of total sulfur oxides, cannot. Characterization of the pollutants requires careful measurement and semi theoretical analysis. The pollutants shown in Table 1 are a function of gas turbine operating conditions and fuel composition. In the following sections, each pollutant will be considered as a function of operating conditions under the broad divisions of gaseous and liquid fuels. [4]

Table (2.1): Gas turbine exhausts emissions burning conventional fuels

Major Species	**Typical Concentration (% Volume)**	**Source**
Nitrogen (N_2)	66 - 72	Inlet Air
Oxygen (O_2)	12 - 18	Inlet Air
Carbon Dioxide (CO_2)	1 - 5	Oxidation of Fuel Carbon
Water Vapor (H_2O)	1 - 5	Oxidation of Fuel Hydrogen
Minor Species Pollutants	**Typical Concentration (PPMV)**	**Source**
Nitric Oxide (NO)	20 - 220	Oxidation of Atmosphere Nitrogen
Nitrogen Dioxide (NO_2)	2 - 20	Oxidation of Fuel-Bound Organic Nitrogen
Carbon Monoxide (CO)	5 - 330	Incomplete Oxidation of Fuel Carbon
Sulfur Dioxide (SO_2)	Trace - 100	Oxidation of Fuel-Bound Organic Sulfur
Sulfur Trioxide (SO_3)	Trace - 4	Oxidation of Fuel-Bound Organic Sulfur
Unburned Hydrocarbons (UHC)	5 - 300	Incomplete Oxidation of Fuel or Intermediates
Particulate Matter Smoke	Trace - 25	Inlet Ingestion, Fuel Ash, Hot-Gas-Path
		Attrition, Incomplete Oxidation of Fuel or
		Intermediates

2.3.1 Nitrogen Oxide Formation and NO x Emissions:[4]

The major factors affecting NO_x production in the gas turbine combustor are:

- Firing temperature
- Oxygen availability in the combustion zone
- Duration of the combustion process

NO formed mainly from combustion air at very high temperatures are high, such Nitrogen oxide (NO) is formed mainly from nitrogen in the combustion air at very high temperatures, such as those found in the flame inside the gas turbine combustor.

The adiabatic flame temperature Tad depends on the excess air ratio λ, composition, and heating value of the fuel. This temperature is the highest in case of stoichiometric combustion with excess air ratio $\lambda$of 1. If $\lambda$is around 1.1, the flame temperature, Is high, but there is less oxygen available to form NO. Above this λlevel, the NO formation decreases as the excess air in the flame zone lowers its temperature [4].

2.3.2 Nitrogen Oxides:

Nitrogen oxides ($NO_x = NO + NO_2$) must be divided into two classes according to their mechanism of formation. Nitrogen oxides Formed from the oxidation of the free nitrogen in the combustion air or fuel are called "thermal NO_x." They are mainly a function of the stoichiometric adiabatic flame temperature of the fuel, which is the temperature reached by burning a theoretically correct mixture of fuel and air in an insulated vessel.

The following is the relationship between combustor operating conditions and thermal NO_x production:

_ NO_x increases strongly with fuel-to-air ratio or with firing temperature

_ NO_x increases exponentially with combustor inlet air temperature

_ NO_x increases with the square root of the combustor inlet pressure

_ NO_x increases with increasing residence time in the flame zone

_NO_x decreases exponentially with increasing water or steam injection or increasing specific humidity Emissions which are due to oxidation of organically.

Bound nitrogen in the fuel—fuel-bound nitrogen (FBN)—are called "organic NO_x."

Only a few parts per million of the available free nitrogen (almost all from air) are oxidized to form nitrogen oxide, but the oxidation of FBN to NO_x is very efficient. For conventional GE combustion systems, the efficiency of conversion of FBN into nitrogen oxide is 100% at low FBN contents. At higher levels of FBN, the conversion efficiency decreases. Organic NO_x formation is less well understood

Than thermal NO_x formation. It is important to note that the reduction of flame temperatures to abate thermal NO_x has little effect on organic NO_x. For liquid fuels, water and steam injection actually increases organic NO_x yields. Organic NO_x formation is also affected by turbine firing temperature. The contribution of organic NO_x is important only for fuels that contain significant amounts of FBN such as crude o;2r residual oils. Emissions from these fuels are handled on a case-by-case basis. Gaseous fuels are generally classified according.

To their volumetric heating value. This value is useful in computing flow rates needed for a given heat input, as well as sizing fuel nozzles, combustion chambers, and the like. However, the stoichiometric adiabatic flame temperatureis a more important parameter for characterizing.

NO_x emission. *Table 2* shows relative thermal NO_x production for the same combustor burning different types of fuel. This table shows the NO_x relative to the methane NO_x based on adiabatic stoichiometric flame temperature.

The gas turbine is controlled to approximate constant firing temperature and the products of combustion for different fuels affect the reported NO_x correction factors. Therefore, *Table 2* also shows columns for relative NO_x values calculated.

For different fuels for the same combustor and constant firing temperaturerelative to the NO_x for methane.

Burning natural gas fuel and No. 2 distillate is shown in Figures (1.4)respectively as a function of firing temperature. The levels of emissions for No. 2 distillate oil are a very nearly constant fraction of those for natural gas over the operating.

Range of turbine inlet temperatures. For any given model of GE heavy-duty gas turbine, NO_x correlates very well with firing temperature. Low-Btu gases generally have flame temperatures below 3500°F/1927°C and correspondingly lower thermal NO_x production. However, depending upon the fuel-gas clean-up train, these gases may contain significant quantities of ammonia. This ammonia acts as FBN and will be oxidized to NO_x in a conventional diffusion Combustion system. NO_x control measures such as water injection or steam injection will have little or no effect on these organic NO_x Emissions [4].

Table (2.2): Relative thermal NO_x emissions

Fuel	Stoichiometric Flame Temp.	NO_x (ppmvd/ppmvw-Methane) 1765°F/963°C – 2020°F/1104°C Firing Time	NO_x (ppmvd/ppmvw-Methane) @ 15% O_2, 1765°F/963°C – 2020°F/1104°C Firing Time
Methane	1.000	1.000/1.000	1.000/1.000
Propane	1.300	1.555/1.606	1.569/1.632
Butane	1.280	1.608/1.661	1.621/1.686
Hydrogen	2.067	3.966/4.029	5.237/5.299
Carbon Monoxide	2.067	3.835/3.928	4.128/0.529
Methanol	0.417-0.617	0.489/0.501	0.516/0.529
No. 2 Oil	1.667	1.567/1.647	1.524/1.614

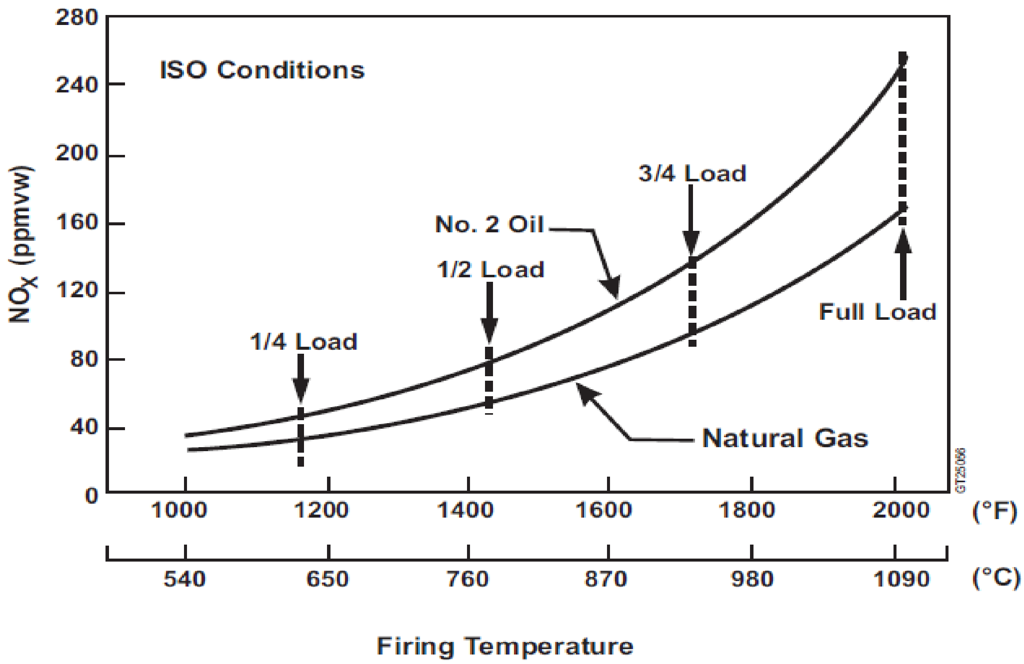


Figure 2.5MS7001EA NO_x emissions [4]

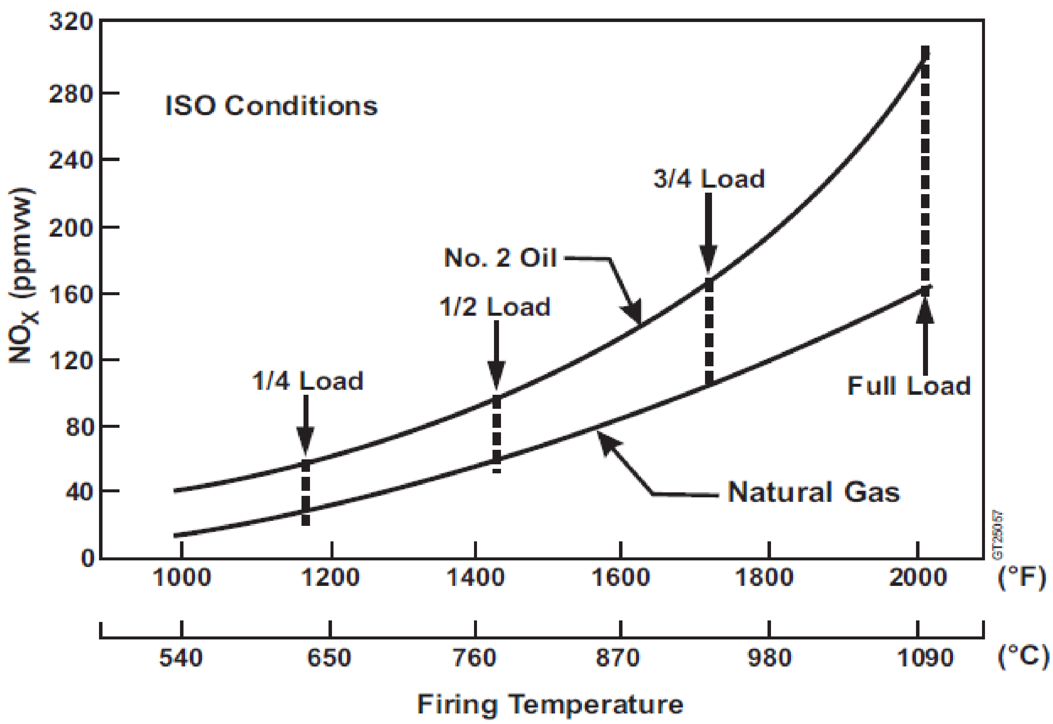


Figure 2.6MS6001B NO_x emissions [4]

2.3.3 Ambient Pressure effect:

NO_x ppm emissions vary almost directly with ambient pressure. Figure (2.7) provides an approximation for the ambient pressure effect on NO_x production on a lb/hr.

Basis and on a ppmvd @ 15% O2 basis. This figure is at constant 60% relative humidity. It should be noted that specific humidity varies with ambient pressure and that this variation is also included in the Figure 18curves.

2.3.4 Ambient Temperature effect:

Typical NO_x emissions variation with ambient temperature is shown in Figure(2.8) this figure is drawn at constant ambient pressure and 60% relative humidity with the gas turbine operating constant gas turbine firing Temperature. For an operating gas turbine the actual NO_x characteristic is directly influenced by the control system exhaust temperature control curve,which can change the slope ofthe curves. The typical exhaust temperature control curve used by GE is designed to hold constant turbine firing temperature in the 59°F/15°C to 90°F/32°C ambient temperature range. The firing temperature with this typical curve causes under-firing of approximately 20°F/11°C at 0°F/–18°C ambient, and approximately 10°F/6°C under-firing at 120°F/49°C ambient. Factors such as load limits, shaft output limits, and exhaust system temperature limits are also not included in the Figure (2.8)curves. Based on the actual turbine exhaust temperature control curve used and other potential limitations that reduce firing temperature, the estimated NO_x emissions for an operating gas turbine are typically less than the values shown in Figure 18at both high and low ambient?

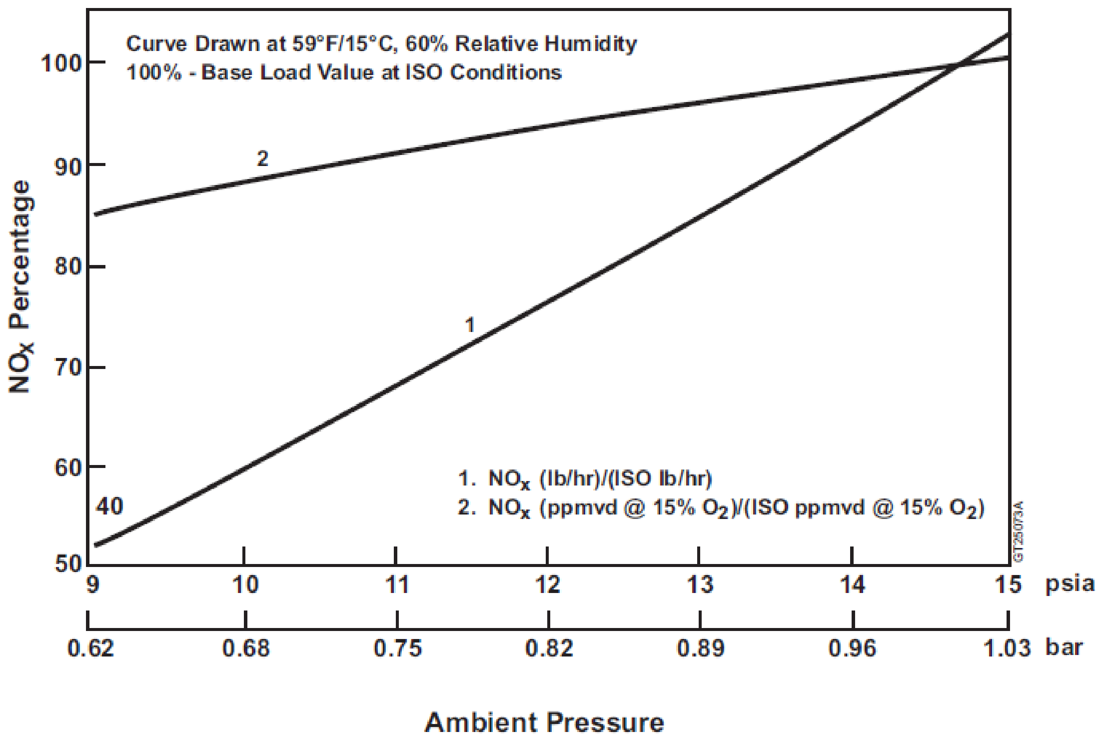


Figure 2.7 Ambient pressure effects on NO_xFrames 5, 6 and 7 [4]

2.3.5 Relative Humidity effect:

This parameter has a very strong impact on NO_x. The ambient relative humidity effect on NO_x production at constant ambient pressure of 14.7 Pisa and ambient temperatures of 59°F/15°C and 90°F/32°C is shown in Figure (2.9).

The impact of other parameters such as inlet/exhaust pressure drops, regenerator characteristics, evaporative/inlet coolers, etc., are similar to the ambient parameter effects described above [4].

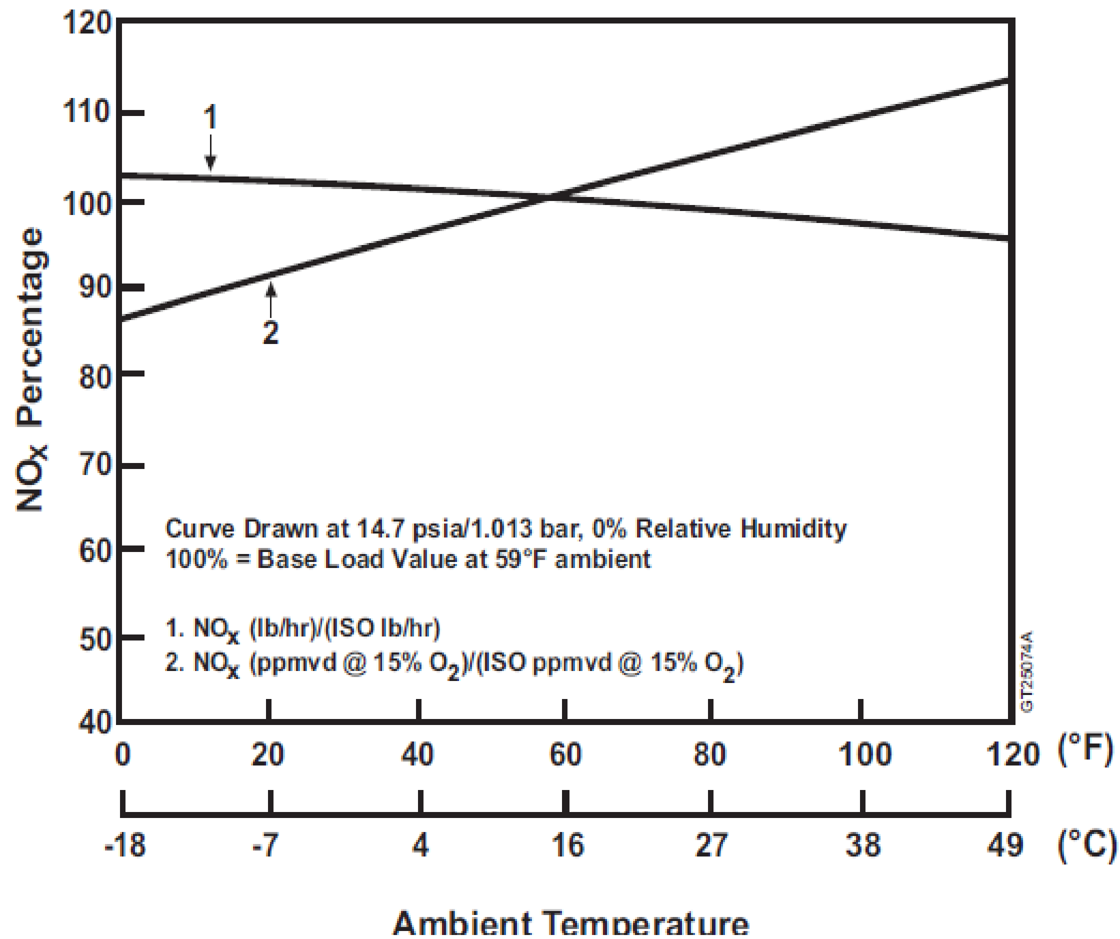


Figure 2.8 Ambient temperature effects on NO_x Frames 5, 6 and 70% Relative Humidity [4]

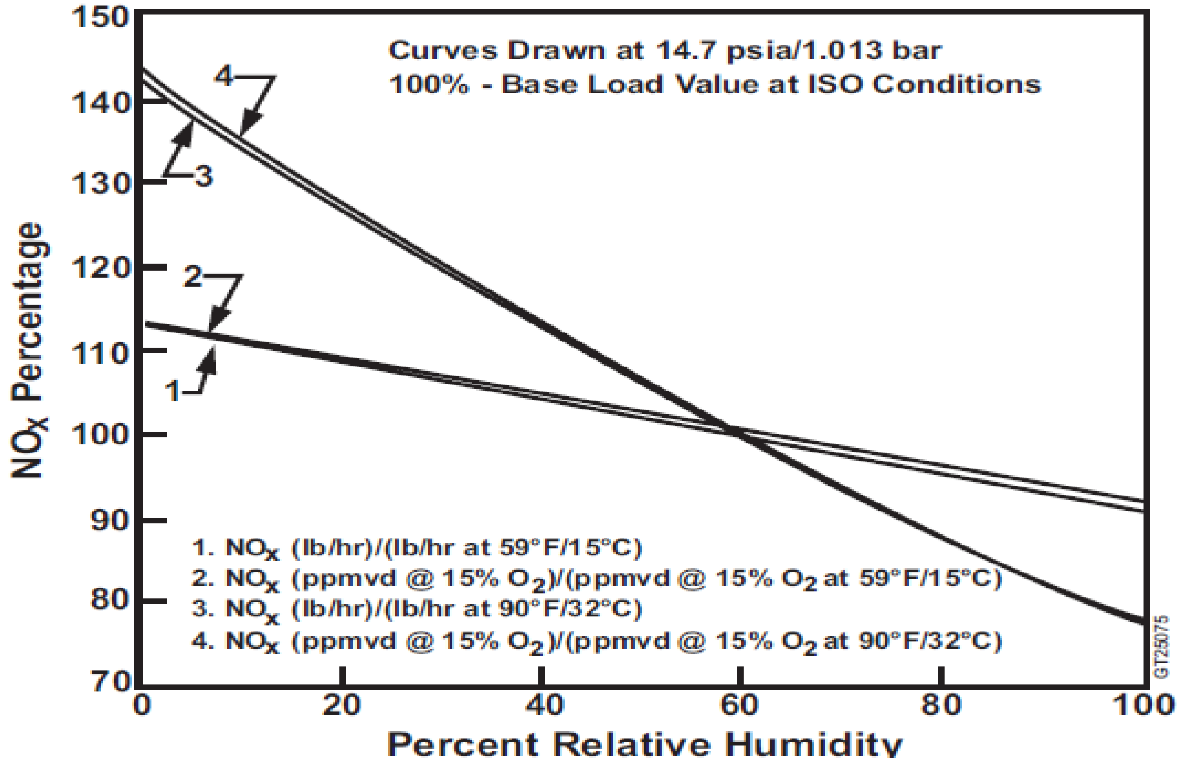


Figure 2.9 Relative humidity effect on NO_x Frames 5, 6 and 7 [4]

2.4 Emission Reduction Techniques:

The gas turbine, generally, is a low emitter of exhaust pollutants because the fuel is burned with ample excess air to ensure complete combustion at all but the minimum load conditions or during start-up. The exhaust emissions of concern and the emission control techniques can be divided into several categories as shown in Table (2.3). Each pollutant emission reduction technique will be discussed in the following sections. [4]

2.5 Nitrogen Oxides Abatement:

The mechanism on thermal NO_x production was first postulated by Zeldovich. This is shown in Figure (2.10) it shows the flame temperature of distillate as a function of equivalence ratio. This ratio is a measure of fuel-to-air ratio in the combustor normalized by stoichiometric fuel-to-air ratio. At the equivalence ratio of unity, the stoichiometric conditions are reached. The flame temperature is highest at this point. At equivalence ratios less than 1, wehave a "lean" combustor. At the values greater than 1, the combustor is "rich." All gas turbine.

Combustors are designed to operate in the lean region. Figure 2.10shows that thermal NO_x production raises very rapidly as the stoichiometric flame temperature are reached. Away from this point, thermal NO_x production decreases rapidly. This theory then provides the mechanism of thermal NO_x control. In diffusion flames combustor, the primary way to control thermal NO_x is to reduce the flame temperature. [4]

Table (2.3) Emission control techniques [4]

NO_x	Lean Head End Liner Water or Steam Injection Dry Low NO_x
CO	Combustor Design Catalytic Reduction
UHC & VOC	Combustor Design
SO_x	Control Sulfur in Fuel
Particulates & PM-10	Fuel Composition
Smoke Reduction	Combustor Design - Fuel Composition - Air Atomization
Particulate Reduction	Fuel Composition - Sulfur - Ash

GT25092

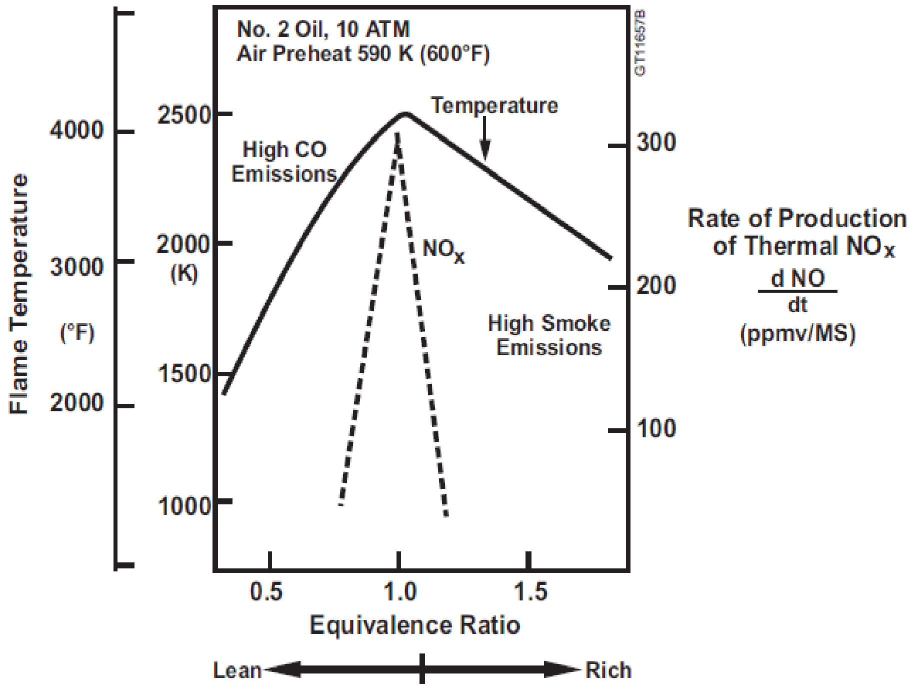


Figure 2.10NO_x production rates [4]

2.6 Water and Steam Injection:

Another approach to reducing NO_x formation is to reduce the flame temperature by introducing a heat sink into the flame zone. Both water and steams are very effective at achieving this goal. A penalty in overall efficiency must be paid for the additional fuel required to heat the water to combustor temperature. However, gas turbine output is enhanced because of the additional mass flow through the turbine. By necessity, the water must be of boiler feed water quality to prevent deposits and corrosion in the hot turbine gas path area downstream of the combustor. Water injection is an extremely effective means.

For reducing NO_x formation; however, the combustor designer must observe certain cautions when using this reduction technique. To maximize

The effectiveness of the water used, fuel nozzles have been designed with additional passages to inject water into the combustor head end. The water is thus effectively mixed with the Incoming combustion air and reaches theflame zone atits hottest point. In Figure 2.11the NO_x reduction achieved by water injection is plotted as a function of water-to-fuel ratio for an MS7001E machine. Other machines have similar NO_x abatement performance with water injection.

Steam injection for NO_x reduction follows essentially the same path into the combustor Head end as water. However, steam is not as effective as water in reducing thermal NO_x. The high latent heat of water acts as a strong thermal Sink in reducing the flame temperature. In general, for a given NO_x reduction, approximately 1.6 times as much steam as water on a mass basis is required for control. There are practical limits to the amount of Water or steam that can be injected into the combustor before serious problems occur. This has been experimentally determined and must be taken into account in all applications if theCombustor designer is to ensure long hardware life for the gas turbine user. [4]

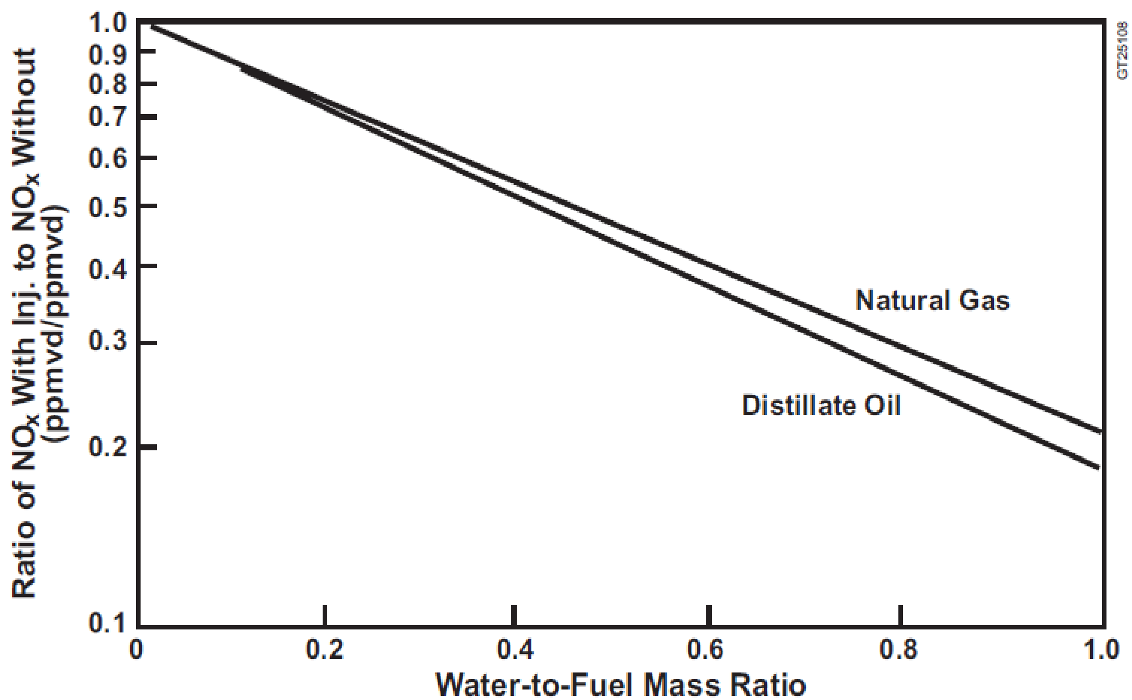


Figure 2.11 MS7001E NO_x reductions with water injection [4]

2.6.1 Water/Steam Injection Hardware:

The injection of water or steam into the combustion cover/fuel nozzle area has been the primary method of NO_x reduction and control in GE heavy-duty gas turbines since the early 1970s. The same design gas turbine equipment is supplied for conversion retrofits to existing gas turbines for either injection method. BothNO_x control injection methods require a microprocessor controller; therefore turbines with older controls need to have their control system upgraded to Mark V or Mark VI SPEEDTRONIC™ controls conversion. The control system for both NO_x control injection methods utilizes the standard GE gas turbine control philosophy of two separate independent methods for shutting off the injection flow. The NO_x water injection system is consists of a water pump and filter, water flow meters, water stop and flow control valves. This material is supplied on a skid approximately 10 x 20 feet in size for mounting at the turbine site. The water from the skid is piped to the turbine base where it is manifold to each of the fuel nozzles using pigtails. The water injection at the combustion chamber is through passages in the fuel nozzle assembly. A typical water injection fuel nozzle assembly is shown schematically in Figure (2.12) for this nozzle design there are eight or twelve [4]

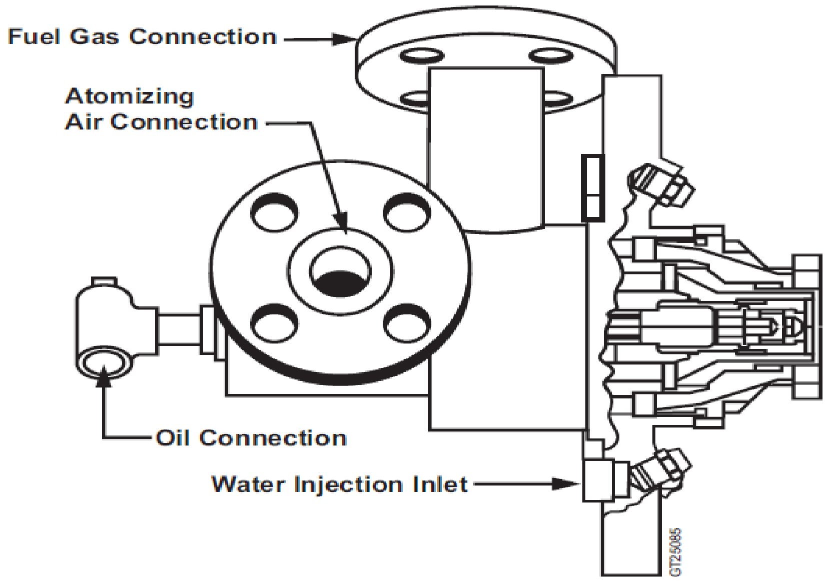


Figure 2.12Water injection fuel nozzle assemblies [4]

Water spray nozzles directing the water injection spray towards the fuel nozzle tip swirler. While this design is quite effective in controlling theNO_x emissions, the water spray has a tendency to impinge on the nozzle tip swirler and on the liner cap/cowl assembly. Resulting thermal strain usually leads to cracks, which limits the combustion inspections to 8000 hours or less.

To eliminate this cracking, the latest design water-injected fuel nozzle is the breech-load fuel nozzle. See Figure (2.13) in this design the water is injected through a central fuel nozzle Passage, injecting the water flow directly into the combustor flame. Since the water injection spray does not impinge on the fuel nozzle swirler or the combustion cowl assembly, the breech load fuel nozzle design results in lower maintenance and longer combustion inspection intervals for NO_x water injection applications**.**

The steam injection flow goes to the steam-injection manifold on the turbine base. Flexible pigtails are used to connect from the steam manifold to each combustion chamber. The steam injection into the combustion chamber is through machined passages in the combustion can cover. A typical steam-injection combustion cover with the machined steam-injection passage and steam injection nozzles is shown in Figure (2.14).

Water quality is of concern when injecting water or steam into the gas turbine due to potential problems with hot gas path corrosion, and effects to the injection control equipment. The injected water or steam must be clean and free of impurities and solids. The general requirements of the injected water or steam quality are shown in Table (2.4) Total impurities into the gas turbines are a total of the ambient air, fuel, and injected water or steam the total impurities. [4]

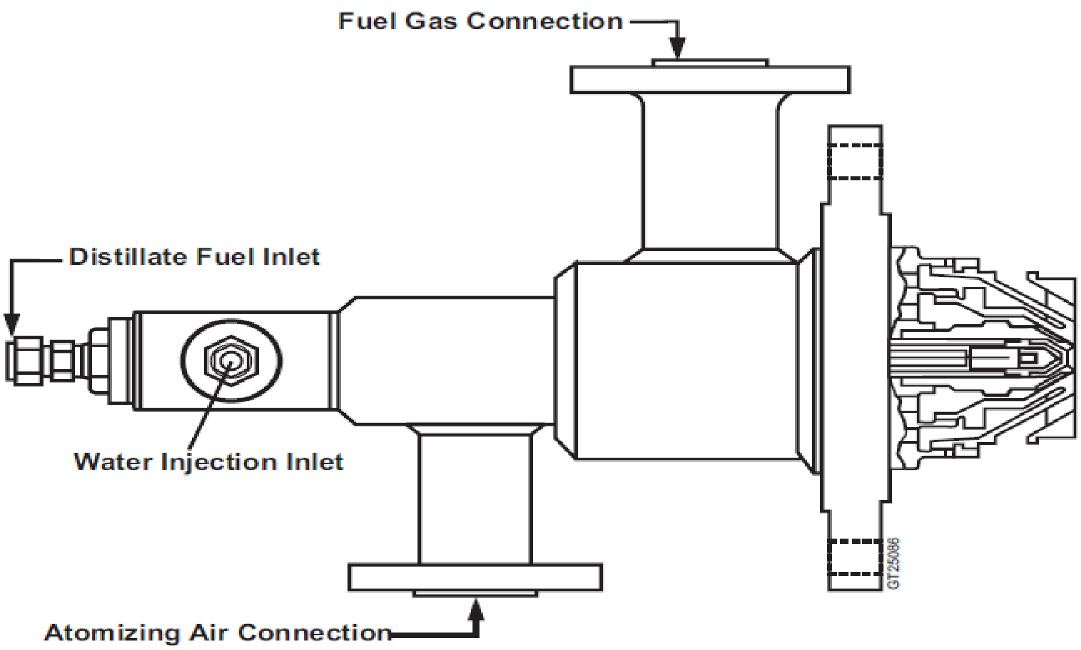
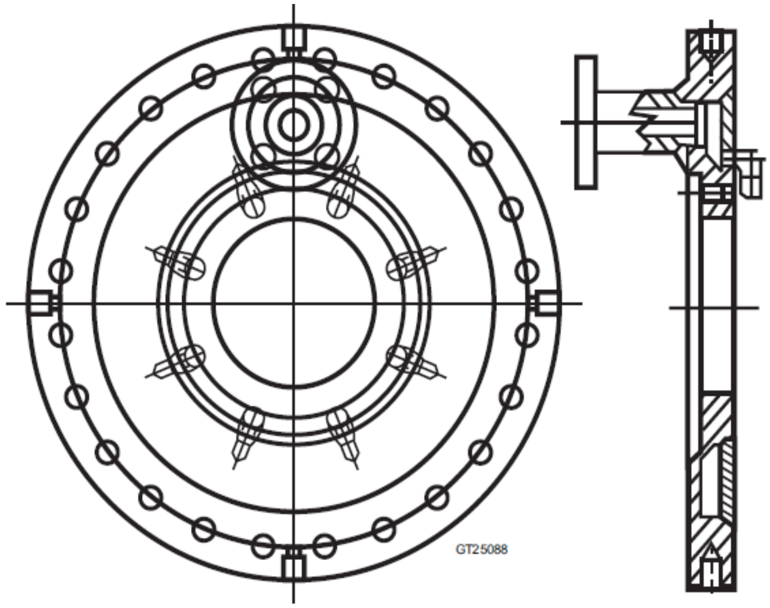


Figure 2.13 Breech-load fuel nozzle assemblies [4]



NOTE: This drawing is not to be used for Guarantees

Figure 2.14Combustion cover – steam injection [4]

Table (2.4)Water or steam injection quality requirements [4]

• **WATER/STEAM QUALITY** Total Dissolved Solids Total Trace Metals (Sodium + Potassium + Vanadium + Lead) pH NOTE: Quality requirements can generally be satisfied by demineralized water.	 5.0 ppm Max. 0.5 ppm Max. 6.5 – 7.5
• **TOTAL LIMITS IN ALL SOURCES** (Fuel, Steam, Water, Air) **Contaminant** Sodium + Potassium Lead Vanadium Calcium	 **Max. Equivalent Concentration (ppm – wt)** 1.0 1.0 0.5 2.0

2.7 Combustion Process:

In its simplest form, combustion is a process in which some material or fuel is burned. Whether it is striking a match or firing a jet engine, the principles involved are the same and the products of combustion are similar.

Combustion of natural gas is a chemical reaction that occurs between carbon, orHydrogen and oxygen. Heat is given off as the reaction takes place. The products ofCombustion are carbon dioxide and water. The reaction is as follows:

$$\begin{matrix} CH_4 + 4O \rightarrow CO_2 + 2H_2O + Heat \\ (Methane + Oxygen)(Carbondioxide + Water + Heat) \end{matrix} \rightarrow (1)$$

Four parts of oxygen are required to burn one part of methane. The products of combustion are one part of carbon dioxide and two parts of water. One cubic foot of methane will produce one cubic foot of carbon dioxide gas.

Oxygen used for combustion occurs in the atmosphere. The chemical composition of air is approximately 21% oxygen and 79% nitrogen, or one part of oxygen to four parts of nitrogen. In other words, for each cubic foot of oxygen contained in the air, there are about 4 ft3 of nitrogen.

Oxygen and nitrogen molecules each contain two atoms of oxygen or nitrogen.

Noting that one part, or molecule, of methane requires four parts of oxygen for complete combustion, and since the oxygen molecule contains two atoms, or two parts, the volumetric ratio of methane and oxygen is as follows:

$$CH_4 + 2(O_2 + 4N_2) \rightarrow 2H_2O + CO_2 + 8N_2 + Heat \qquad (2)$$

The preceding equation is the true chemical equation for the combustion process.

One cubic foot of methane actually requires 2 ft3 of oxygen for combustion. Since the oxygen is contained in air, which also has nitrogen, the combustion reaction can be written as follows:

$$1CH_4 + 2(O_2 + 4N_2) \rightarrow 1CO_2 + 8N_2 + 2H_2O + Heat$$
$$(Methane + Air)(Carbondixide + Nitrogen + Water + Heat) \quad (3)$$

One cubic foot (0.03m3/ of methane requires 10 ft3 (0.28 m3/ of air, 2 ft3 (0.06m3)f oxygen, and 8 cubic feet (0.23 cubic meter) of nitrogen for combustion. The products are carbon dioxide, nitrogen, and water. The combustion product of 1 ft3 of methane yields a total of 9 ft3 of carbon dioxide gas. In addition, the gas burned contains some ethane, propane, and other hydrocarbons. The yield of inert combustion gas from burning a cubic foot of methane will be 9.33 ft3 (0.26 m3/. If the combustion process created only the reactions shown in the previous discussion, no provision would be necessary for control. Unfortunately, other reactions occur in which undesirable products are formed.

The chemical reaction that occurs in the formation of nitric acid during the combustion process is as follows:

$$2N + 5O + H_2O \rightarrow 2NO + 3O + H_2O \rightarrow 2HNO_3$$
$$(Nitirc\ Oxide)(Nitirc\ Oxide) \quad (4)$$

The water required in the previous reaction comes from the water of combustion.

The intermediate reaction shown previously (nitric acid formation) does not occur. During the combustion process, but occurs after the nitric oxide is further oxidized to nitrogen dioxide (NO_2/ and cooled. Consequently, it is necessary to control the formation of nitric oxide during the combustion process to prevent its ultimate conversion to nitric acid. The formation of nitric oxide during combustion can be retarded by reducing the temperature at which combustion occurs. Normal combustion temperatures range from 3400 to 3500 _F (1871–1927 _C). At this temperature, the volume of nitric oxide in the combustion gas is about 0.01%. If the combustion temperature is lowered, the amount of nitric oxide is substantially reduced. By maintaining a temperature below 2800 _F (1538 _C) at the burner, the nitric oxide volume will be below.

The maximum limit of 20 parts per million (0.002%) This minimum isattained by injecting a noncombustible gas (flue gas) around the burner to cool the combustionzone.

Sulfuric acid is another common by-product of combustion. Its reaction is as follows:

$$\underset{}{H_2S + 4O} \rightarrow \underset{Sulfur\ Oxide}{SO_2 + H_2O} \rightarrow \underset{Sulfur\ acid}{H_2SO_4} \qquad (5)$$

The formation of sulfuric acid cannot be economically retarded in the combustion process. The best method of eliminating sulfuric acid as a combustion product is to remove sulfur from the incoming fuel gas. Two separate sweetening processes are used to remove all sulfur from the fuel gas that will be burned.

The amount of oxygen in the combustion gas is regulated by controlling the ratio of air to fuel in the primary section. As previously mentioned, the ideal volumetric ratio of air to methane is 10:1. If less than 10 volumes of air are used with one volume of methane, the combustion gas will contain carbon monoxide [5]. The reaction is as follows:

$$1CH_4 + 1.5(O_2 + 4N_2) \rightarrow 2H_2O + 1CO_2 + 6N_2 + Heat \qquad (6)$$

The basic equation used in compressor ratio calculations is :

$$r_p = P_2/P_1 \qquad (7)$$

The compressor efficiency is calculated using the following equation:

$$T_2/T_1 = 1 + \left(r_P\text{^}((\gamma - 1)/\gamma)\right)/\dot{\eta}_c \quad (8)$$

The Specific heat of air at constant pressure can be determined by the following equation:

$$Cp_{air} = 1.0189 * 10^3 - 0.13784T_a + 1.9843 * 10 \quad (-4) * T_a^2 + 4.2399 * T_a^3 * 10\text{^}(-7) - 3.7632 * 10\text{^}(-10) * T_a\text{^}4 \quad (9)$$

The Reference temperature is calculated using the following equation:

$$T_a = (T_1 + T_2)/2 \quad (10)$$

The compressor work per unit masscan be determined by the following equation:

$$Wc = C_{p,a} * T_1 * \left(\left(r_p\text{^}(\gamma_a - 1)/\gamma_a\right)\right)/\dot{\eta}_c \quad (11)$$

The turbine efficiency is calculated using the following equation :

$$T_4 = T_3\left[1 - \dot{\eta}_t\left(1 - r_p\text{^}\left((1 - \gamma_g)/\gamma\right)\right)\right] \quad (12)$$

The turbine work per unit mass can be determined by the following equation:

$$W_T = C_{p_{air}} * T_3 * \dot{\eta}_t\left(1 - r_p\text{^}\left(\left(1 - \gamma_g\right)/\gamma_g\right)\right) \quad (13)$$

The basic equation used in net work of gas turbine is :

$$\mathrm{W_{Net}} = \left(\dot{m}_{air} - \dot{m}_{arc} + \dot{m}_f + \dot{m}_w\right) * W_T - \dot{m}_{air}W_c \quad (14)$$

The basic equation used in net work with out water injection in gas turbine is :

$$\mathrm{W_{Net}} = \left(\dot{m}_{air} - \dot{m}_{arc} + \dot{m}_f\right) * W_T - \dot{m}_{air}W_c \quad (15)$$

The equation of combustion chamber balance is:

$$\dot{m}_{air} * C_{p_{air}} T_2 + \dot{m}_f * LHV \dot{m}_f * C_{p_f} + \dot{m}_W \left(C_{p_W} * T_W LH_W \right) = \left(\dot{m}_{air} - \dot{m}_{arc} + \dot{m}_f + \dot{m}_w \right) * C_{p_g} * T_3 + \dot{m}_{arc} * C_{p_{arc}} * T_{arc} (16)$$

The specific fuel consumption can be determined by the following equation:

$$S.F.C = \dot{m}_f / W_{net} \ (17)$$

The equation of heat input rateis:

$$Q_{in} = \dot{m}_f * LHV \quad (18)$$

2.8Gas Turbine Combustors:

The volume of nitric oxide in the combustion gas is about 0.01%. If the combustion temperature is lowered, the amount of nitric oxide is substantially reduced.

The use of natural gas and the use of the new dry low NO_x combustors have reduced NO_x levels below 10 ppm. Since September 1979, when regulations required that NO_x emissions be limited to 75 ppmvd (parts per million by volume, dry), thousands of heavy- and medium-duty gas turbines have accumulated millions of operating hours

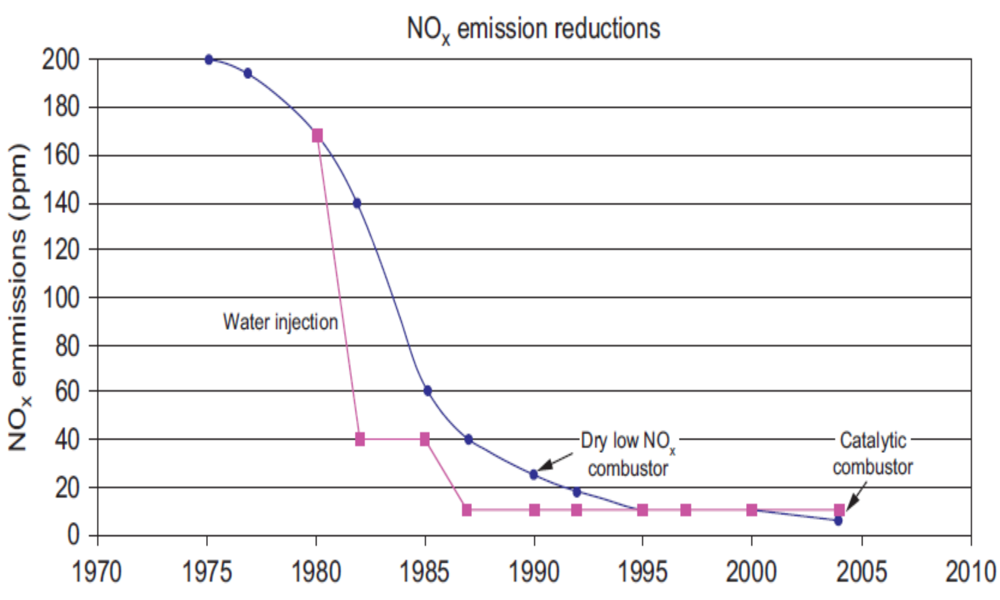


Figure 2.15 Control of gas turbine NO_x emissions over the years.

Using either steam or water-injection to meet required NO_x emissions levels, sometimes producing levels even lower than required. Figure **2-15** shows the reduction of NO_x in the past 30 years. The amount of water required to accomplish this is approximately 0.5–0.75% of the fuel flow. However, there is a 1.8% heat rate penalty associated with using water to control NO_x emissions for oil-fired simple-cycle gas turbines. Output increases by approximately 3%, making water (or steam) injection for power augmentation economically attractive in some circumstances (such as peaking applications). Single-nozzle combustors that use water or steam injection are limited in their ability to reduce NO_x levels below 42 ppmvd on gas fuel and 65 ppmvd on oil fuel. Since October 1987, multi-nozzle combustors using separate fuel and water/steam injection nozzles (see Figure **2-16**) have achieved 25 ppmvd NO_x (at 15% O2/ on gas fuel and 42 ppmvd on oil. The use of steam injection (wet combustors) in the diffusion combustors, and then in the1990s, the dry low emission/ NO_x combustors have greatly reduced the NO_x output. Most new turbines have progressed to dry low emission or dry low NO_x (DLE/DLN) combustors from the wet diffusion combustors, which were injected by steam in the primary zone of the combustor. The diffusion combustors have a single nozzle while most DLE combustors have multiple fuel nozzles for each can. New units under development have goals, which would reduce NO_x levels below nine ppm, and values of as low as two ppm can be attainable in the future [7]

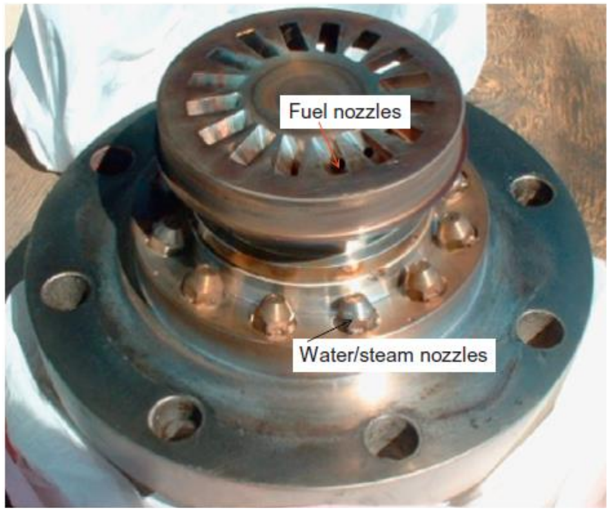


Figure 2.16A typical fuel nozzle showing both water and fuel nozzles for a typical wet diffusion combustor used for NO_x reduction (NO_x _ 25 ppm).

2.9 Current Studies:

In 2002, K. Mathioudakis analyze the effect of water injection in the combustion chamber of an industrial gas turbine by analytic relations and estimated the deviation of performance parameters because of injection of water into the exit of the compressor for reduction of NO_x emissions. It has been shown that the main parameter determining the magnitude of deviation is the water to fuel ratio, which also happens to be the quantity determining the amount of reduction of nitric oxide emissions.

Data from testing of a single-shaft gas turbine, employed for electricity generation, have been used to verify that the proposed relations can give a very good tool for estimating performance parameter deviations [9].

In April 2011, Lai-Cheong Chan and Yan Sudeveloped A practical thermodynamic model for the thermal efficiency of a simple cycle water injection gas turbine (WIGT) with rotor air cooling (RAC) and validated by comparing to the real data from the power plant in Macau. The results obtained from the present model has higher accuracy compared to previous models due to the application of dynamic values of the parameters such as injected water/steam ratio and rotor cooling air temperature. The optimal value for the ratio of injected water/steam is obtained based on the present thermodynamic model and validated by the measured electrical efficiency. The present model is helpful in the operation of the WIGT power plant [10].

CHAPTER III

METHODOLOGY

CHAPTER III

METHODOLOGY

3.1 Preface:

The data has been collected from Garri (1, 2) combined cyclepower station and directly from the gas turbine (GT) monitoring and control screen (GT1 case study).

3.2 Information of Gas Turbine No (1) In GARRI Power Plant:

The studied WIGT system is medicated GE Company with Model: PG6581B. The gas turbine nominally rated at 42MW, and is a self-contained, electric power generating system. The combustion turbine packaged plant is used in either simple cycle or heat recovery applications. The gas turbine compressor is axial flow type, and with 17stages. Under normal operation conditions, the rotor speed is about 5,163rpm and the compression ratio is 12.2.1 compressor inlet guide vane (IGV) is variable angle type, and they are driven by pneumatic control. A schematic diagram for simple gas turbine is shown in Fig. 1. This turbine system is designed to run 24 hours per day and 7 days per week. The temperatures and pressure for the inlet and outlet of the compressor and the turbine are measured. Also shown in Fig. 1, the mass flow rate of the injected fuel gas, injected water are measured and the bypass rotor cooling air. The data are measured at an interval of 1 min, and the hourly averaged values for the following parameters are recorded automatically by the software installed.

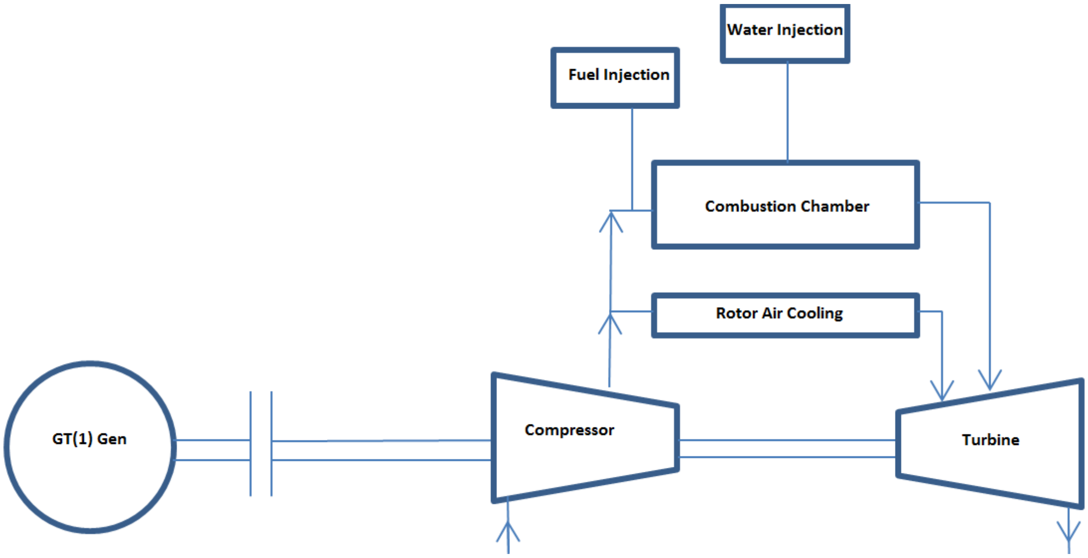


Figure 3.1the schematic diagram for a gas turbine in the power plant

3.3 Steam and Water Injection for Power Augmentation:

Injecting steam or water into the head end of the combustor for NO_x abatement increases mass flow and, therefore, output. Generally, the amount of water is limited to the amount required to meet the NO_x requirement in order to minimize operating cost and impact on inspection intervals.

Water injection for power augmentation has been an available option on GE gas turbines for Over 30 years. When water is injected for power augmentation, it can be introduced into the compressor discharge casing of the gas turbine as well as the combustor. The effect on output and heat rate is the same as that shown in Figure 3.2.

When either steam or water is used for power augmentation, the control system is normally designed to allow only the amount needed for NO_x abatement until the machine reaches base (Full) load. At that point, additional steam or water can be admitted via the governor control.[2]

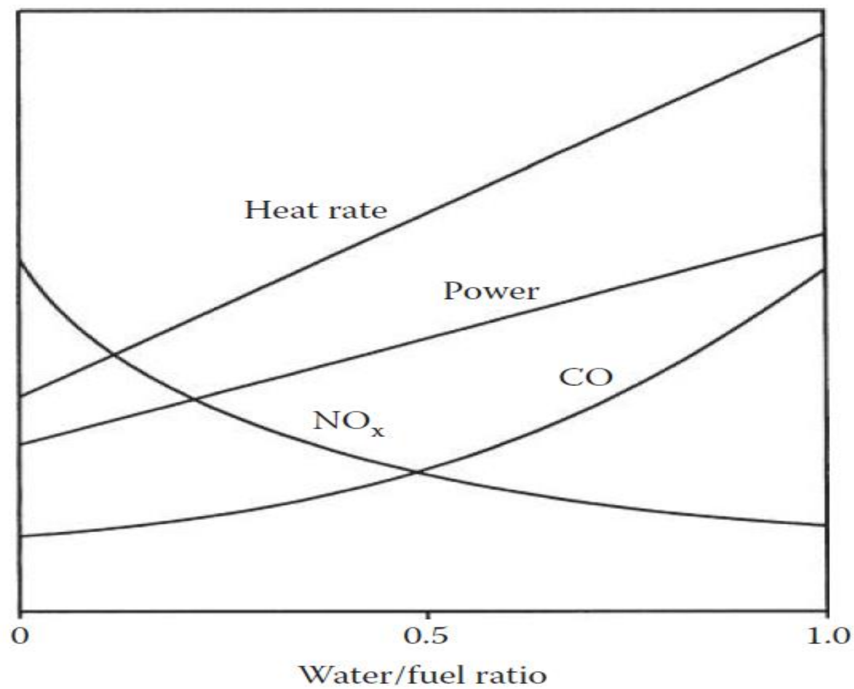


Figure 3.2Effect of water/fuel ratio on NO_x and CO emissions, power output, and heat rate of the gas turbine power plant [3]

many methods does not give the exact solution of the giving problem so we go to approximation with another problem have a nearly solution[21].

3.4 Technical parameters Of GT1:

3.4.1 Technical parameter GT1 (Design data):

Technical parameters Of GT1	
Model	PG6581B
Type	Base-mounted, heavy-duty, single-shaft, single-cycle, outdoor.
Number of turbine stages	3
Design speed	5163 rpm
Type of compressor	Axial flow, heavy-duty.
Stage of compressor	17
Compress ratio	12.2（ISO condition)
Type of combustor	double fuel
Quantity of combustor	10
Power output	42MW
Firing Temperature °C	(2084/1140)
Type of fuel	natural gas & heavy fuel oil
Exhaust Temperature °F/°C	1016/546
Heat Rate† .BTU/kWhr	10,724

3.4.2 Technical parameters of GT1 at base load (30MW)

T_1=32°c+273=305k, T_2=356°c+273=629k, T_4=573°c+273=846k,

T_3=1140°c+273=1413k, $\dot{m}f$ =1.83 kg/sec, power = 30 MW

$r_p=P_2/p_1$= 9.1

From equation (8) we found

629/305=1+ (9.1^ ((1.4-1)/1.4)/$\acute{\eta}_c$

$\acute{\eta}_c$= 0.828

From equation (10) we found

T_a= (305+629)/2

T_a=467K

From equation (9) we found

Cp_{air}=1.0189*10^3-0.13784(467) + 1.9843*10^ (-4) * (467) ^ 2 + 4.2399 * (467) ^ 3*10 ^ (-7)-3.7632*10^(-10)*(467)^4

Cp_{air}=1.023 Kj/KgK

From equation (11) we found

Wc = 1.023*305*((9.1^ ((1.4-1)/1.4)-1)/.828

Wc =331.37 Kj/Kg

From equation (12) we found

846/1413=[1-$\acute{\eta}_t$ (1-(9.1)^((1-1.33)/1.33)]

$\acute{\eta}_t$= 0.951

From equation (13) we found

W_T =1.11*1413*0.951*(1-(9.1)^((1-1.33)/1.33)

W_T =629.218Kj/Kg

$W_{Net} = (\dot{m}_{air}-\dot{m}_{arc}+\dot{m}_f+\dot{m}_w)*W_T - \dot{m}_{air}W_c$

Without water injection

From equation (15)

Assume $\dot{m}_{air}$=0.05*$\dot{m}_{air}$

30*10^3 = ($\dot{m}_{air}$ -0.05$\dot{m}_{air}$+1.83)*629.218-331.37

$\dot{m}_{air}$=108.3Kg/s

$\dot{m}_{arc}$=5.4Kg/s

With water injection

From combustion chamber balance , equation (16)

T_f= 51°c+273=324K

LHV= 45859Kj/Kg

Cp_f = 1.9Kj/Kg

∴108.3*1.023*629+$\dot{m}_f$*45859+$\dot{m}_f$*1.9*324+$\dot{m}_w$*(4.18*298-2260)=(108.3-504+$\dot{m}_f$*(1.11*1413)+5.4*1.023*467

69687.5+46474.6*$\dot{m}_f$–1014.36*$\dot{m}_w$=161391.45+1568.43*$\dot{m}_f$+1568.43*$\dot{m}_w$+2579.9

∴17.4*$\dot{m}_f$-$\dot{m}_W$= 36.5 ⟶ (19)

At case (1) $\dot{m}_W = \dot{m}_f$

$\dot{m}_f$= $\dot{m}_W$= 2.23Kg/s

∴W_{Net} = (108.3-5.4+2.23+2.23)*629.218-331.37*108.3

W_{Net} = 31.7MW

With water injection power increase by 1.7MW

At $\dot{m}_W$=0.5 $\dot{m}_f$

∴$\dot{m}_f$ = 2.16Kg/s, $\dot{m}_W$= 1.1Kg/s

W_{Net} = (108.3-5.4+2.16+1.1)*629.218-108.3*331.37

∴W_{Net} =30.9MW

∴ power in this case increase by 0.9MW

At case (1)

$\dot{m}_f = \dot{m}_f$

Power output = 31.7 MW

$Q_{in} = \dot{m}_f$ * LHV = 2.23*45859 = 102265.57KW

The thermal efficiency

$\acute{\eta}_{th} = W_{Net} / Q_{in}$ = (31.7*10^3)/ (102.266*10^3)

$\acute{\eta}_{th}$ = 0.3099 ≈ 31%

Heat rate = 3600/$\acute{\eta}_{th}$- 3600/0.31 =11612.9Kj/KWh

S.F.C = $\dot{m}_f / W_{Net}$ = 2.23/31.7 = 0.07*10^-3Kg/Kj

At case (2)

$\dot{m}_w = 0.5\dot{m}_f$

$\dot{m}_f = 2.16Kg/s$, $\dot{m}_W = 1.1Kg/s$

Power output = 30.9

$Q_{in} = \dot{m}_f$ * LHV = 2.16*45859= 99055.44KW

$\acute{\eta}_{th} = W_{Net} / Q_{in}$ = (30.9*10^3)/(99.06*10^3)=31.2%

Heat rate = 3600/31.2 = 11538.5Kj/Kwh

S.F.C= 2.16/30.9=0.0699Kg/kj

CHAPTER IV

RESULTS AND DISCUSSIONS

CHAPTER IV

RESULTS AND DISCUSSIONS

4.1 Preface:

This chapter shows all the results of GT (1) at Garri power plant. Power output, Efficiency, fuel consumption and the heat rate, which calculated in the case of water injection and without water injection.

4.2 Water injection effect in gas turbine:

4.2.1 Heat rate:

We found that heat rate directly affected with the increase in water injection parentage as shown in figure below:

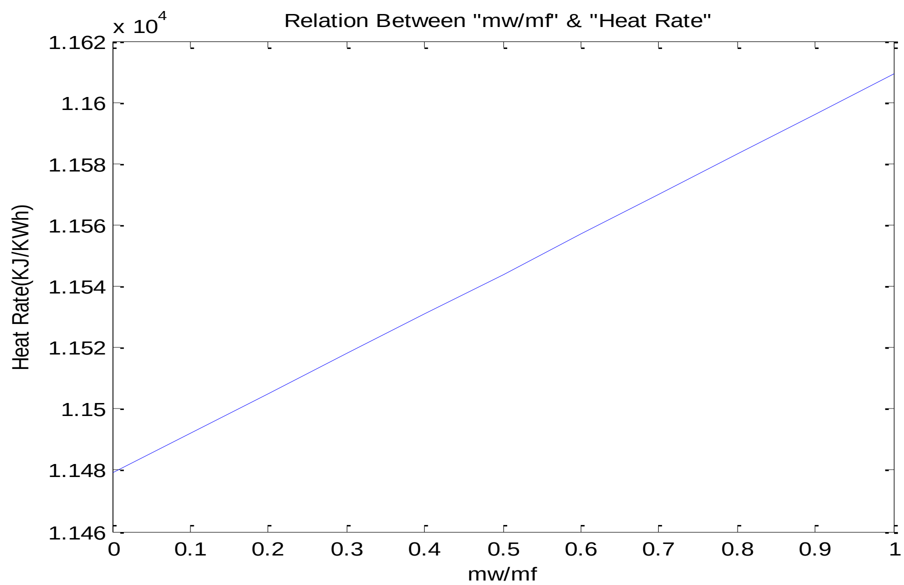


Figure 4.1Relation between water injection & the heat rate

4.2.2 Work net:

The results shows that the net work output from the unit increased due to the increase in water injection as below:

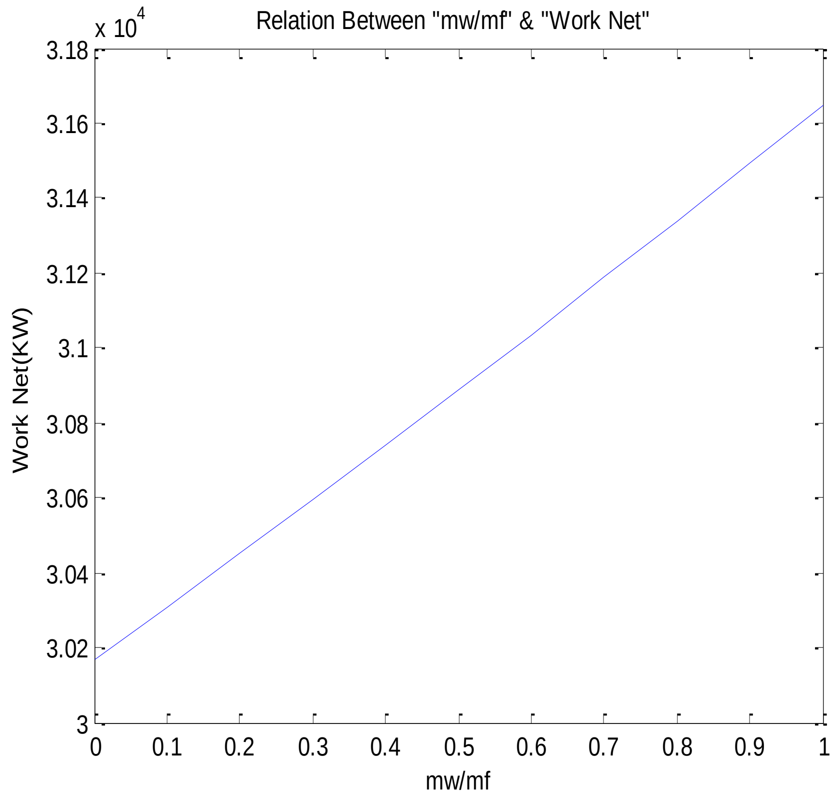


Figure4.2Relation between water injection &Net work output

4.2.3 Specific fuel consumption:

Fuel consumption increased with water injection but the overall efficiency is better than without water injection due to the increase in the Net work output.

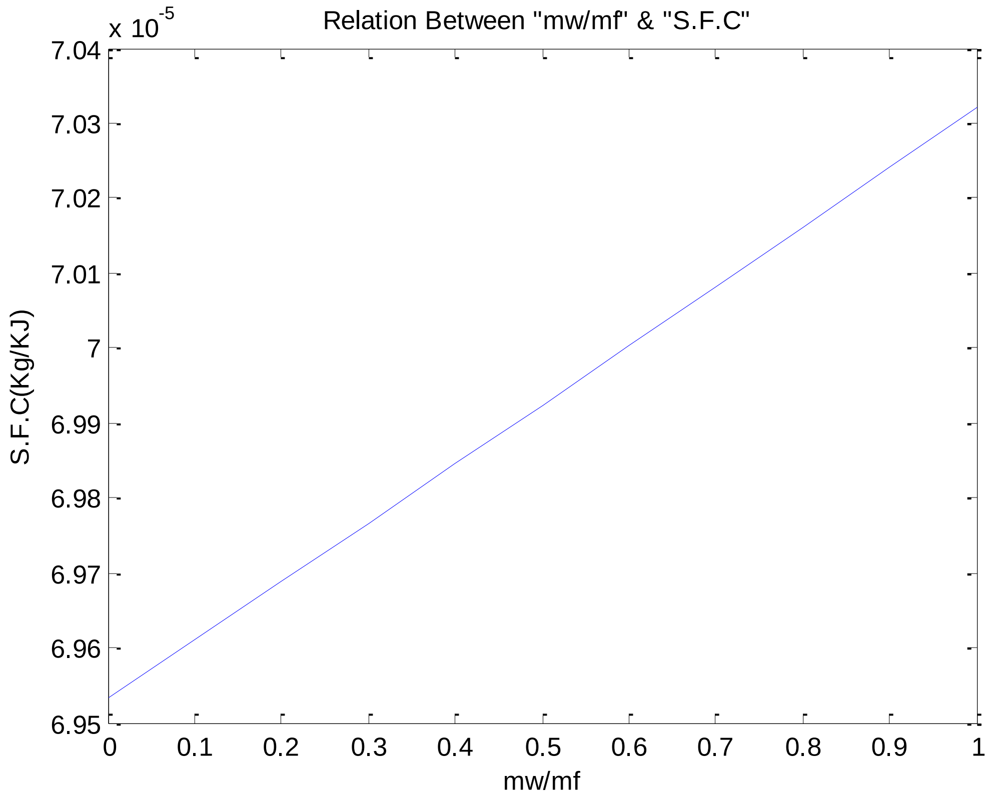


Figure 4.3Relation between water injection & specific fuel consumption

After the analysis found that it's better to use water injection systems to increase the heat rate, Network output which lead to better and higher efficiency. Also the most important thing that water injection directly can decrease the pollution from the burned gases.

CHAPTER V
CONCLUSIONANDECOMMENDATIONS

CHAPTER V
CONCLUSION & RECOMMENDATIONS

5.1 Conclusions:

Thermal power plant used as a base load plants in Sudan. Garri (1, 2) one of the biggest thermal power Sudan also it is the only combined cycle plant in Sudan.

This research investigate the water injection system in gas turbine In this research, we present the results of thermodynamic analysis, regarding gas turbine installations (GTIs) with total water injection in the combustion chamber (CC). Among existing GTIs, the majority are without water or steam injection in the working fluid, while some of them employ partial water/steam injection (GTIWI). The technical solution proposed with total water injection (GTITWI), is designed to realize the desired temperature in the CC by means of the injected water exclusively. The introduced air flow is only the quantity strictly needed to produce combustion. As a result, for many cases, we have higher values for the thermal efficiency in a GTITWI compared with the GTIs without water/steam injection. Consequently, from this point of view, GTIWI are situated between GTI and GTITWI. Also, like for all GTIWI, for GTITWI too, using water injection results in much lower values for NO_x emissions.

Water is a more efficient flame coolant than steam owing to evaporation inside the flame. The injection of steam or water can occur at various points along the gas path in the gas turbine. Owing to the application of cooling, the firing temperature can be further increased, and thus the gas turbine Efficiency can be enhanced.

5.2 Recommendations:

- Water injection system requires large quantities of treated water and it lowers the gas turbine efficiency, especially when water is injected so chemists and operators must trained.
- Put clear guidelinesto chemistry section about the stability of the quality of the water.
- Fix parameters reading and protection issues to avoid over injection of water.
- Operators must train well to avoid human errors, which lead to system failure.
- When either steam or water used for power augmentation, the control system normally designed to allow only the amount needed for NO_x abatement until the machine reaches base (full) load. At that point, additional steam or water can be admitted via the governor control.

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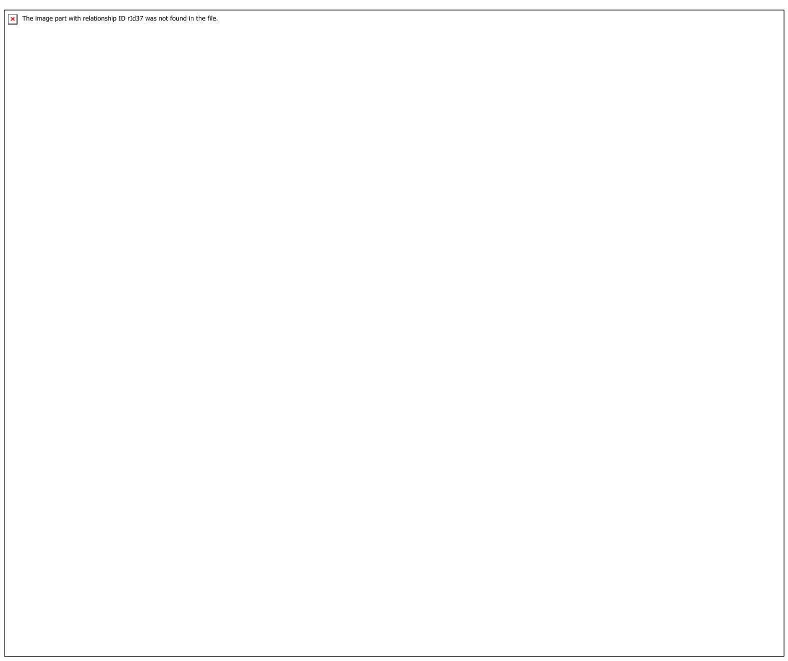
Appendix

Garri One Combined Cycle



Gas Turbine Number One

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Gas Turbine Combustion Chamber

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Gas Turbine Mark V

Mat lab code:

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Figure 3.3 Matlab code