ELECTRIC POWER RESEARCH INSTITUTE

Fuel Composition Impacts on Combustion Turbine Operability

1005035

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Technical Update, March 2006

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This document was prepared by

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This document describes research sponsored by the Electric Power Research Institute (EPRI)

This publication is a corporate document that should be cited in the literature in the following manner:

Fuel Composition Impacts on Combustion Turbine Operability. EPRI, Palo Alto, CA: 2006. 1005035.

ABSTRACT

Most new CT plants today area permitted at low emission limits for NO_x and CO, leading to greater use of lean, pre-mix combustion of natural gas in dry, low-NO_x (DLN) combustors. These combustors are typically fine-tuned for a narrow range of fuel properties. At the same time, the increasing variability of natural gas supplies, deregulation of the gas industry, and increasing use of liquefied natural gas (LNG) has led to more variability in fuel properties and a need for greater flexibility in firing gaseous fuels in combustion turbines. This report reviews the various issues and options associated with changes in gaseous fuel composition for low-emission combustion turbines. Specific topics addresses include fuel flexibility, fuel cleanliness and conditioning, turbine monitoring and DLN tuning.

ACKNOWLEDGEMENTS

This project was the result of multiple technical inputs from key individuals. EPRI would like to acknowledge the following individuals and their organizations:

Mike Klassen and Rick Roby of Combustion Science and Engineering who reported on research to understand the effect fuel composition on gas turbine emissions and operability.

Colin Wilkes of General Electric Company who provided valuable information on industry experience concerning fuel gas cleanliness issues and fuel gas conditioning equipment design practices.

Geo Richards of the Department of Energy who reported on advance, low-emission combustion design fundamentals.

Bruce Rissing of Siemens Power Corporation who provided technical data on fuel gas interchangeability protocols.

Carlo .Castaldini of CMC Engineering who provided general industry practices on combustion dynamics monitoring and dry, low NO_x combustion system tuning.

Mike Dwyer and Dan Turley of Portland Gas and Electric (PGE) Company who hosted EPRI to demonstrate in-house procedures implemented to monitor and avert damaging combustion dynamics in a 7FA machine.

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ISSUES AND OPTIONS FOR LOW-EMISSION, FUEL-FLEXIBLE COMBUSTION TURBINES

Introduction

The natural gas supply profile world wide is shifting due to the narrowing gap between supply and demand. As a result, supply sources may not be processed to historical levels due to the changing market for natural gas liquid (NGL). Unprocessed gas supplies result in an increase in higher hydrocarbons and an increase in hydrocarbon dew point (HDP). To meet increasing world demand,, increased use of liquefied natural gas (LNG) imports, which also may contain higher hydrocarbons, will be necessary . The transition from historical fuel compositions to the evolving fuel profile including unprocessed gas as well as LNG imports presents a technical challenge for combustion turbine operators and equipment manufactures.

The advent of dry, low NO_x (DLN) combustion systems in the power generation industry has changed the stability range of these machines as compared to those with older, diffusion flame combustion systems. This movement has changed the sensitivity of these machines to fuel composition. The physics and chemistry of lean-premixed (LPM) combustion makes DLN combustion turbines more sensitive to the composition of fuel than traditional diffusion flame systems. A change in composition effects fundamental thermo-physical and chemical properties of the fuel, as well as the chemical kinetics of the combustion process. The changes in these properties manifest themselves in various phenomena, including lean blowout, autoignition, and elevated CO and NO_x emissions.

This report focuses on the role of natural gas composition on combustion turbine operations including machine life and pollutant emissions production. It is intended to provide the technical guidance necessary to help assess and mitigate the impact of natural gas composition on both combustion stability and emissions of modern combustion turbines.

For readers unfamiliar with the combustion, Chapter 1 is provided as a tutorial on concepts and nomenclature that are used throughout this report. Chapter 2 presents information on natural gas properties and interchangeability issues. The technical reasons for fuel sensitivity in combustion turbine systems are outlined in Chapter 3. Mitigation strategies including fuel gas conditioning and combustion turning options are presented in Chapters 4 and 5, respectively.

2
HOW FUEL BURNS

Combustion Principles and Terminology

The fuel/air ratio (f/a) expressed on a volume or mass basis is a basic parameter used to characterize a combustion process. The theoretical amount of air to consume all of the fuel is referred to as the stoichiometric f/a ratio. Adding air produces fuel-lean combustion, and reducing air produces fuel-rich combustion. Because each fuel type has a different stoichiometric f/a ratio, the convention is to normalize the fuel/air ratio yielding the widely used equivalence ratio Φ.

$$= \frac{(f/a)}{(f/a)_{stoich}}$$

Using the equivalence ratio, the combustion of different fuels is referred to as lean if $\Phi<1$ or rich if $\Phi>1$.

The flame temperature is another basic combustion process descriptor. Flame temperatures are directly related to the energy balance between combustion reactants and products. In principle, the highest temperatures would result at $\Phi=1$ because all the fuel and oxygen are theoretically consumed. In practice, the effects of species dissociation and heat capacity shift the actual peak temperature to $\Phi=1.05$.

The fuel composition directly affects the flame temperature. The list below compares calculated flame temperatures of two hydrocarbons, carbon monoxide and hydrogen for stoichiometric combustion in ambient air.

Methane	CH_4	2223°K, 4033°F
Propane	C_3H_8	2261°K, 4101°F
Carbon Monoxide	CO	2381°K, 4318°F
Hydrogen	H_2	2370°K, 4298°F

As shown the methane temperature is 150°K, 303°F lower than hydrogen and carbon monoxide. Thus natural gas containing a high percentage of methane inherently produces less emission than syngases containing high percentages of H_2 and CO.

Premixed Flames

There are two basic flame categories: premixed flames and diffusion flames. With a premixed flame, fuel and oxidizer are mixed before arriving at the flame. The combustion rate can be characterized by the flame speed—a measure of how fast a flame will propagate through an unburned mixture. A Bunsen burner is a classic example of a premixed combustion. The flame cones that are visible on these burners result from the flame speed equaling the premixed gas velocity. The balance between the flow rate and flame propagation is modulated by the flame surface area. Within limits, the flame surface can contract or grow to adjust for a given flow rate, or changes in the flame speed due to fuel type, fuel concentration or ambient temperature. The flame will blow-off if the flow rate is too high compared to the flame speed. Conversely, the flame will propagate into the approach flow creating a flashback condition when the flame speed is much higher than the local gas velocity. Thus, the flame speed effectively determines many of the features of a particular combustion process.

Many hydrocarbon fuels have similar (maximum) laminar flame speeds, around 40 cm/sec, 16 in/sec at standard conditions. However, hydrogen has a much greater flame speed, more than 240 cm/sec, 95 in/sec at standard conditions. This presents design challenges for fuels with significant hydrogen content. While laminar flame speed is an important combustion parameter, practical engine applications operate in the turbulent regime with mixtures preheated by compression. The joint effect of premixed gas temperature and turbulence levels is to significantly increase the flame propagation speed. Equivalence ratio also significantly influences flame speed. For hydrocarbons, the flame speed is highest just above $\Phi=1$. Whereas hydrogen and carbon monoxide have their peak flame speed near $\Phi=2$. For sufficiently lean conditions, the flame speed drops to zero, extinguishing the flame. This condition is referred to as the weak limit.

Autoignition is another important consideration in premixed combustion. Flame propagation is not needed to initiate a combustion reaction. At sufficiently high ambient temperatures reactions can occur spontaneously throughout the premixed gases. The time needed for initiating spontaneous combustion is referred to as the autoignition time. Autoignition time is influenced by ambient pressure and fuel composition and is a key gas turbine premixed combustor design parameter.

As discussed in Chapter 3, longer hydrocarbons (greater carbon/hydrogen ratios) or fuels with significant hydrogen content tend to have shorter autoignition times, making these fuels difficult to use without uncontrolled ignition in high-pressure premixed gas turbines.

Diffusion Flames

Fuel and oxidizer are supplied to a reaction zone in an unmixed state to create a diffusion flame. As the two species diffusively mix and react, heat is released. A stoichiometric reaction zone is created where the diffusion of reactants balances the generation of products and heat. In this reaction zone, flame temperatures approach the stoichiometric value, independent of excess air mixed in downstream. If the reaction rate is very rapid, combustion is essentially controlled by species diffusion, and the impact of fuel type can be related to the physical properties of the fuel. For example, in industrial burners, changing fuel types can affect the length of the flame reaction zone, with hydrogen and CO flames being the shortest at comparable conditions simply because these fuels require less oxidant to complete combustion.

Diffusion Flame versus Lean Premixed Combustion Systems

In conventional diffusion flame combustors, fuel and air are fed separately into the flame zone. The heat release rate depends on the flow rates and the degree of mixing. The flame temperature is near the maximum limit for the mixture provided. The flame is stable, but because of high primary zone temperatures, NO_x emissions can be high, and radiation from the flame can cause the combustor liner life to decrease. Older turbines, especially those built prior to 1990, burn in diffusion flame mode.

In lean-premixed combustors, the fuel and air are mixed upstream of the combustion zone, and the combustion takes place at a specific fuel-air mixture. Flame temperature is controlled by the mixture fed into the combustor. Most modern, low emission turbines operate in lean premixed mode referred to as dry, low NO_x (DLN) turbines. By controlling the primary zone temperature, the burner is tuned to produce an optimal balance between CO and NO_x emissions.

The advantages of a lean-premixed combustion mode include low NO_x emissions and increased combustor liner life due to the lowered primary zone temperatures in the combustor. Figure 2-1 indicates the relationship between NO_x production and flame temperature. Lean-premixed combustion allows the primary zone temperature to be controlled to limit NO_x and CO exhaust emissions. In diffusion combustors, spraying water into the flame zone lowers the primary zone temperature and controls the production of NO_x. Unfortunately, this technique is limited since the addition of water essentially quenches the flame, increasing CO and unburned hydrocarbon emissions. As a result other technology must be used to substantially reduce NO_x emissions in diffusion combustors, such as exhaust gas catalytic converters.

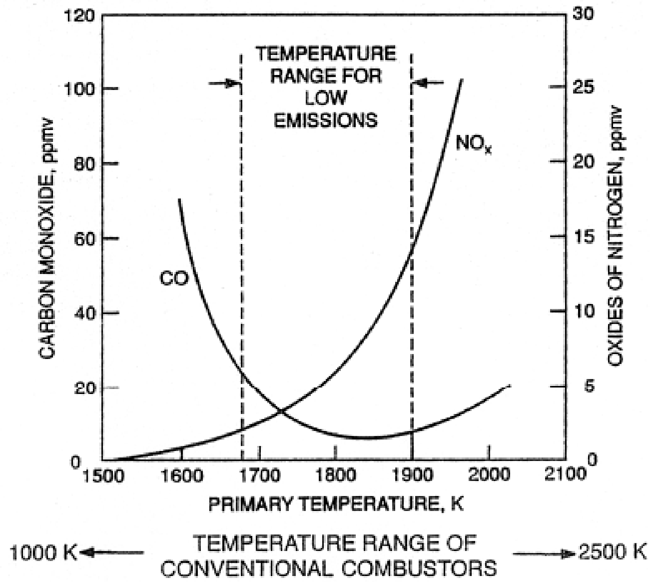


Figure 2-1
CO and NO_x Emissions versus Combustion Zone Temperature [Lefebvre, 1998]

Pollutant Species

There are three major pollutant species routinely considered in generation from combustion: unburned hydrocarbons (UHC), carbon monoxide (CO), and the oxides of nitrogen (NO and NO_2, collectively called NO_x). Both UHC and CO are the products of incomplete combustion. Given sufficient time and high enough temperatures, these two pollutants will be further oxidized to carbon dioxide and water. In practice, completing oxidation is often a design challenge. For instance, poorly controlled cooling flows in turbine combustors can entrain partial combustion products, quenching CO oxidation. Low heating value fuels produce a relatively low flame temperature and require long residence time for complete combustion. NO_x pollutants are generated by oxidation of nitrogen in the combustion air, or in some instances, in the fuel. Diffusion burning occurs at temperatures high enough to oxidize atmospheric nitrogen, producing the pollutant NO, which can further oxidize to NO_2, with both pollutants collectively described as NO_x. Because NO_x plays a role in the production of photochemical smog and acid rain, stringent regulations of NO_x have been established in the U.S. and abroad. Reducing NO_x emissions is a major motivation for turbine combustor development programs.

Fundamentals of NO_x Formation

The collective emissions of NO and NO_2 are referred to as oxides of nitrogen (NO_x). NO_x is generated during combustion of hydrocarbon fuels through a variety of mechanisms. There are five primary pathways of NO_x production in the gas turbine engine combustion process: thermal NO_x, prompt NO_x, NO_x from the N_2O intermediate reactions, fuel NO_x, and NO_x formed through reburning. When burning lean mixtures of natural gas, the thermal NO_x and N_2O intermediate pathways are of primary importance.

Thermal (Zeldovich) NO_x Mechanism

Thermal (or Zeldovich) NO_x is created through the oxidation of nitrogen introduced with the combustion air. This mechanism of NO_x generation is highly temperature dependent, and the production rate is non-linear. Hence, small increases in temperature can lead to large increases in NO_x. This trend is shown in Figure 2-1, where the exponential nature of NO_x as a function of temperature is evident.

Thermal NO_x reactions take place quickly (on the order of milliseconds) and are mostly influenced by temperature, atomic oxygen concentration and residence time. Thermal NO_x is exponentially dependent on temperature, so lowering of the peak combustion temperature can help reduce the amount of thermal NO_x from a gas turbine. A change in fuel composition can cause the flame temperature of the fuel to change, which in turn can affect the NO_x output.

N2O Intermediate NO_x Mechanism

For lean, premixed combustion at gas turbine conditions (i.e. high temperature and high pressure), the nitrous oxide (N_2O) pathway is an important mechanism for NO_x formation. This pathway involves the attack of flame radicals on the N_2O molecule, creating nitrogen and nitric oxide. Emissions of N_2O are not significant, but N_2O serves as an intermediate to NO emissions. NO_x emissions as a result of the N_2O mechanism are significant at the low equivalence ratios (less than 0.6) that are prevalent in current DLN combustors.

NO_2 Formation

NO_2 is a reddish-brown, highly reactive gas. It is important because concentrations of this species above 10 ppm the exhaust of typical power producing gas turbines will create a visible plume (i.e. a ‘brown’ or ‘yellow’ plume). A visible plume can cause environmental compliance problems and is undesirable from a public relations point of view. NO_2 plumes are not usually an issue for gas turbines with NO_x exhaust treatment units or with total NO_x concentrations less than 10 ppm. In gas turbine applications, NO_2 is generally created by conversion of NO to NO_2 after the product gases have left the combustion chamber. The conversion process predominately occurs in the exhaust plume at lower temperatures (800 – 1000°K, 980 – 1340°F), and is helped by reactions with unburned hydrocarbons and CO. Temperatures in this range are present at some points after the exhaust gases have left the combustion chamber.

The conversion of NO to NO_2 has been an area of active study for over a decade due to the desire to eliminate the visible plumes from power plants. The composition of the exhaust stream plays an important role in the conversion of NO to NO_2. Hence, the presence of higher hydrocarbons found in unprocessed natural gas and some LNG imports increases the tendency for NO_2 conversion by up to several orders of magnitude as compared to methane.

Fuel Composition Impacts

Gas composition impacts the amount of emissions generated, and the stability of combustion. It also has an impact on the characteristics that are related to the durability of the combustion system. As discussed in Chapter 5, the dynamic behavior of the combustion system is influenced by the type of fuel used. Changes in fuel composition can impact critical parameters, such as flame position, flashback and autoignition inside the combustor.

As discussed in Chapter 2, fuel gases with varying Wobbe Index have to be evaluated for combustion stability over the entire load-range and for the capability of the turbine to follow load transients. An important parameter for these considerations is the range of flammability limits.

The flammability range or the ratio of flammability limits is the upper flammability limit divided by the lower flammability limit. The upper flammability limit is the maximum fuel percentage (volumetric) mixed with air that will still light and burn when exposed to a spark or other ignition source. The lower is the minimum fuel percentage to sustain combustion. Different gases have different ranges of flammability. As the fuel/air ratio changes with the engine load, the combustor must support combustion over a range of these ratios to prevent flame-out.

As discussed earlier, flame temperature impacts the amount of NO_x emissions. Across the flammability range the air-fuel mixtures will burn at different temperatures. As the fuel/air ratio is increased from the lower flammability limit, the flame temperature will increase. The flame temperature has a significant impact on the NO_x production rate (Figure 2-1). Non-combustibles lower flame temperatures, while hydrogen, carbon monoxide, and heavier hydrocarbons increase it.

3

UNDERSTANDING FUEL GAS COMPOSITION AND INTERCHANGEABILITY

Changing Natural Gas Composition

Table 3-1 shows the range of composition of processed natural gas (NG) supplies in the U.S. as determined by the Gas Research Institute in 1992. As can be seen, processed natural gas is comprised primarily of methane (CH_4), with minor quantities of higher hydrocarbons such as ethane (C_2H_6), and propane (C_3H_8) and larger hydrocarbon constituents that are characterized by the number of carbons, as C_4+. It also contains measurable quantities of inert gases including carbon dioxide (CO_2), and nitrogen (N_2).

Table 3-1
Concentration of Fuel Constituents for Natural Gas in the United States [Liss, 1992]

		Mean	Minimum*	Maximum*	10^{th} Percentile	90^{th} Percentile
Methane	Mole %	93.9	74.5	98.1	89.6	96.5
Ethane	Mole %	3.2	0.5	13.3	1.5	4.8
Propane	Mole %	0.7	0	2.6	0.2	1.2
C 4 +	Mole %	0.4	0	2.1	0.1	0.6
$CO_2 + N_2$	Mole %	2.6	0	10	1	4.3
Heating Value	MJ/m^3	38.46	36.14	41.97	37.48	39.03
Heating Value	Btu/scf	1033	970	1127	1006	1048
Specific Gravity		0.598	0.562	0.698	0.576	0.623
Wobbe Index	MJ/m^3	49.79	44.76	52.85	49.59	50.55
Wobbe Index	Btu/scf	1336	1201	1418	1331	1357
Air/Fuel Ratio	Mass	16.4	13.7	17.1	15.9	16.8
Air/Fuel Ratio	Volume	9.7	9.1	10.6	9.4	9.9
Molecular Weight	g/mol	17.3	16.4	20.2	16.7	18
Critical Compression Ratio		13.8	12.5	14.2	13.4	14
Methane Number		90	73.1	96.2	84.9	93.5
Lower Flammability Limit	Volume %	5	4.56	5.25	4.84	5.07
Hydrogen: Carbon Ratio		3.92	3.68	3.97	3.82	3.95

***without peakshaving**

Compared to processed natural gas, unprocessed natural gas is richer in higher hydrocarbons such as C_4, C_5, and C_6+. With liquefied natural gas (LNG), impurities such as water, CO_2 and heavy hydrocarbons (C_5+) are removed during processing. As discussed later, LNG imports contain varying concentrations of C_2 and C_3 components that result in elevated heating values relative to traditional domestic supplies.

Historically, fuel heating value and Wobbe Index have been key measures of fuel interchangeability. As discussed later in this chapter, the Wobbe Index is related to the thermal input to a burner (Btu per hour). Table 3-1 shows the mean higher heating value (HHV) for domestic natural gas has been 1033Btu/scf, with tenth and ninetieth percentile levels of1006 Btu/scf and 1048 Btu/scf, respectively. Likewise, the mean Wobbe Index for traditional domestic gas has been 1336, with a narrow range between the tenth and ninetieth percentile levels of 1331 and 1357, respectively.

As can be seen from Table 3-1, natural gas in the U.S. has been relatively consistent in composition, especially in a given region and point of consumption. As a result, most gas turbines are ‘tuned’ for operation on a given fuel composition.

Although the data indicate that gas composition is reasonably consistent, there may be significant local exceptions. Exceptions can result from the following:

- Local gas production from oil producing reservoirs, where heavier hydrocarbons such as propane and ethane are produced along with NG
- Local gas sources such as refinery gases or landfill gases
- Elevated levels of higher hydrocarbons due to insufficient blending
- Insufficient removal of higher hydrocarbons from processing plant upsets
- “Peakshaving,” blends of non-methane gases (primarily propane) with air to meet peak winter demand

In addition to fuel composition variation, elevated levels of water vapor can be encountered in natural gas. As discussed in Chapter 4, normal pipeline levels of water vapor (typically less than 7 lb/million scf, 112 kg/million scm) can be exceeded due to several factors, including gas dryer failures at flooded underground storage facilities, failures in other upstream processing equipment, and leaks in low-pressure piping that passes below bodies of water or the local water table.

Local distribution companies rely on various sources of supplemental gas to meet short-term peak gas loads. These sources include gas from underground storage, stored liquid natural gas, and blending propane and air with pipeline gas. Propane/air peakshaving blends are adjusted to approximate the heating value and Wobbe Index of the natural gas while the oxygen in the dilution is kept below the flammability limit. Blending is necessary since propane has about twice the volumetric heating value of natural gas.

Table 3-2 shows the mixture composition by mole percents of constituent gases for the given base natural gas at varying levels of peakshaving. Note that the increased concentration of propane and nitrogen becomes significant with increasing levels of peakshaving. Methane levels also fall significantly with increasing levels of peakshaving. As noted throughout this chapter, even brief excursions in fuel composition can create serious problems for high-performance gas turbines.

Table 3-2
Representative Peakshaving Mixtures: Air/Propane/Natural Gas [Schaedel et al]

Constituent (mol%)	Base Gas	10% Peakshaving	15% Peakshaving	20% Peakshaving	25% Peakshaving
Methane	94.69	85.22	80.48	75.77	71.04
Ethane	2.93	2.68	2.55	2.42	2.30
Propane	0.22	3.85	5.44	7.48	9.29
C4 and Higher	0.05	0.13	0.19	0.21	0.25
Nitrogen	1.44	6.04	8.53	10.63	12.93
Oxygen	0.0	1.26	1.94	2.52	3.15
Carbon Dioxide	0.67	0.60	0.57	0.54	0.50
HHV (Btu/scf)	1015.6	1014	1007	1013	1012
Wobbe Index	1329.5	1262	1225	1203	1177

The traditional natural gas supply profile described above is shifting due to the narrowing gap between supply and demand. Through the late 1990's, a growing natural gas liquids (NGL) market and relatively low cost supply had created an incentive to maximize the removal of higher hydrocarbons from the domestic gas supply, particularly in the Gulf Coast and Mid-continent supply regions. The 2001–2005 steep increases in gas prices increased exploration, but the production from these new wells has not offset declines in the historic supply basins. As a result, domestic supply sources are no longer being processed to historic levels. Unprocessed gas supplies result in an increase in higher hydrocarbons coupled with an increase in hydrocarbon dew point.

In addition, regions like the Appalachian Basin, Rocky Mountains, and Canada have begun supplying larger quantities of gas with their own specific gas composition. As a result, divergent gas compositions between regions have begun to develop, and this situation is likely to persist and further evolve as supply continues to change. Most prominent has been the increase in coal-seam production in the Rocky Mountains and Appalachian basin. This gas is composed almost entirely of methane and inerts (nitrogen and CO_2) that yields a heating value significantly lower than traditional domestic production.

Finally, with three of the four existing regasification terminals regaining active status (the fourth remains in service), LNG imports have begun to rise, and future imports are forecasted to account for a more significant percentage of total North American supply. A similar structure is occurring overseas. For instance, depletion in North Sea gas production is anticipated to be offset by LNG imports to the U.K. and European countries. Table 3-3 shows the variability of natural gas composition worldwide and includes a 'typical' U.S. gas composition for comparison. Table 3-3 shows that most LNG imports will have considerably higher concentrations of C_2, C_3, higher heating values, and higher Wobbe Index than historically found in the U.S.

Table 3-3
Fuel Composition from Various Fuel Sources around the World
[Source: Siemens-Westinghouse Power Corporation]

		Methane	Ethane	Propane	C4+	LHV	HHV	Wobbe
		mol %	mol %	mol %	mol %	BTU/scf	BTU/scf	Index
Typical US		95.7	3.2	0.7	0.4	949	1052	1379
Known GT Experience		89.6	8	1.5	0.9	1006	1112	1412
	Brunei	89.76	4.75	3.2	2.29	1036	1144	1429
	Trinidad	96.14	3.4	0.39	0.07	940	1041	1374
	Algeria	87.83	8.61	1.18	0.32	991	1099	1405
	Indonesia	90.18	6.41	2.38	1.03	1010		1279
	Nigeria	90.53	5.05	2.95	1.47	1017		1283
LNG Source	Qatar	89.27	7.07	2.5	1.16	1018	1126	1419
	Abu Dhabai	85.96	12.57	1.33	0.14	1020	1127	1420
	Malaysia	87.64	6.88	3.98	1.5	1045	1155	1434
	Australia	86.41	9.04	3.6	0.95	1036	1145	1429
	Oman	86.61	8.31	3.32	1.76	1051	1161	1438

The economics of LNG transportation are such that LNG marketers prefer to have the ability to purchase LNG from a wide range of supply sources, most of which contain almost no inerts (such as CO_2 and N_2) and more non-methane hydrocarbons, such as ethane, propane and butanes, than historical U.S. supplies. Non-methane hydrocarbons have a higher energy density; that is to say, they contain more Btu's per cubic foot. Higher energy density results in a more efficient production, storage and transportation of LNG, thereby increasing profit margins.

Interchangeability

Historically, the gas industry has used a variety of calculation methods to define the interchangeability of fuel gases for traditional end use equipment. These methods have been based on empirical parameters developed to fit the results of interchangeability experiments. These experiments focused on industrial diffusion flame burners and partially premixed home appliances. The result of these studies has been a set of single and multiple index calculations that apply to industrial diffusion flame burners. These studies took place before the advent of lean-premixed gas turbines that are prevalent today. Recently a more robust definition of interchangeability has been created to better account for the specific issues that end-users face.

In 2005, the Natural Gas Council + Gas Interchangeability Task Group reached consensus on the following updated definition of interchangeability as:

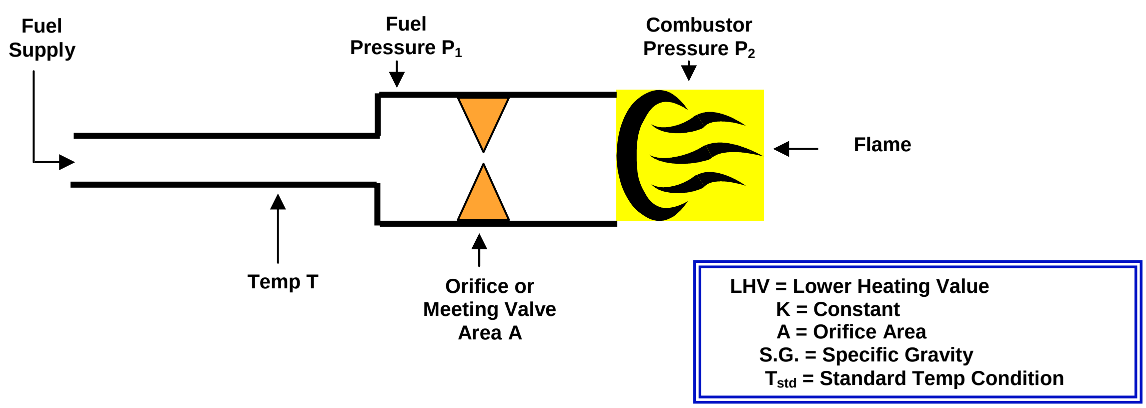
> "The ability to substitute one gaseous fuel for another in a combustion application without materially changing operational safety, efficiency, performance or materially increasing air pollutant emissions."

This new definition accounts for the additional environmental and operational issues faced by many natural gas suppliers and end users. Generally, interchangeability has been determined through indices calculations. The most commonly used indices are the Wobbe and Weaver Indices as discussed below.

Wobbe Index

The most common single index parameter is the Wobbe Index sometimes referred to as the *Interchangeability Factor*. The Wobbe Index characterizes the similarity of gas mixtures based on the heat released for a given orifice-controlled device. The definition of the Wobbe Index is based on the heating value and specific gravity of a gas, and is related to the burner thermal input (Btu per hour). While the Wobbe Index is an effective, easy to use screening tool for interchangeability, the Wobbe Index alone is not sufficient to completely predict gas interchangeability because it does not adequately predict all combustion phenomena.

Wobbe Index is proportional to the heat release from a fuel at constant pressure as shown in Figure 3-2. Two different gaseous fuels with the same Wobbe Index supplied to fixed metering orifice at the same fuel pressure P_1 will produce the same heat output at a given combustor pressure P_2. The equation for the heat input can be derived by multiplying the standard orifice flow equation by the fuel heating value.



$$Heat\ Input = KA \underbrace{\frac{LHV}{\sqrt{S.G.}}\sqrt{\frac{T_{std}}{T}}}_{\text{Wobbe Index}} \sqrt{P_1(P_1 - P_2)}$$

Figure 3-1
Definition of the Wobbe Index [Richards, G. A. et al, 2001]

As shown, this leads to the definition of a temperature-corrected Wobbe Index also called the Modified Wobbe Index and defined as:

$$\textit{Modified Wobbe Index} = \frac{HV}{(S.G.)^{0.5}}\left[\frac{T_{std}}{T}\right]^{0.5}$$

Where HV is the heating value, S.G. is specific gravity of the fuel, T_{std} is the standard temperature condition in units of absolute temperature (degrees Kelvin or Rankine) and T is the fuel delivery temperature.

Any change in fuel heating value will require a corresponding change in fuel flow rate to the combustion turbine to maintain the proper total energy input. For older, diffusion-type combustors, the control system likely can handle variations in the Modified Wobbe Index as large as ±15 percent. However, for dry low NO_x (DLN) combustors, the control window may be as narrow as ±3 percent. Corresponding velocity changes through the DLN system's precisely sized fuel-nozzle orifices can cause flame instability, resulting combustion dynamics as discussed in Chapter 5. At the low end of the trouble spectrum, Wobbe Index variations will force users to conduct more frequent tuning of their DLN combustors. At the high end of the trouble spectrum, the variations could be large enough and prolonged enough to cause pressure pulsations that catastrophically destroy the combustor and downstream hot gas path components.

Moreover as described in Chapter 3, two fuels with the same Wobbe Index may not necessarily produce same fuel air ratio within a DLN combustor. Thus, two fuels even with the same Wobbe Index can produce significantly different emission and flame stability characteristics.

Weaver Indices

Since the Wobbe Index is only concerned with matching heat release for a given burner, other indices have been developed that monitor for interchangeability of other flame properties. Multiple index methods date back to the late 1940's and includes the AGA Bulletin 36 Indices and the Weaver Indices. The multiple index techniques have a history of widespread and satisfactory use. However, as empirical models, the multiple index methods also have limitations based on the burner designs and fuel gases tested in the development research. In general, the new gas supply, called "substitute gas" is evaluated for behavior of specific combustion phenomena, including flame lifting, flashback, yellow tipping and incomplete combustion, relative to an "adjustment gas" or the gas normally used in the past with properly adjusted equipment.

The Weaver Indices incorporate the effect of a change in gas composition by compensating for the change in air needed for combustion at stoichiometric conditions. These indices were developed to create a set of calculations that could be performed easily in order to test the interchangeability of a natural gas for use in industrial and home appliances. The experiments were carried out with burners that can be classified as industrial burners with diffusion flames or a limited amount of premixing.

These indices were developed before the introduction of DLN combustors, and the combustion fundamentals that apply to industrial burners (diffusion) do not apply to DLN (lean-premixed) combustors. Therefore, the fundamental properties of fuel mixtures must be re-investigated to understand how the properties of the fuel change due to the variation in fuel mixture composition.

Interim Interchangeability Guidelines

In 2004, the Washington-based Natural Gas Council (NGC) formed a working group to identify gas-quality criteria interchangeability and provide recommendations the Federal Energy Regulatory Commission (FERC). The working group is comprised of natural gas consumers (power generators, industrial and commercial users), suppliers, distributors, and equipment manufacturers. The NRG working group produced and presented a white paper to FERC in March 2005 which urged FERC to establish the following specific criteria (details available at www. aga.org and www.ngsa.org):

> Maximum Wobbe Index = 1400 (based on gross or higher heating value at standard conditions of 14.73 psia, 60°F),
>
> Maximum variation in Wobbe Index = ±4%,
>
> Maximum Higher Heating Value = 1110 Btu/scf,
>
> Maximum mole percent of inert gas = 4%,
>
> Maximum mole percent of butanes-plus (C4+) = 1.5%.

Regrettably for combustion turbine operators, the NGC working group's interim guidelines do not meet gas-quality requirements specified for DLN turbines currently in service. Rather, the NGC working group has challenged combustion turbine manufacturers and users to prove within two to three years that the interim guidelines are unacceptable. Although a great deal of research has been performed by combustion turbine OEMs to assess interchangeability indices, access to much of this data has not been possible because the research was performed on a proprietary basis. As a result, EPRI, DOE and others are developing data to define the impact of fuel gas variability and technologies to mitigate the impacts of quality variations.

If the interim guidelines are adopted and become federal regulations in their current form, the potential impacts to combustion turbine owner/operators may be significant. For example, additional combustion turbine control and/or emissions reduction equipment may be necessary to accommodate fluctuations in gas quality and composition. As discussed in Chapter 4, opacity is one concern given the high butane level found in some LNG supplies. Also, the NGC working group's white paper recommends less-stringent gas-quality standards for supply areas that historically have exceeded the standards proposed in the interim guidelines. These "grandfathered" areas could limit future development of low-emissions combustion turbine-based generating plants.

The bottom line is that operators of DLN combustion turbines can no longer assume that gas supplies will comply with their needs. Current industry trends demand that users be more active in gas-quality and emissions monitoring, and focus more attention on regulatory activity concerning gas quality.

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4

ISSUES FOR DRY LOW NO_x (DLN) COMBUSTORS WITH FUEL VARIABILITY

Fuel Composition Affects on Fundamental Combustion Properties

Fuel composition directly affects many combustion properties including heat release rate, flame speed, autoignition tendencies and flame temperature. Changes in these properties impact turbine performance, including emissions production, combustion dynamics and maintenance schedules. The effects of increases in higher hydrocarbon concentrations in natural gas as applies to DLN combustion systems are summarized below.

Air/Fuel Ratio and Wobbe Index

Table 4-1 compares the fuel properties of methane, ethane, propane, and six natural gas compositions having the same Wobbe Index. The six natural gas compositions were derived by changing the ratio of hydrocarbons (methane, ethane, propane), and adding dilution in the form of nitrogen to an arbitrarily chosen natural gas (NG1) composition. The ideal gas densities listed are calculated at standard temperature and pressure (0°C, 1 atm). The air to fuel ratio at stoichiometric conditions, taking into consideration the effect of dilution nitrogen, is calculated and tabulated in Table 4-1.

Table 4-1
Fuel Properties for Pure Gases and Six Natural Gases of Constant Wobbe Index [Roby, and Klassen et al, 2004]

	Methane	Ethane	Propane	NG1	NG2	NG3	NG4	NG5	NG6
Methane (volume %)	100	0	0	90	85	82	79	77	75
Ethane (volume %)	0	100	0	5	15	0	20	10	0
Propane (volume %)	0	0	100	5	0	14.5	0	9.6	19
Nitrogen (volume %)	0	0	0	0	0	3.5	1	3.4	6
Lower Heating Value (Btu/lb)	21503	20417	19930	21222	21228	20701	21026	20708	20498
Ideal Gas Density (STP, lb/ft^3)	0.0447	0.0837	0.1228	0.0505	0.0506	0.0572	0.0528	0.0572	0.0614
Wobbe Index (Btu/ft^3)	1288	1675	1980	1352	1352	1352	1352	1352	1352
A/F ratio (mass basis)	17.11	15.98	15.57	16.83	16.82	15.79	16.44	15.78	15.22
Equivalence Ratio (Φ) (100 lb air / 3 lb fuel)	0.513	0.479	0.467	0.505	0.504	0.450	0.484	0.450	0.421
Percentage Change in Equivalence Ratio (Φ) from NG1	Not Constant Wobbe			0.00	0.13	10.85	4.18	10.95	16.56

To illustrate how fuel gas with the same Wobbe Index but different composition can effect the operation of a DLN combustor, a comparison of equivalence ratios (Φ) is shown in Table 4-1 for each gas. As discussed in Chapter 1, equivalence ratio is an indicator of flame temperature, and the flame temperature is an indicator of NO_x emissions. The equivalence ratio in the primary zone of the combustor can also indicate if the combustor is approaching lean-blowout, which can result in combustion instabilities.

As indicated in Table 4-1, for a theoretical combustor operating on NG1 with fixed air and fuel flow rates, the equivalence ratio inside the combustor is calculated to be 0.505. If NG1 is replaced with same mass flow rate of NG6, even though the Wobbe Index remains constant, the equivalence ratio drops by 16.6 percent, to a value of 0.421. A change in natural gas composition, such as the one described above, is likely to cause a range of operational issues, such as lean-blowout, poor dynamics, and an increase in CO emissions. DLN combustors are usually tuned over a narrow range of equivalence ratios, and, therefore, must be re-tuned if the fuel composition is significantly altered.

Autoignition Temperature

The minimum temperature at which a fuel-air mixture spontaneously ignites at a specified set of conditions (i.e. pressure, stoichiometry) is referred to as the autoignition temperature. Premature autoignition in the turbine combustor can cause the flame zone to shift to areas not designed to handle the heat and flame impingement. This leads to premature failure of combustion hardware and potentially to extensive damage to downstream turbine blades and vanes.

Figure 4-1 plots autoignition temperature as a function of fuel molecular weight for a range of saturated hydrocarbons. Methane has the highest autoignition temperature of most common hydrocarbons. Also, autoignition temperature decreases exponentially with increasing carbon number. The addition of significant quantities of C_4 and higher hydrocarbons can lower the autoignition temperature to levels around those of the air entering DLN combustors. This substantially increases the possibility of spontaneous ignition prior to the flame zone.

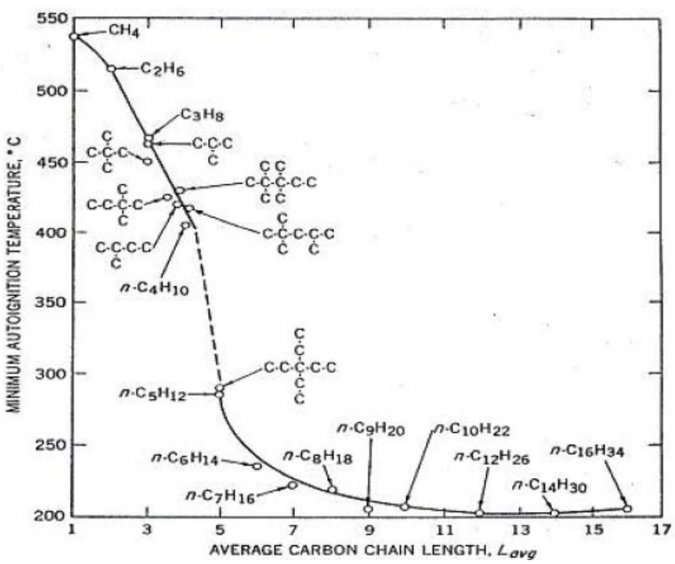

Figure 4-1
Minimum Autoignition Temperature vs. Average Carbon Chain Length [Zabetakis, 1965]

Flashback and Lean Blowout

As discussed in Chapter 1, the velocity at which a flame propagates through a fuel/air mixture is dependent on the mixture composition, including the fuel components and the ratio of fuel to air. The burning velocity (flame speed) of the mixture is directly related to flashback and lean blowout within the combustor. Lean blowout can be defined as the point where flame extinction occurs because the rate of heat released during combustion is below the amount of heat needed to ignite a fresh fuel-air mixture entering the combustion zone. The effect of molecular structure of hydrocarbons on flame speed has been studied extensively. Figure 4-2 shows the laminar burning velocities of various fuels as a function of equivalence ratio.

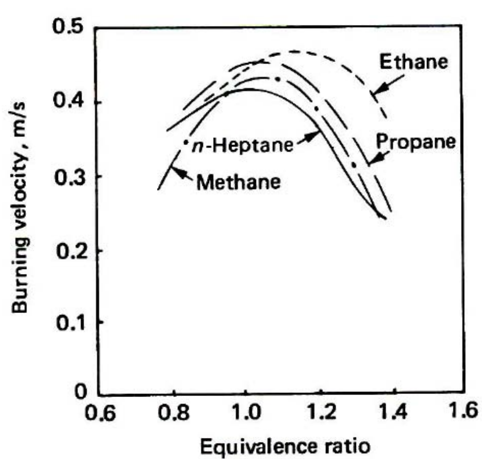

Figure 4-2
Laminar Burning Velocity as a Function of Equivalence Ratio and Fuel Type

[Lefebvre, 1983]

Laminar burning velocity of straight chain hydrocarbons (methane, ethane, propane) reaches its maximum value on the slightly rich side of the stoichiometric point. Of great importance for DLN combustion turbines is the widening difference in burning velocity between methane and other hydrocarbons as the mixture becomes fuel lean (equivalence ratio less than 1).

The variation of laminar burning velocity as a function of equivalence ratio is shown in Figure 4-3. The natural gas sources compared in the study were from Netherlands. The laminar burning velocity of pure methane is shown for comparison. The study shows that laminar burning velocity can vary by over 10% for fuels with constant Wobbe Index. The variation in burning velocity found for constant Wobbe Index fuels can be important because relatively small differences in laminar burning velocity may become amplified in highly turbulent flow as is the case for large industrial combustion turbines.

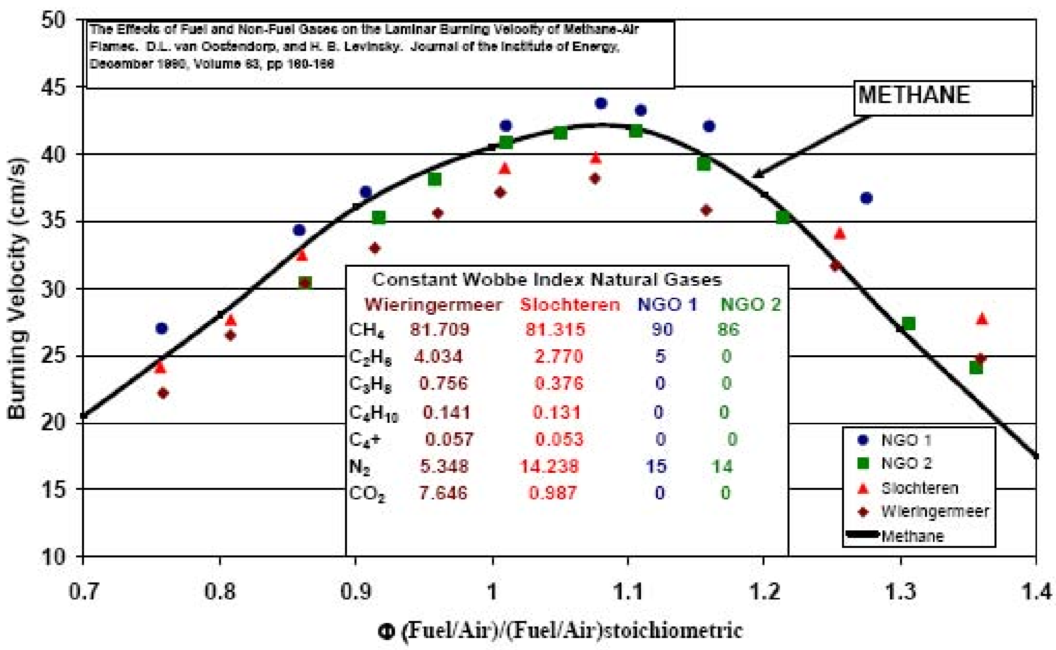

Figure 4-3
Burning Velocities of Various Fuels with a Constant Wobbe Index
[Oostendorp, and Levinsky, 1990]

Changes in burning velocity due to fuel composition differences may increase the propensity for DLN combustors to flashback or blowout under normal operating conditions. These changes can also lead to increased emissions and combustion dynamics, to shortened hardware lifetimes, and to increased maintenance.

Combustion Dynamics

The introduction of DLN technology into the combustion turbine industry compounded the problem of combustion dynamics as discussed in Chapter 5. Combustion dynamics, generally present in the form of pressure fluctuations within the turbine combustor, can occur in any combustion device. DLN machines are particularly susceptible to this issue and in many cases combustor dynamics have limited the operating capacity of a DLN machine in order to maintain hardware life or meet emissions restrictions.

The pressure oscillations associated with combustion dynamics can lead to externally audible tones, sometimes referred to as "humming". Furthermore, if the pressure waves cause combustion hardware to become resonant, significant damage can occur. Combustion dynamics occur due to the coupling of pressure oscillations with the heat release rate. To reduce emissions DLN combustors operate near the lean flammability limit causing the combustion process to become inherently less stable. A number of factors, including fuel composition, influence combustion stability for a given combustor. As revealed in Figure 4-4, changes in fuel composition (shown as changes in the lower heating value (LHV)) affected the level of pressure pulsations (combustion dynamics) of the machine, increasing the pressure pulsations by a factor of 50%.

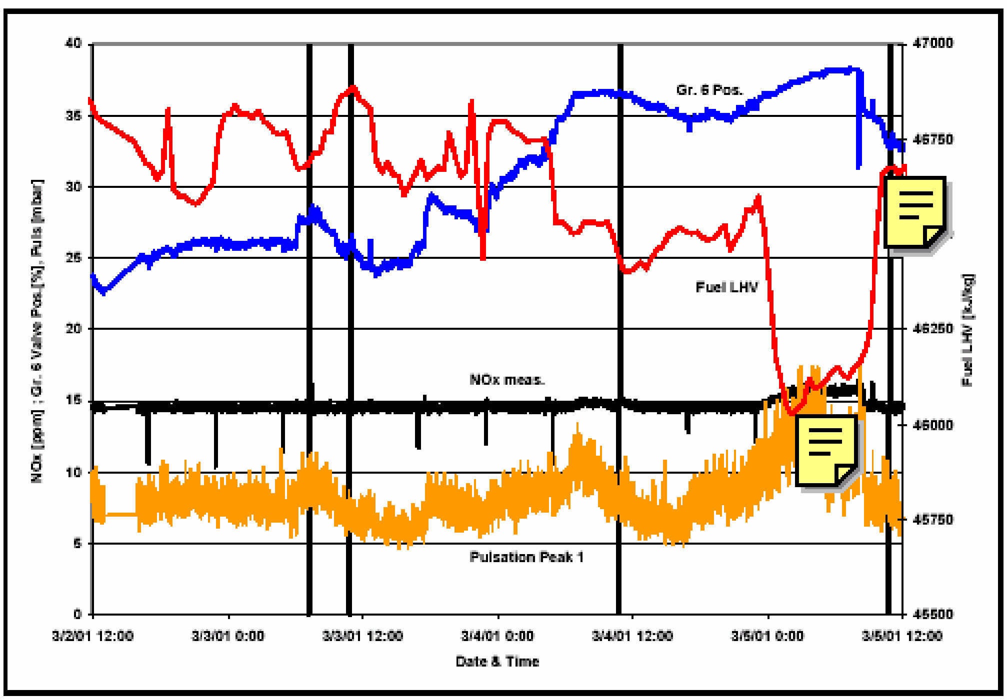

Figure 4-4
Combustion Pulsations and NO_x Emissions as a Function of Fuel Composition (Fuel Lower Heating Value (LHV)) [Nord and Andersen, 2003]

Visible Plume due to NO_2 Discharge

Visible plumes due NO_2 discharge have been observed from turbines without post combustion emission controls and burning fuels with elevated higher hydrocarbon concentration. Typically, this occurs at part load operation, when the combustion efficiency is low, allowing more unburned hydrocarbons to pass through to the exhaust, enhancing the conversion of NO to NO_2. Table 4-2 compares the fuel compositions at two plants with nearly identical combustion turbine hardware. Both combustion turbine plants were producing approximately 25 ppm NO_x (i.e. NO and NO_2) and neither plant had an exhaust treatment unit. One plant in Asia was operating using LNG with fairly high ethane (C_2) concentrations and significant propane (C_3). The second plant located in North America operated with natural gas containing little higher hydrocarbons. The Asian power plant had a consistent brown plume that was not observed for the North American plant.

Table 4-2
Natural Gas Composition of Two Similar Gas Turbine Sites
[Roby, and Klassen et al, 2004]

Component	Plant 1 Composition (%) North America	Plant 2 Composition (%) Asia
Methane	95	89
Ethane (C2)	2	8
Propane (C3)	<1	2
C4 +	<1	<1
Nitrogen	2	<1

Following careful analysis of all the potential causes, it was found that the differing fuel composition could best explain the brown plume. An analysis of the propensity of each of the fuel gases to convert NO to NO_2 in the exhaust was undertaken. Table 4-3 summarizes the results. The Asian plant with its higher concentrations of C_2 and C_3 hydrocarbons was shown to more effectively convert the NO produced in the combustor to NO_2. The added NO_2 from the Asian plant was sufficient to create a visible plume (over 16 ppm as shown in Table 4-3) while the North American plant remained below the visible plume level. The total NO_x is not changed by the conversion of the NO to NO_2.

Table 4-3
Conversion of NO to NO_2 for Two Plants with Differing Natural Gas Compositions [Roby, and Klassen et al, 2004]

50 ppm UHC Total ($[NO]_o$ =25 ppm)						
	Asia	% Conversion	$[NO_2]$	**America**	% Conversion	$[NO_2]$
C1	45	32	8	49.5	35	8.75
C2	4	25	6.25	0.5	<1	~0
C3	1	10	2.5	0.02	<1	~0
I-C4	0.1	<1	~0	0.002	<1	~0
n-C4	0.1	<1	~0	0.002	<1	~0
		Final $[NO_2]$	**16.75 ppm**		**Final $[NO_2]$**	**8.75 ppm**

Note that total NO_x for both plants is 25 ppm.

The conversion of NO to NO_2 due to the presence of higher hydrocarbons also affects post combustion, selective catalytic reduction (SCR) methods of NO_x removal. SCRs use ammonia and reduction catalysts to remove NO_x from the exhaust stream of combustion turbines. This process is most efficient when the ratio of NO_2 to NO is approximately 1:1. The production of greater amounts of NO_2 in the exhaust stream will reduce the efficiency of an SCR, increasing the need for additional ammonia injection and subsequently increasing the amount of ammonia escaping in the exhaust (known as “ammonia slip”).

Bottom Line

Several key combustion properties are altered as the composition of the natural gas changes. Even for constant Wobbe Index fuels, the fundamental differences in burning velocity and autoignition properties of different hydrocarbons can potentially have significant effects on the operation of DLN combustion turbines. Upon installation, DLN combustion turbines are ‘tuned’ by the OEMs or other specialists to achieve the best performance in terms of emissions and combustion dynamics. As described in Chapter 5, tuning involves subtle changes in air and fuel flow splits between different nozzles and inlets. This tuning is necessary for a number of reasons, including changes in natural gas composition that alter the burning velocity (flame speed) or the location of the flame zone within the combustor. Slight movement of the flame zone can have significant effects on emissions and combustion dynamics. Also, the quality and composition of the fuel impacts turbine life and emissions levels.

The need to tune individual combustion turbines indicates the requirement for a consistent natural gas composition. DLN machines, with a narrow stable operating range, cannot handle significant fluctuations in natural gas composition over a short period of time while maintaining their very low pollutant emissions. Even a significant one-time change in composition will require re-tuning of the machine or an expensive hardware change. This conclusion is understandable from basic combustion science and observed anecdotally by combustion turbine users.

Anecdotally, there have been numerous operational issues that have been attributed to natural gas composition changes. For example, an industrial plant operating DLN combustion turbines in Germany had access to both Russian pipeline gas and North Sea natural gas. The machine had been tuned to run on the North Sea gas, which had fairly large C_2 and C_3 content with approximately 82% methane. The Russian gas was mostly methane (over 95%). In attempting to switch from the North Sea gas to the Russian gas, the combustion dynamics were so bad that the combustion turbine automatically shut down to prevent severe damage to the rotating components. In this case, switching between the natural gas sources required retuning of the combustor.

The number of occurrences of operation problems due to natural gas variability appears to be increasing. The owners and operators of the combustion turbines also echo this conclusion at various combustion turbine user group meetings. However, the number of well documented cases that have been made available to the public are sparse.

The results presented above show that the Wobbe Index alone is not sufficient to guarantee interchangeability of natural gases for modern DLN combustion turbines. The complex combustion systems of these machines are designed to produce very low emissions while maintaining acceptable hardware lifetimes. As shown above, there are several variables important to stable operation of these combustion turbines, such as burning velocity and air to fuel ratio, that are not captured by Wobbe Index. Therefore, additional parameters must be used in order to assure that future natural gases of varying composition meet the fuel requirements of modern combustion turbines.

Such parameters might have broadly acceptable values nationally, but more restrictive values within a given region to assure that frequent retuning of combustion turbines is not required. It is important to note that stakeholders throughout the industry generally recognize that Wobbe alone is not sufficient to demonstrate interchangeability. Studies involving residential appliances typically conclude that additional combustion performance parameter(s) are recommended to adequately describe fuel gas interchangeability. Unfortunately, combustion testing involving combustion turbines is typically not included in these studies.

5
FUEL CLEANLINESS ISSUES AND CONDITIONING OPTIONS

Gas Cleanliness and Quality Issues

The fuel specified for a given application is usually based on availability and price. The composition of fuel gas can vary widely in quality, from "pipeline-quality" gas consisting mostly of methane to fuel gas with significant amounts of non-combustible gases, such as nitrogen or carbon dioxide, through gas with significant amounts of heavier hydrocarbons (butane and heavier). Gas fuel quality and cleanliness issues that affect the gas turbine operation are:

- Variation in fuel composition and heating value,
- Flashback and autoignition caused by hydrocarbon liquids,
- Particulates that lead to erosion and plugging,
- Water in the gas that produce hydrates and acids.

Variations in Heating Value

Variation in the heating value as a result of gas phase composition variation affect gas turbine emissions, output and combustor stability. For conventional diffusion flame combustion systems, changes greater than 10% require gas control hardware modifications. For modern dry low NO_x combustion systems, changes greater than 5% may require gas combustion system hardware modifications.

As discussed in Chapter 3, local distribution companies use propane/air injection as a “peak shaving” practice for stabilizing variations in heating value. Since the quantity of air injected is well below the rich flammability limit of the gas, this practice does not pose a safety issue.

Variations in heating value are a particular concern if gas is purchased from a variety of sources based on daily or weekly variations in gas price. In this situation, the user should ensure that the variations are within the values prescribed by the original equipment manufacturer (OEM) fuel specification. On-line instruments that determine and monitor heating value are available from several suppliers and should be used if significant variations are expected. Slugging of hydrocarbon liquids affects the energy delivered to the turbine and can result in significant control problems and potential hardware damage. For this and other reasons described below, all liquids must be eliminated from the gas supplied to the turbine.

Autoignition of Hydrocarbon Liquids

As gas quality has decreased in recent years, removal of liquids has become of great concern to gas turbine owners and operators. Liquids are formed from the condensable higher hydrocarbons found in natural gas, generally those higher than pentane (C5), as well as moisture from water vapor. Moisture is undesirable because it can combine with methane and other hydrocarbons to generate solids in the form of hydrates. Hydrate formation and prevention is discussed later in this chapter.

Hydrocarbon liquids are a much more serious issue because liquids can condense and collect over long periods of time, then result in liquid slugging as gas flow rates are increased after a period of reduced power operation. Since turbine fuel systems meter the fuel based on it being a gas, heavy hydrocarbon gases that are present as liquids provide many times the heating value per unit volume than they would as a gas. This creates a safety problem, especially during the engine start-up sequence when the supply line to the turbine may be cold and these heavy hydrocarbons occur as liquid. This can lead to:

- Uncontrolled heat addition
- Autoignition at compressor discharge temperature
  (625°F to 825°F, 329°C to 451°C range)
- Potential for promoting flashback and secondary/quaternary re-ignitions,
- Varnish-like deposits

If sufficient quantities are present, carry-over of liquids to the turbine can result in possible damage to the hot gas path. A more common problem, however, is with the exposure of small quantities of hydrocarbon liquids to compressor discharge air. DLN combustion systems require pre-mixing of gas fuel and compressor discharge air in order to produce a uniform fuel/air mixture and to minimize locally fuel-rich NOx-producing regions in the combustor. Typical autoignition temperatures (AIT) , the temperatures required for spontaneous combustion with no ignition source, for these liquids are in the 400°F to 550°F (204°C to 288°C) range and fall below compressor discharge temperature. Exposure to compressor discharge air above the AIT will result in instantaneous ignition of the liquid droplets, causing, in some cases, premature ignition of the pre-mixed gases, often called "flashback. For these reasons, fuel gas must be kept above its dew point, and for fuel gas containing heavier hydrocarbons or water this may require additional heating of the gas. As described below to prevent condensation in the gas fuel manifolds, which is caused by gas expansion through the control valves, OEM fuel specifications require a minimum of 28°C, 50°F of superheat at the turbine speed ratio valve inlet flange. This value provides a degree of safety and is within the ASME recommended values for dry gas fuel.

Particulates in the Gas Stream

Operating issues with particulates in the gas stream include fuel nozzle plugging, erosion and deposition. Of the three, fuel nozzle plugging has the most impact on normal operation. Since DLN fuel nozzles have smaller passages than those used in diffusion flame combustors, they are more prone to plugging. Fuel nozzle plugging will result in increased emissions and exhaust temperature spreads. Also plugging can cause autoignition and flashback.

High levels of particulate in fuel gas can erode fuel nozzle metering orifices and gas control valves. For this reason, OEM fuel specifications typically call for removal of particulates greater in size than approximately 10 microns to prevent erosion. Particles that are smaller than about 10 microns tend to follow the gas stream resulting in reduced erosion rates.

Along with fuel nozzles and control valves, particulates can affect turbine life. Deposition on turbine vanes and blades can be problematic even with particles less than 10 microns in diameter. Hence, OEM fuel specifications typically limit the concentration of particulates from all sources and sizes to no more than 600 ppb at the first stage vane inlet.

Formation of Gas Hydrates

Water occurring as liquid in the fuel gas may combine with other constituents to produce hydrates or, in the presence of H_2S or CO_2, form acids that can attack fuel supply system components. Gas hydrates are crystalline materials that are formed when excess water is present in a high-pressure gas line. These solids are formed when the gas temperature is below the equilibrium hydrate formation temperature. Formation temperatures can be significantly above 32°F, 0°C at pipeline delivery pressures. Hydrate formation varies with water content. Hydrates deposit in stagnant areas upstream and downstream from orifice plates, valves, tee sections and instrumentation lines, causing plugging and lack of process control.

To avoid equipment blockage problems, pipeline companies typically limit water to a nominal value of between 4 and 7 lbs per million standard cubic feet (64.1 to 112.1 kg per million standard cubic meters). It is removed to this level by chemical scrubbing with methanol or ethylene glycol. Occasionally, a process upset may occur and spillover of inhibitors into the gas supply can present a hazard by raising the hydrocarbon dew point. Preventative measures include gas heating upstream from pressure-reducing stations to maintain the gas temperature above the hydrate formation temperature.

Formation and Carry-Over of Liquids: Hydrocarbons and Moisture

As the gas fuel is brought to the gas turbine, it often passes through a series of pressure-reducing stations before it enters the gas manifolding system. When a gas experiences a drop in pressure (e.g., due to a flow orifice or control valve) and no heat or work is exchanged with the environment, the temperature of the gas will actually drop for most hydrocarbon gases. This behavior is called the Joule-Thompson effect. Since any fuel system will cause a pressure drop, even a gas that is above the dew point at the system inlet could experience liquid dropout due to the fall in temperature (Figure 5-1). The situation may be aggravated in winter when the surrounding temperature is below the fuel supply temperature.

Hydrocarbon dropout will vary depending on the gas composition as shown in Figure 5-1 for LNG, pipeline quality gas and unprocessed natural gas. Also, temperature reductions for a typical natural gas in winter and summer are illustrated in Figure 5-1.

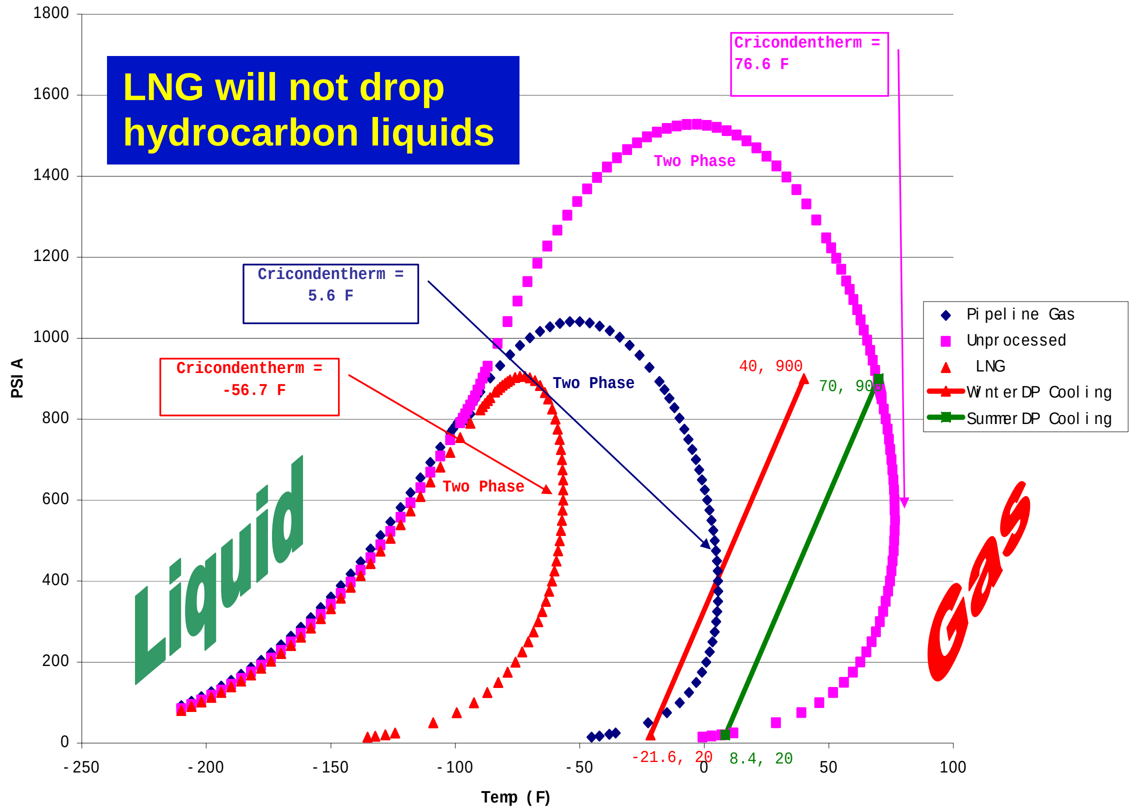

Figure 5-1
Fuel Gas Dew Point Curve [Source: Siemens-Westinghouse Power Corporation]

For F-class technology machines, gas entering the site at 900 psia, 6.205 MPa and 60°F, 16°C can experience a temperature reduction of 31°F, 17°C prior to entering the gas module at the maximum allowable pressure of 450 psia, 3.102 MPa. As can be seen in Figure 5-1, further temperature reductions experienced as the gas passes through the control valves could cause hydrocarbon dropout for typical pipeline natural gas. This concern will be the greatest at low load when control valve throttling is at the highest level.

Reports of frost appearing on the outside of the gas piping downstream from the control valves are relatively common, but is not a cause for alarm provided the hydrocarbon and moisture dew points are significantly less than the local gas temperature.

OEM fuel specifications typically call for superheating fuel gas a minimum of 50°F, 28°C above the hydrocarbon dew point at the entry to the gas turbine at all operating conditions. Superheating is required because separation of the free liquids alone results in a saturated gas with a dew point equal to the gas temperature. Further reductions in temperature down-stream from the separation equipment will, therefore, result in immediate condensation and formation of additional liquids. For incoming wet gas, a filter separator and a superheater are essential to prevent the formation of liquids.

Clean-up Equipment Evaluation and Sizing

In addition to individual equipment size and removal capability, the overall solids and liquids removal system should be assessed when evaluating gas clean-up equipment. For example, if liquid separation equipment is required, including a coalescing filter will accomplish solids removal as well.

Only a particulate removal filtration system may be required, if the gas is known to be dry and meets the 50°F, 28°C minimum superheat requirement and no liquids removal equipment is installed (e.g., some LNG meet this requirement).

Particulate Removal System

A filter system designed with an absolute removal rating of 3 microns or less is recommended. This equipment is normally available in a vertical configuration and consists of a series of changeable, parallel filter elements attached to a tube sheet.

For peaking units, one filter vessel is acceptable. For base loaded units, two units configured in a duplex arrangement are recommended. The duplexed arrangement permits isolation of one vessel for maintenance while the other is in operation. Using the bypass line for maintenance purposes is not an acceptable alternative.

Equipment size is based on dirt holding capacity and allowable pressure drop, which determine the number of filter elements. Filter elements must meet the maximum gas temperature requirements, if the gas is to be heated prior to filtration.

The strainer in the inlet supply pipe is permanently installed (removable for cleaning purposes) and protects the fuel nozzles and acts as a flag to indicate non-compliance with OEM fuel gas specifications. A well-designed filtration system will prevent particulate build-up on the strainer once the initial dirt and other contaminants have been removed from the system.

Liquids Removal System

A properly designed liquids removal system should include the following sequence of individual equipment:

1. Pressure-reducing station,
2. Dry scrubber,
3. Filter/Separator,
4. Superheater.

If the inlet gas pressure is above approximately 1000 psia, 6.895 MPa, or if the gas has high moisture content, an additional heater may be required upstream from the pressure reducing station. Also, the section upstream of the pressure-reducing valve may require heating to avoid the formation of hydrates and slugging of condensed hydrocarbons that would otherwise remain in the gas phase throughout the liquids removal process. Typically, this heater will not provide sufficient energy to meet the 50°F, 28°C minimum superheat requirement at the gas control module inlet.

For gas turbine applications heaters upstream of scrubbers and filter/separators are not acceptable. Heating the fuel upstream from a separator could raise the gas temperature above the dew point, and little or no liquids will be removed.

First Steps

The first step to ensure that the correct equipment is specified for a given gas fuel, is to obtain an accurate gas sample and analysis from the site taken upstream and prior to any fuel gas treatment equipment, and also at the combustion gas turbine fuel gas manifold.

The second step is to evaluate equipment location, sizing and efficiency requirement based on the fuel gas analyses obtained. It is important to note that equipment efficiency will diminish with reduction throughout. Therefore, the design point of inertial separation equipment should be selected at 5% to 10% below the maximum expected flow rate. Most inertial separators will maintain high efficiency up to 10% above the design flow rate; check with the supplier for details.

Equipment should be located as close as possible to the combustion gas turbine. This is especially true of the superheater since liquids can condense in the line downstream from the heater after the unit has shut down -- the shorter the line, the lower the volume of condensates.

Processing Equipment Description

A brief description and simplified sketches of the various types of clean-up equipment follows.

Dry Scrubbers

Dry scrubbers are inertial separators that employ multiple-cyclones (multi-clones) to remove liquid and solid materials without the use of scrubbing fluids. Figure 5-2 shows a typical cross-section of a dry scrubber. Except for blowdown of the drain tank, dry scrubbers are effectively maintenance-free.

Since multi-clone scrubbers are limited to a 4: 1 turndown in volumetric flow rate, only one gas turbine should be placed downstream of each dry scrubber. Combining dry scrubbers with coalescing filters can provide protection over the entire operating range of the gas turbine.

Occasionally, a dry scrubber may be installed to protect a pressure-reducing station serving multiple gas turbines. In such cases, liquid carry-over can be expected when the gas demand is low.

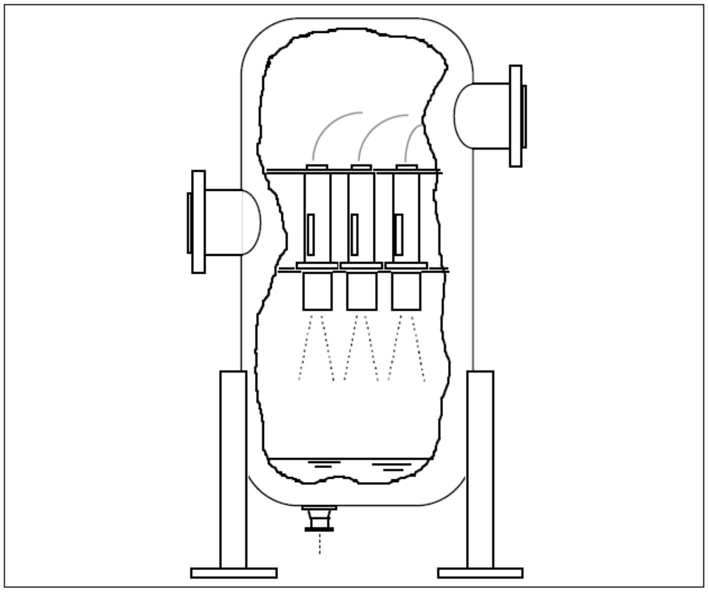

Figure 5-2
A Vertical Multi-clone Dry Scrubber [Source: General Electric Company]

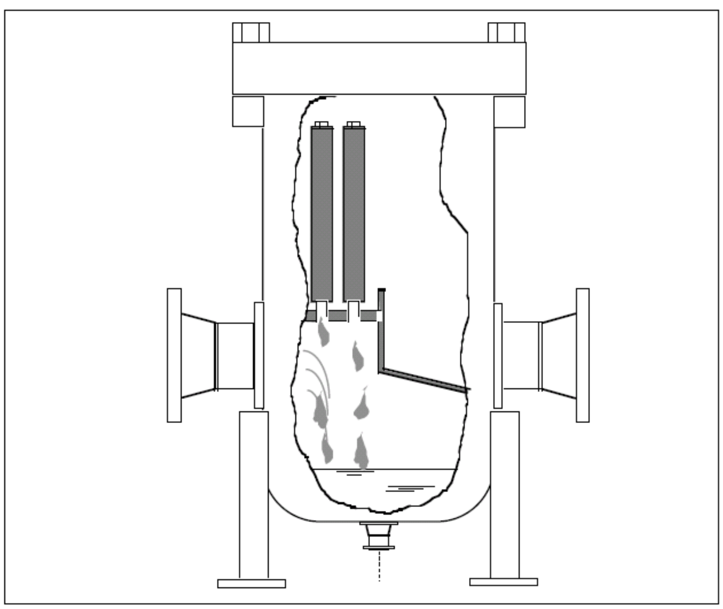

Figure 5-3
Coalescing Filter [Source: General Electric Company]

Coalescing Filter

Where high removal rates of liquid droplets are required, coalescing filters are used in conjunction with a dry scrubber. Coalescing filters will remove all droplets and solids larger than about 0.3 microns. Figure 5-3 shows a sectional view of a coalescing filter. Typically, the filter unit consists of a vertical pressure vessel that contains a baffle and a number of parallel tubular filter cartridges. Wet inlet gas flows through the filter tube where fine droplets collide with the fibrous filter material. The droplets coalesce with others to form larger droplets that are then removed from the filter element by gravity and collected in the sump. If operated beyond its design flow rate, the efficiency of this device will drop off dramatically.

Combination Separators

As illustrated in Figure 5-4, the filter separator combines changeable filter elements and vane mist eliminator into a single vessel. Wet gas first passes through the filter elements, enabling smaller liquid particles to coalesce while the solids are removed. The coalescing effect allows the vane mist eliminator to remove more free liquid particles than a dry scrubber alone. This combines the efficiency of the vane separator with that of the coalescing filter in one vessel. The filter separator maintains its guaranteed separation efficiency from 0% to 100% of its design flow capacity, as does the coalescing filter described earlier. Filter separators are often used in lieu of filters when high liquid rates are expected.

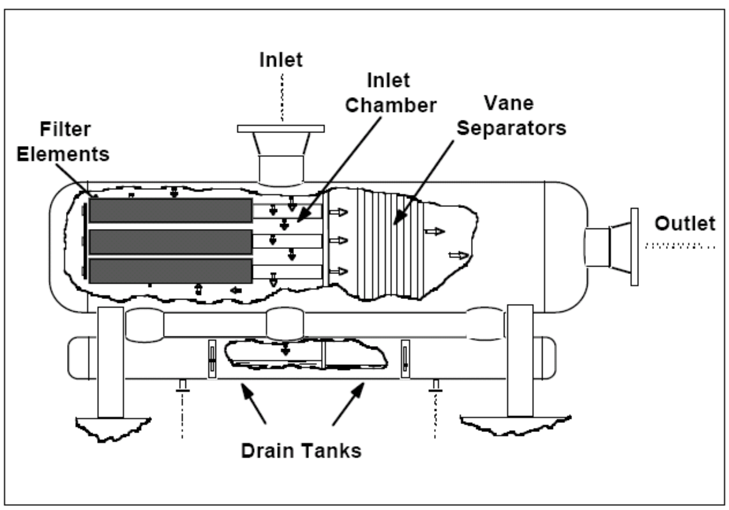

Figure 5-4
Combined Filter-Separator [Source: General Electric Company]

Fuel Heating

OEM fuel specifications require fuel heating to raise the temperature of the gas to 50°F, 28°C above the hydrocarbon dew point. The three types of heater used in gas turbine applications include electric heaters, indirect gas- or oil-fired heaters and waste heat-fired heaters.

Electric Heater

The electric heater is simple in construction and operation making it the most convenient type of fuel heater to use and install. Figure 5-5 shows a sectional view of an electric heater. A simple control system can maintain a constant exit temperature or a constant temperature rise within the capacity limits of the equipment as fuel flow rate varies.

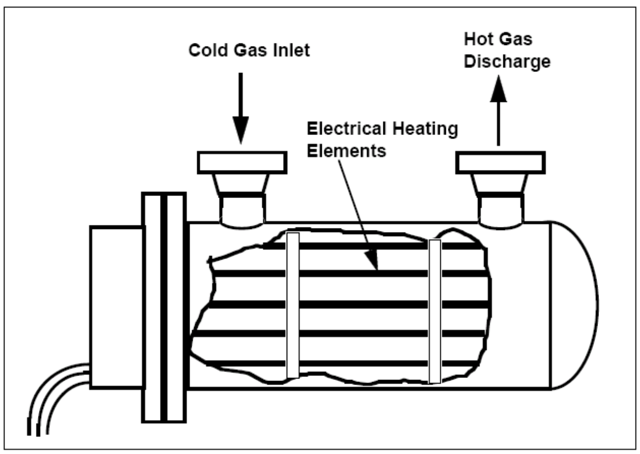

Figure 5-5
Electrical Gas Heater [Source: General Electric Company]

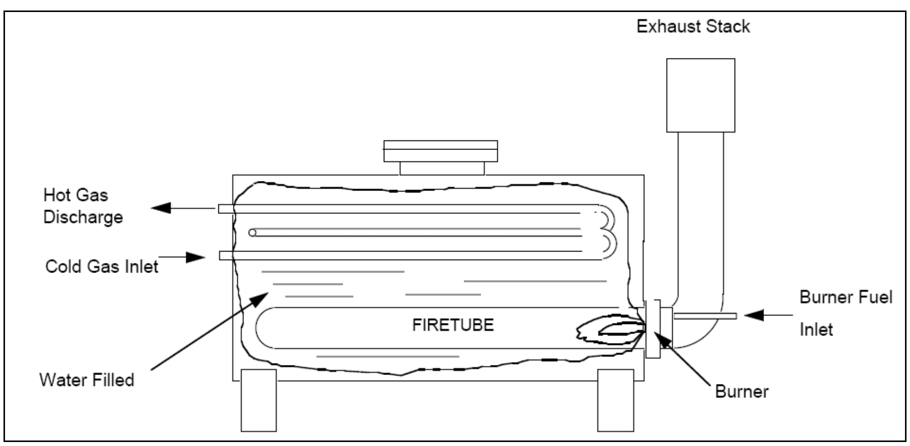

Figure 5-6
Indirect-Fired Gas Heater [Source: General Electric Company]

The overall energy efficiency of an electric heater is approximately one-half, or less than that of gas- or oil-fired heaters but maintenance cost are low. Heating elements can be easily replaced and no intermediate heat transfer fluid is required, a concern in freezing climates, which reduces maintenance costs.

Indirect Gas- or Oil-Fired Heaters

Indirect gas- or oil-fired heaters are used worldwide. Figure 5-6 shows a sectional view of this type of heater. An intermediate heat transfer fluid is generally used for safety purposes. This type of heater can have difficulty tracking rapid fuel demand changes of the gas turbine and may be an issue for peaking and load following units or during startup. Larger foundations are required for this type of heater, and several burners may be required in order to provide improved thermal response and turndown capabilities. Operating costs are significantly lower than an electrical heater, but maintenance and capital costs are higher.

Waste-Heat-Fired Fuel Heaters

This is an option best suited to combined-cycle units where low-grade heat (hot water) may be readily available. The advantage of this type of heater is that no fuel penalty is incurred and the overall thermal efficiency of the power plant may be increased. Disadvantages are higher capital cost, increased maintenance and installation costs for larger foundations and lack of heating during startup. Construction is of the tube and shell type and is heavier than the indirect-fired heater to accommodate the 400+ psia, 2.758+ MPa pressurized water supply. A typical shell and tube heater is shown in Figure 5-7.

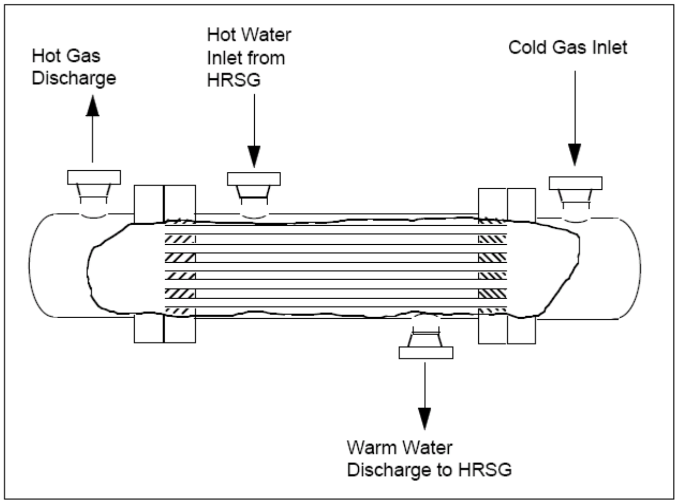

Figure 5-7
Waste-Heat Gas Fuel Heater [Source: General Electric Company]

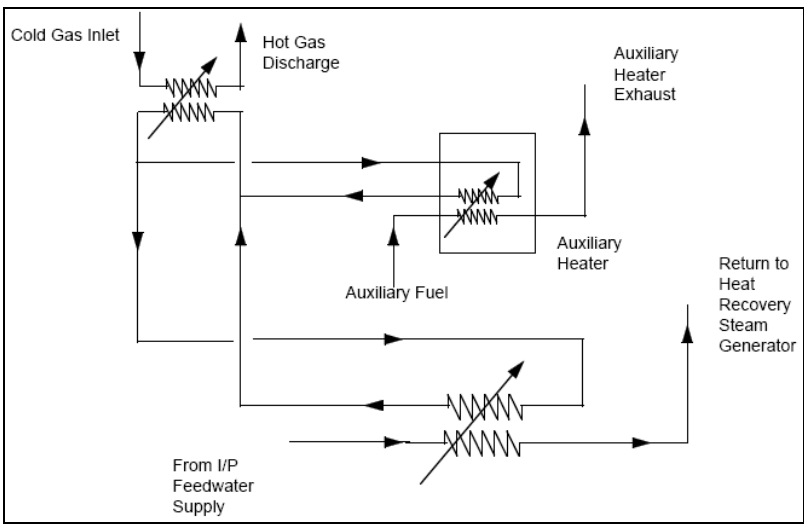

Figure 5-8
Simplified Schematic for a
Dual-source Gas Fuel Heater [Source: General Electric Company]

Dual-Source Heaters

Dual-source heaters are similar to the waste- heat-fired heater but can also be fired using a remote gas burner. This ensures that the gas is completely free of liquids during all phases of the gas turbine operation including simple-cycle mode and startup.
Figure 5-8 shows a simplified schematic of this type of heater.

Equipment Arrangement

For sites where the specific quality of the gas is unknown, a vertical gas separator followed by either duplex multi-tube filters or filter separator and superheater is recommended. Each of the duplex units must be designed for 100% of the system flow rate so that one can stay on-line while maintenance is being performed on the other. Four gas conditioning systems are described below to illustrate the simplest to the most complex skid package typically found in the field. For dry gas with ample superheat, a simplex or duplex particulate filter, as shown in Figure 5-9, is all that may be required. Burning LNG at a site where the supplier has guaranteed no hydrocarbons higher than C5 and where the gas temperature delivered to the site is well above the hydrocarbon dew point is an example of this kind of applications. A gas with a moisture and hydrocarbon dew point of less than -50°F, -46°C and a gas delivery temperature of about 55°F, 13°C would meet this description.

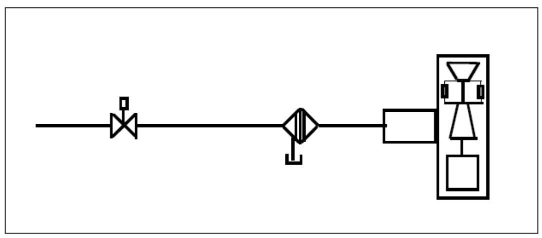

Figure 5-9
Simple Particulate Filtration used for Dry Gas [Source: General Electric Company]

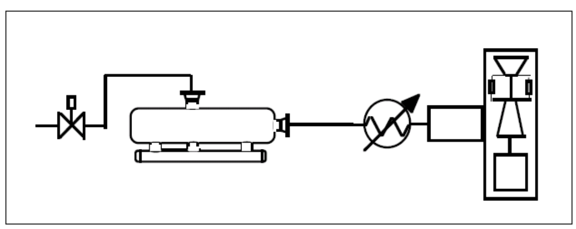

Figure 5-10
For Wet Gas with Non-slugging Conditions Upstream from Pressure Reducing Station [Source: General Electric Company]

However, allowance should be made for temperature drop through the pressure-reducing station. A superheat temperature of 105°F, 35°C would prevent liquid condensation.

Regardless of the quality of the gas, there is a need for particulate removal, since particulates can be generated by spallation of rust and other corrosion products within the pipeline. Stainless steel piping is required downstream from the particulate filter.

For wet gas but without excessive liquids and no slugging potential upstream from the pressure-reducing station, single- or duplexed-filter/separators are recommended, followed by a heater that will provide a minimum of 50°F, 28°C of superheat. Figure 5-10 shows this arrangement with a single filter/separator.

A pressure drop through the pressure- reducing valve greater than about 300 psi, 2.068 MPa could cause slugging downstream. In this situation, dry scrubber upstream from the filter separator may be required depending on the manufacturer's recommendations. Figure 5-11 shows this arrangement.

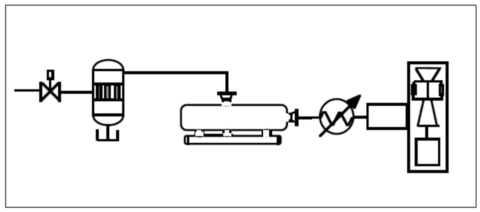

Figure 5-11
Dry Scrubber installed to Protect Filter Separator against Excessive Slugging Conditions [Source: General Electric Company]

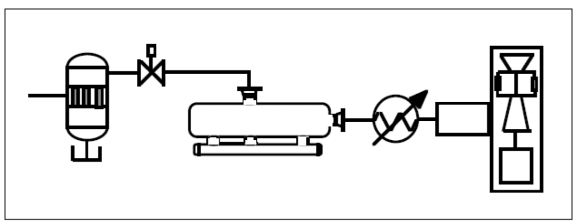

Figure 5-12
Incoming Wet Gas with Slugging Potential Upstream from Pressure Reducing Station [Source: General Electric Company]

Wet gas with slugging present may require the addition of a dry scrubber upstream of the pressure reducing station. Figure 5-12 illustrates this arrangement. Also, filter separator is required to provide protection over 100% of the flow range and to minimize any liquid carry-over to the heater.

A dry scrubber and heater may be required upstream from the pressure-reducing station, if the incoming gas has a potential for hydrate formation. A filter/separator and superheater are required as before. The heat input can be minimized upstream to a level that avoids hydrate formation while allowing the downstream filter/separator to remove liquids by physical separation.

Depending on gas composition and moisture content, the hydrate formation temperature may be above or below the hydrocarbon dew point temperature. If it is above the hydrocarbon dew point, then a re-arrangement of equipment may be beneficial to avoid installation of two heaters. A minimum superheat temperature of 50°F, 28°C must be maintained at the gas turbine module inlet.

If multiple units are present on-site, a common clean-up system is often used to protect the pressure-reducing station, but individual filter/separators and heaters must then be installed downstream to protect each unit.

A typical gas compression system used where the incoming gas supply pressure is too low to meet the OEM pressure requirements is shown in Figure 5-13 shows. In this case, advantage can be taken of the heat of compression to avoid the cost of a superheater. The heat of compression added to the gas stream is normally much greater than the 50°F, 28°C minimum requirement.

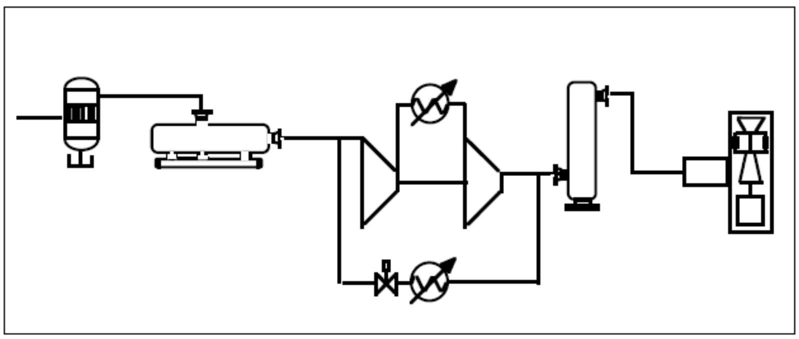

Figure 5-13
Two-Stage Gas Compressor providing more than 50°F, 28°C Superheat [Source: General Electric Company]

To avoid potential spill-over of compressor lubricating oil, installation of a coalescing filter or absolute separator should be included downstream of the gas compressor. If the heat loss in the gas line to the turbine is excessive, then a coalescing filter and superheater may be required downstream from the gas compressor to regain the 50°F, 28°C superheat.

The gas clean-up systems described here are only examples. The specific needs of an individual site must be carefully assessed, and the equipment and system evaluated accordingly. It is not sufficient, however, to independently select equipment based on claimed high efficiency alone; the entire system must be evaluated and preferably modeled to determine the overall system sensitivity to changes in gas composition, pressure, temperature, and mass flow rate.

6
DLN MONITORING AND TUNING APPROACHES

Introduction

Tuning of Dry Low NO_x (DLN) combustion systems may be required for several reasons. The first reason is to follow the OEM guidelines that combustion tuning be performed subsequent to any change out of combustion hardware. Because of the high sensitivity of these systems to changes in fuel/air ratio, subtle changes in the liners or caps (which effect air flow) or fuel nozzles (which effect fuel flow) can significantly impact the emissions, dynamics, or stability of the combustion system. Tuning after a hardware change out is a quality check to ensure that the combustor is still operating with sufficient emissions margin while also optimizing the dynamics and lean blow out margin of the combustor.

A second reason for performing tuning is to monitor and optimize the combustion dynamics as they change in response to seasonal ambient temperature variations. Combustion dynamic levels of DLN combustion systems increase with colder ambient temperatures. Some operators will routinely perform tuning in winter to minimize the increased dynamic levels as much as possible while still enabling the unit to meet their site emissions limits.

A third reason for performing tuning is to troubleshoot problems that arise during the course of normal operation as a result of fuel composition changes. Such problems include emissions excursions outside permitted site levels, flame out of the combustion system, or dynamics excursions beyond recommended amplitude levels. The last effect is only discernable on units that have permanent dynamic monitoring systems installed. Depending on the nature of the problem, tuning may or may not be sufficient to resolve the issue. For instance, in cases of damaged hardware that result in excessive emissions or dynamics levels, tuning may only be able to diagnose that a more serious problem exists. On the other hand, for excursions that arise from the changes in fuel composition, or fuel metering and air flow changes that occur as a unit ages – tuning may easily resolve the problem.

As indicated, changes in fuel composition, combustion system hardware and/or ambient air temperature can lead to the onset of combustion instabilities, pressure pulsations, and resonant acoustics, also referred to as humming. Large pressure pulsations can cause degradation and catastrophic failure of hot section components and loss of peak power production. Without immediate operator’s intervention and periodic combustion tuning, dynamics in DLN combustors can result in emergency shutdown with significantly lengthier and costlier repair down times.

In response to these operational risks, EPRI has undertaken a program to document the equipment and methods for monitoring and controlling the onset of combustion dynamics and to map the operating regime for stable combustion in modern DLN combustors.

EPRI has produced a series of reports on the subject. The first report was released in March 2004, *Product ID 1005036, Guidelines for Combustor Dynamic Pressure Monitoring,* which describes available dynamics monitoring and data analyses approaches.

The second report in the series, Product ID 1005037, Tuning Approaches for DLN Combustor Performance and Reliability, March 2005, describes the combustion tuning techniques employed for power generation combustion turbines to mitigate the adverse effects of combustion dynamics. This report includes combustion tuning practices and guidelines for General Electric, Siemens-Westinghouse, Mitsubishi and Alstom DLN combustion turbines.

The third report in the series, *Product ID 1010479, General Electric DLN-2.6 Operational Description and Tuning Guideline,* available March 2006, describes the detailed, model-specific procedures to perform self-tuning of the General Electric DLN-2.6 combustion system. Follow-on reports in 2006 will focus on analyzing CDM data to provide specific operating guidelines applicable to the development of predictive maintenance tools, and for improved operational flexibility at low load and part load operating conditions.

Combustion Dynamics

Combustion instabilities, sometimes referred to as pulsations, humming (low frequency), screech (high frequency) or dynamics are considered to be the one of the major problems that the gas-turbine industry faces today. Rapid oscillations in the rate of heat release can occur when the premixed fuel mixture approaches its flammability limit (the point when stable combustion is no longer self-sustaining). For a premixed methane/air mixture with self-ignition, the lean blow out (LBO) typically corresponds to an equivalence ratio of 0.5 and is only weakly dependent on pressure. Therefore, as equivalence ratios are lowered in the quest for greater NO_x reduction, the potential for lean blowoff and onset of combustion dynamics increases. As the temperature is increased, the flammability range widens, and at a sufficiently high temperature, self-ignition of fuel occurs. Thus, when the combustion air temperature is lowered, say with a drop in ambient temperature, the mixture becomes less flammable and more likely to become unstable, especially at low equivalence ratios.

Ignition time also increases exponentially with equivalence ratio. At extremely low equivalence ratio, even small changes in air or fuel flows can induce a sequence of rapid changes in ignition time, affecting the heat release rate. Externally induced pressure pulsation can also initiate or add to pressure fluctuations in the combustion chamber. For example, oscillations in fuel delivery pressure to the combustor will create heat release fluctuations, which can translate to further changes in fuel supply pressure at the nozzle. Additionally, small changes in natural gas composition will also affect fuel kinetics and heating value with a resultant change in heat release rate and flame temperature. Unsteady heat release from slugs of hydrocarbon liquids in fuel gas can cause large amplitude pressure pulsations in the combustor. Reductions in ambient temperatures also lowers fuel/air ratio and heat release rate affecting combustion stability at already low equivalence ratios. In fact, significant dynamics in selected combustors has occurred during large reductions in ambient temperatures bringing fuel equivalence ratios to the zone of instabilities. These are then exacerbated by pressure pulsations in the chamber, which affect the flow of fuel.

Impact of Combustion Pulsations

Combustion instabilities deteriorate the engine performance and reliability, and increase the frequency of required maintenance. Combustion instability can cause flashback, flame blowout due to an increase in the LBO limit mixture ratio, starting problems, damage to combustor hardware, switchover problems, High Cycle Fatigue (HCF) of hot gas path components, and Foreign Object Damage (FOD) to turbine components. In the worst case, system failure can occur due to extensive structural damage and loss of control. Because of increased awareness of the sources and effects of combustion dynamics (CD), many of the early catastrophic failures experienced with early DLN designs have been largely reduced in intensity and frequency. In fact, considerable effort has been invested by gas-turbine companies, academia, and government agencies to study the mechanisms that drive these instabilities, identifying the conditions under which they occur, and developing practical approaches for their monitoring and control, either passively or actively.

Figure 6-1 illustrates damage to one of the five nozzles of a DLN-2.0 combustor that resulted from flashback episode following periods of combustion humming. The damage to the pilot nozzles caused fuel to leak from the side of the nozzle. The presence of the added diffusion flames during the fully premixed operating mode of the DLN 2.0 was detected by an increase in NO_x emissions. The DLN-2.0 combustion system was developed specifically for the F technology machines and has been widely applied to the General Electric 6FA, 7FA (early versions), and 9FA combustion turbines.

Figure 6-1
Nozzle Tip Burnout due to Flashback

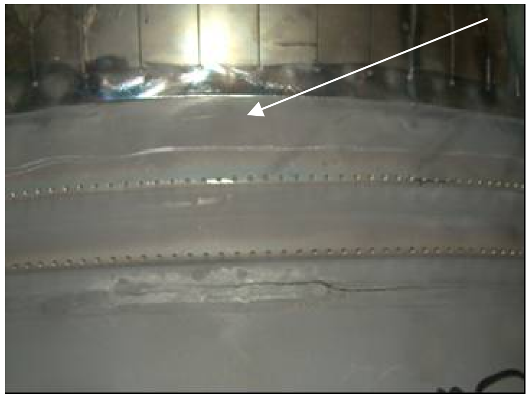

Figure 6-2
Impact of Combustion Pulsations

The damage from combustion dynamics can easily extend to other high-temperature components. Figure 6-2 illustrates the damage on a cracked liner, the result of transaxial low frequency (<200 Hz) vibrations.

The location of the initial failure is often dependent on the resonant characteristics of the particular hardware. Once CD-induced damage occurs to combustors and transition pieces, the resulting debris will create havoc to turbine blades. Figures 6-3 and 6-4 illustrate the varying degrees of severe vibration damage to transition pieces and damage to first and second stage blades of an affected S/W 501F engine.

Figure 6-3
Complete Failure of Transition Piece and Evidence of Excessive Metal Temperature

Figure 6-4
First and Second Stage Blade Damage from CD-Induced Failure of S/W 501F Transition Pieces

Because of potential cost impact and other maintenance needs of CD-induced vibrations, all modern DLN-equipped GE, Alstom, and S/W gas turbines come equipped for ready mounting of CD sensors for intermittent measurement and burner tuning. OEMs perform tuning in the commission stage and after a major overhaul.

Online combustion diagnostic is the most important capability of Combustion Dynamic Monitoring Systems (CDMS). From a combination of gas turbine data and CD data, operators can identify serious malfunctioning of the burners and prevent substantial damage of the combustion chamber. For example, the distribution of the exhaust temperatures will change when a particular part of the combustion section deviates from normal behavior. It is even possible to extract from the temperature distribution which combustor or burner section is faulty. Sudden changes in NO_x concentration, coupled with changes in CD frequencies and amplitudes even if they are small, can also be ascribed to fuel composition changes.

CD Data Monitoring

The approach and instrumentation used in monitoring pressure amplitudes and frequencies of dynamic oscillations in each combustor is essentially the same among all OEMs. EPRI report *"Guidelines for Combustion Dynamic Pressure Monitoring", Product ID 1005036, March 2004*, provides a more in-depth review of the information presented in this section.

Table 6-1 lists commercial vendors of electronic systems that process, record, and analyze CD data for online or intermittent diagnostic and analysis. KEMA products Flamebeat™ and MagicCorr™ are tools developed for operators and consultants to display and analyze gas turbine combustion data. Other companies that provide software for combustion diagnosis are GE Bently Nevada, Power Systems Manufacturing (PSM), Control Center, and Alta Solutions, all from the USA. Siemens uses combustion monitoring to operate their Automatic Injection Control-system on annular gas turbines. Both Siemens and Alstom have acoustical or vibrational controlled safeguards on their gas turbines that generate trips if the integrity of components is jeopardized. GE has a tuning system that operates with a dynamic pressure sensor attached to each of the combustors.

Table 6-1
Partial List of Major Commercial Vendors of Combustion Dynamics Data Monitoring Equipment and Diagnostic Services

Vendor	System and Equipment Trade Name	Principal Applications
General Electric	Semi-Continuous Monitoring Systems	Onsite and remote DLN CD monitoring, diagnostic, and tuning services
Siemens	Active Control System (ACS)™	Continuous CD monitoring and diagnostic system with onsite tuning and control capabilities
KEMA	Flamebeat™ and MagicCorr™	CD monitoring, tuning and data management and diagnostic
Power Systems Manufacturers (PSM)	Dynamic/Combustion Acquisition System (DDAQ)™	CD monitoring, tuning, diagnostic and data management coupled with Calpine remote sensing capabilities
Control Center	Uses Alta Solution SpectralMon™ Instrumentation	CD monitoring, tuning, diagnostic and onsite/remote data management. Principally serving S/W and GE equipment
Alta Solutions	AS-250 SpectralMon™	I/O Hardware and computer used by Control Center in SpectraMon™ electronic panels coupled with sensor alarm.

Most CDMS consist of installing:

1. Transfer tubes and sensors
2. Purge and acoustic dampening coil boxes
3. Signal processing instrumentation cart
4. Computer and software

Figures 6-5 and 6-6 illustrate the typical monitoring arrangement supplied by GE and some independent installers servicing GE equipment. Figure 6-7 illustrates the installation of a low-and high-temperature pressure transducer sensor directly on the combustor for greater sensitivity in the signal. Figure 6-8 illustrates the sensor installation on the 501F SWPC combustor. Note that for 501F engines, the signal transfer tubing accesses the combustor liner through the head of combustor and a waveguide extends to the liner. In all CDMS applications, a purge system and a signal dampening system to remove background noise are necessary. Figure 6-9 illustrates the GE junction box and acoustic dampening coil box directly below.

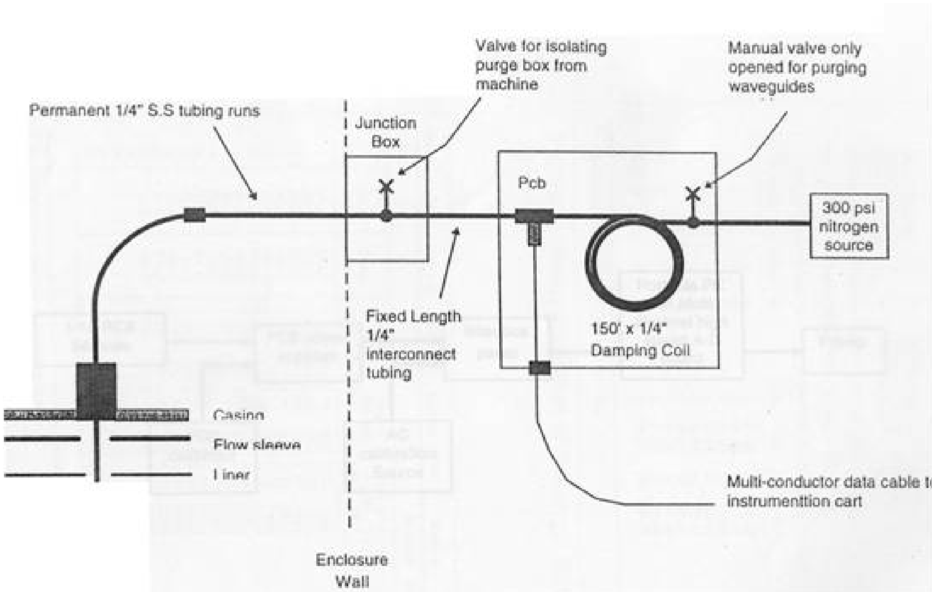

Figure 6-5
Layout of Combustion Dynamics Measurement System (CDMS) with PCB Sensor and Waveguide – GE Installations

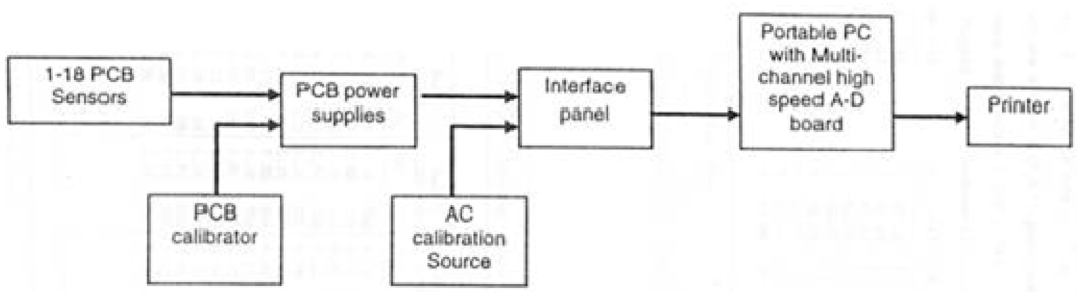

Figure 6-6
Schematic Instrumentation Cart –GE Installations

Figure 6-7
Installation of Transfer Tube for Low-Temperature Sensor (on the left) and Installation of a High-Temperature Sensor on GE Combustors

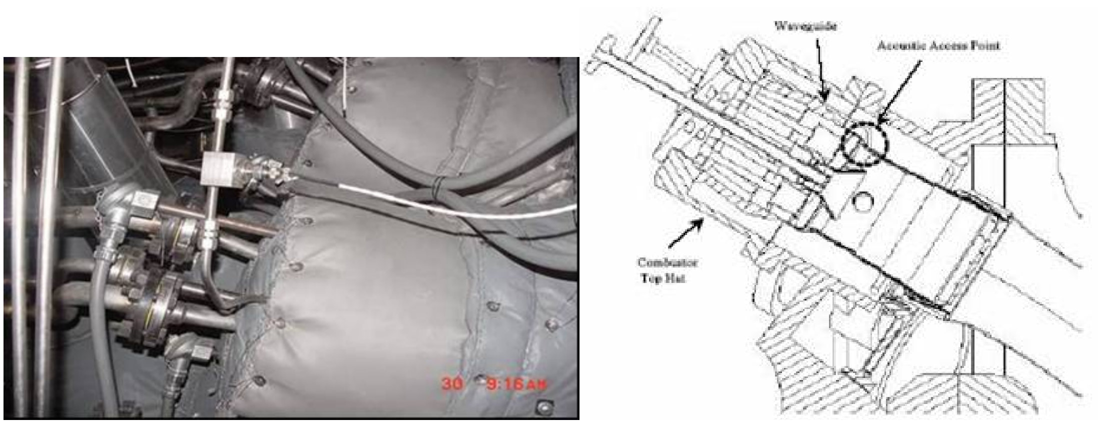

Figure 6-8
CD Sensor Installation on 501F Location of Acoustic Sensor Waveguide [Sewell, 2004]

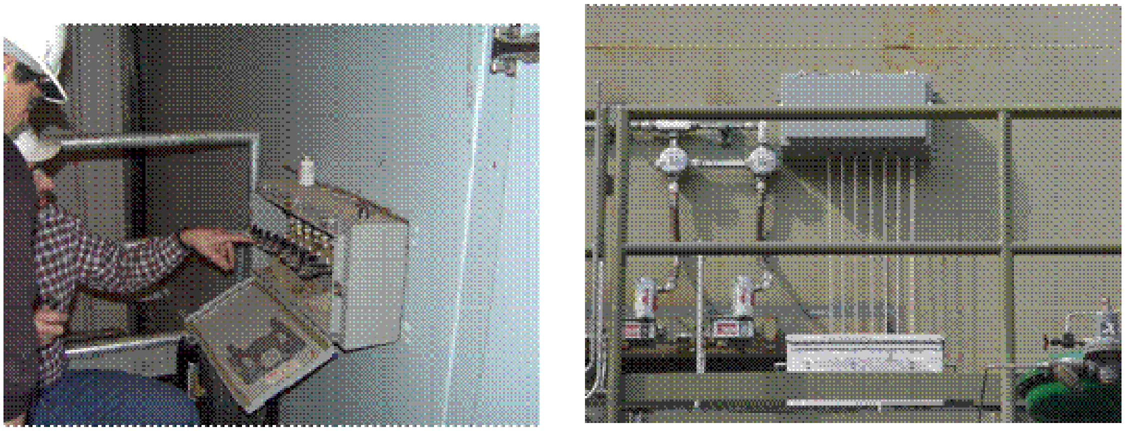

Figure 6-9
Purge Box and Acoustic Dampening Coil Box – GE Installations

Installing the purge boxes requires connecting to a 300 psig, 2.068 MPa nitrogen supply; up to 9 ft length “jumpers” of ¼ inch, 6.35 mm copper tubing; a multi-conductor data cable and a power connection for the box heater. The jumpers make a connection between the pressure transducers and 150 ft dampening coils located in the purge box and the permanent ¼ inch stainless steel tubing connected to each of the combustor cans. The supply pressure for the purge nitrogen is based on the fact that the permanent tubing sees 250 psi, 1.724 MPa compressor discharge pressure. The purge nitrogen removes condensed moisture prior to sample reading. Since liquid accumulation inside the wave guides is unpredictable, it is always highly recommended to periodically purge the system with compressed nitrogen. A sensor that requires purging will normally produce a lower dynamics signal. Another mode of transducer failure is the intermittent shorting. The latter can be easily identified by spikes on the oscilloscope as sudden changes in DC level. Manufacturers can readily provide plots of transducers with good response and those with reduced sensitivity for ease of comparison and problem identification.

The electronic data acquisition system and charge converter are part of the process instrumentation cart. A computer provides the FFT analytical software and data display and analysis capability, including alarm setting. The instrumentation cart connects the cables from the purge boxes and supplies standard 120- or 220-v 60-Hz power. The CDMS signal is calibrated by sending a known AC signal at 100 Hz into the system and checking the reported voltage and frequency.

Some systems provide storage of a long-term dynamics data. These data can be used as an equipment-specific signature of the combustion characteristics of present hardware. In this way a large database of CD spectra is created that can be used to develop long-term trends for the benefit of identifying slow equipment degradation prior to reaching the alarm stage.

Signal Processing and Display

Pressure oscillations occur at discreet frequencies that are associated with the natural acoustic properties of the surrounding enclosure. Once the acoustic signature is profiled, it is then possible to monitor changes in the pressure amplitudes in the frequency ranges that are characteristic of a given combustor design. This can lead to effective monitoring, online diagnostics, and combustion tuning techniques.

Figure 6-10 illustrates typical combustion dynamic signatures of major fleets from GE, Siemens, Alstom and Mitsubishi. The data plots pressure amplitudes versus oscillation frequencies. For example, the recommended frequency ranges for monitoring CD changes in the 7FA DLN-2.0 combustors are: 110-130 Hz; 150-180 Hz; and 2500 Hz. For the 9FA DLN-2.0, the frequency ranges where dynamics pressures show the greatest increase due to combustion instabilities are: 150-170 Hz; 190-220 Hz; and 2350 Hz.

For Siemens/Westinghouse Power Corp (SWPC) 501F DLN, the dominant frequency is around 125 to 140 Hz. On that basis, monitoring is established for selected frequency bands. Alarm limits are set for these bands to indicate excessive pressure amplitudes, as illustrated in the Mitsubishi chart.

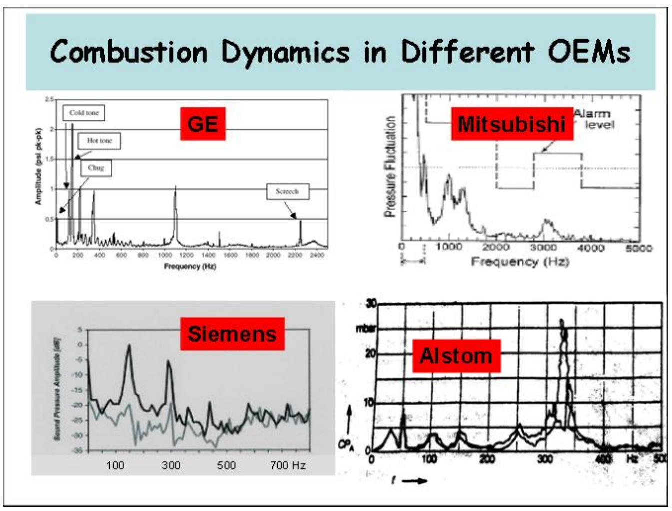

Figure 6-10
Combustion Dynamics Spectra in CTs from Different OEMs

The data display is done in different formats for each of the main CDMS equipment suppliers, although in each case the formats can generally be adjusted and changed over time if required. This is especially the case when an in-depth analysis of the data is attempted, often requiring a comparison of several data sets from the data historian coupled with selected engine supervisory data.

Online Diagnostics

The degree of analyses between these systems is varied, ranging from simply alerting the operator with an alarm when pressure pulsations in any one of the combustors exceed set points (as in the case of Alta Solutions) or more sophisticated data evaluation that permits greater pinpointing capability of potential hardware distress. Some systems integrate CD data with key data from the engine digital control system (DCS) with the intent to display and analyze parameters that describe the condition of the installation, its performance, and the risk of failure of components.

To interpret the spectral data correctly, simultaneously measured gas turbine parameters are merged with the spectra. Selected parameters are extracted from the dynamic pressure spectra such as the amplitude averaged over the full spectrum, and the position and amplitude of the major peaks. As discussed previously, it is important to measure selected frequency ranges, where combustion-induced pressure pulsations occur within a given combustor design. Direct information of the combustion process should be complemented with data from the gas turbine, such as the computed firing temperature, exhaust temperatures, NO_x emission, and load, to confirm the presence of unsafe combustion conditions and to help identify corrective actions.

Permanent combustion monitoring allows operators a regular view of the status of the hot path of the gas turbine. They can use the available controls to minimize combustion dynamics and to avoid trips due to flame loss, initiated by the flame detector due to a fuel composition change. Available control options include de-icing, pilot flame adjustment, load shedding, re-tuning of gas valves, inlet guide vane adjustment within the allowable limits, air- and fuel gas pre-heating. Archived dynamic pressure data can also be used. At a minimum, the operator requires alarm indicators when combustion dynamics change unexpectedly.

The following section provides an example of CD diagnostic capability that led to the implementation of corrective actions and combustor tuning preventing potential hardware damage and safeguarding operation.

Combustion Diagnostics Worksheet

The Combustion Diagnostic Worksheet in Table 6-2 contains a matrix of hardware faults or failures with manifested changes in three performance monitoring areas, namely CDMS, engine supervisory data, and continuous emission monitoring (CEM) system. The following fault parameters are included:

- Off-calibration fuel nozzle,
- Eroded or burned fuel nozzle,
- Clogged fuel nozzle,
- Obstructed air passages,
- Clogged air filter,
- Compressor fouling,
- Cracked combustion liner,
- Cracked transition piece,
- Crossfire tube failure.

Only combustion-related faults are considered in the matrix. The number and corresponding shading in the matrix highlight the most likely impact of each fault on the combustion dynamics frequency signature and the engine emissions. Thus a heavily shaded number “3” indicates that pressure amplitudes will likely change in selected frequency range and changes in either NO_x or CO are also most likely and can be used as a confirmation that engine may have the hardware failure indicated. In the case of engine supervisory data, the numbering system also addresses the relative severity of the recorded impact. Thus a number “3” indicates that if a noticeable change is recorded, then an action is highly recommended.

All the fault condition parameters have one thing in common in that they all affect the local fuel/air ratio to the fuel nozzles. Thus if refurbished parts are not properly calibrated, then some nozzles will be more apt to initiate combustion instabilities that can aggravate over time and ultimately result in failure. Whether the fuel or air flows are affected by failed nozzles or obstructed or reduced compressor discharge conditions, then these conditions can lead to the onset of dynamics in one or more combustors. Even failures of combustor liners or transition pieces can bypass sufficient air away from the nozzles to cause a localized fuel rich condition that can increase NO_x emissions. The hardware damage is sufficient to cause combustion signature changes in both frequency and amplitude.

The following sections present some sample cases of combustion tuning done with the aid of CDMS technology coupled with other available engine performance data

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Table 6-2
Combustion Diagnostic Worksheet – Combustion Related Operational Faults

Faults	Dynamic Spectra Signature				Engine Supervisory Data					CEM
	Low Hz Press. Spike[1]	Med. Hz Press. Spike[2]	Hi Hz. Press. Spike[3]	Hz Shift	Flashbac k Thermo-couple	Exhaust Temp. Spread	Inlet Press. Drop	Hot Exhaust Temp.	Cold Exhaust Temp.	Changes in NOx and CO
Off-Calibration Fuel Nozzle	2	1								2
Eroded/Burned Fuel Nozzle	3				3	2				3
Clogged Fuel Nozzle		1	2	1		1	1`			1
Obstructed Air Passage	2	1						2		2
Clogged Air Filter	2	1						2		2
Compressor Fouling	2	1						2		1
Cracked Combustor Liner		2	1	3		1			3	
Cracked Transition Piece		1	3			3				3
Crossfire Tube Failure		2	1			1			3	

(1) Low frequency tone humming - <50 Hz for SW501F DLN and <140 Hz for GE 7FA DLN
(2) Med frequency: 50-100 Hz for SW501F DLN and 140 to 250 Hz for GE 7FA DLN
(3) High frequency: 100-250 Hz for SW501F DLN and 250-2500 Hz for GE 7FA DLN

Note: **Ambient temperature can affect the affect low to medium frequency spikes**
Changes in load tend to shift frequency, so always compare spectral signature at same engine load and ambient temperature

GE DLN 2.0 Combustor Tuning Example

Figures 6-11 through 6-13 illustrate the key elements of the DLN 2.0 fuel nozzles, cap and combustor liner. The primary supply feeds the diffusion burners that are used for start-up and for stabilization of the premix burners. There are four secondary burners and one tertiary burner - these are the lean premix burners. Premix burners are only used at higher loads. The tertiary burner comes in first, assisted by four primary diffusion burners. This is called the lean-lean mode. Subsequently the four secondary burners with the four diffusion burners situated in the middle of these premix burners raise the power output to 50-70% of base load. Then the diffusion burners are switched off and the premix steady state is reached. In this mode the quaternary burners come in. The diffusion burner inside the tertiary burner is not connected to the primary gas supply in the present GE DLN 2.0 burner system.

The different regimes during run-up of power are:

- Primary Mode- gas through the four diffusion burners only,
- Lean-Lean Mode- gas through the diffusion burners and the tertiary burner,
- Piloted Premix Mode- the four diffusion burners and all premix burners are on,
- Premix Steady State Mode- all premix burners and quaternary burners on.

An important design feature of the DLN 2.0 is its fully premixed operation at full load. This makes the combustor more susceptible to minor changes in equivalence ratio and ambient temperatures.

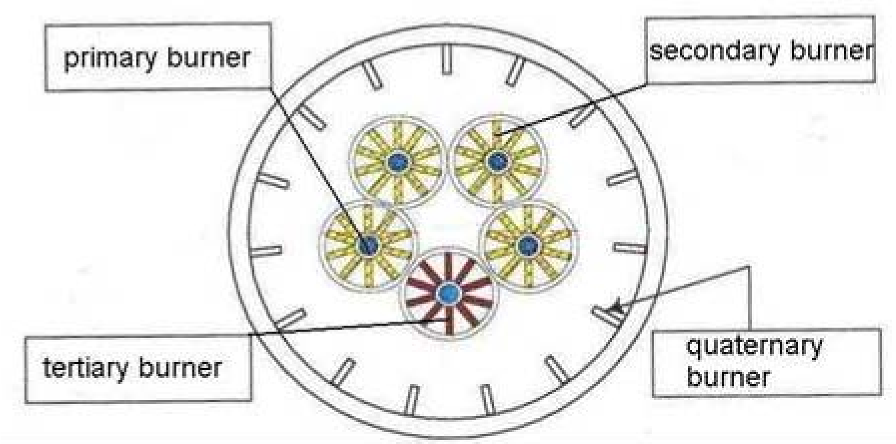


Figure 6-11
Cross-Section of the Four Types of Burners Present in a GE DLN 2.0 Gas Turbine Combustor

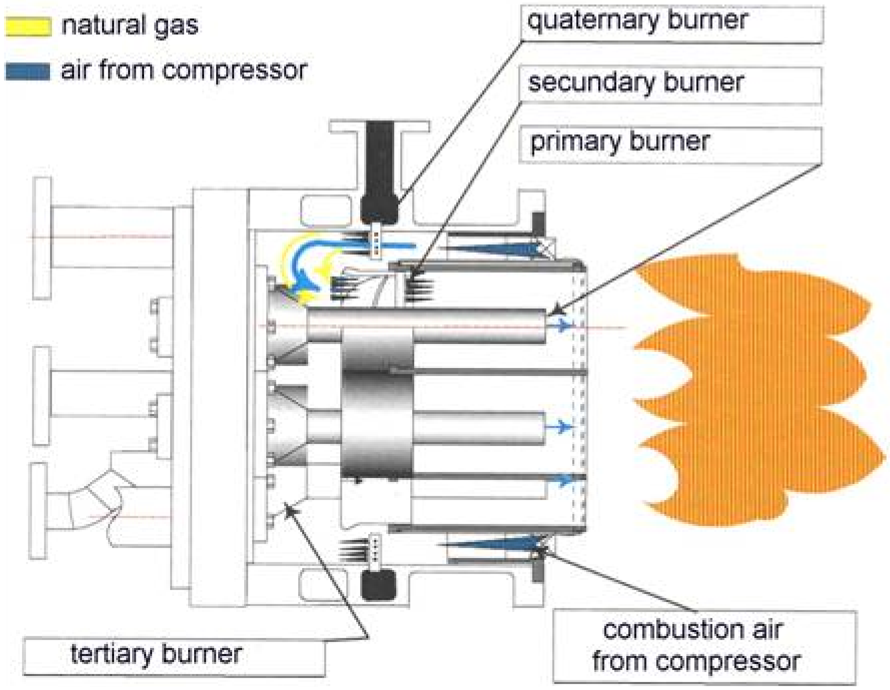


Figure 6-12
Side View of the Burner System of the DLN 2.0 Gas Turbine Combustor

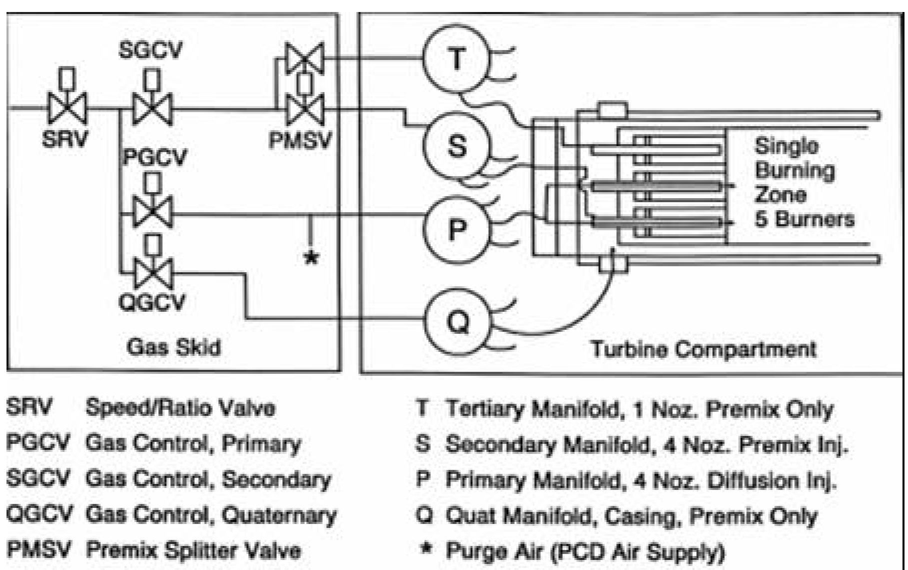


Figure 6-13
DLN-2.0 Gas Fuel System

To achieve the required emission performance in premix operation the combustion occurs close to or even below lean blow out (LBO) when sustained by the tertiary gas flow. Flame instability and dynamics can develop if operating parameters are outside the design limits.

The key parameters affecting the combustion process are:

- Air flow and its distribution in the combustor,
- Inlet air pressure and temperature that are influenced by ambient conditions, compressor performance, load, grid frequency,
- Gas supply pressure in which small fluctuations can lead to dynamics,
- Gas flow rate,
- Gas fuel composition,
- Fuel nozzles and combustion chamber design and condition.

Frequency spectrum of the 7FA DLN 2.0 shows a fundamental mode of 90 Hz and a harmonic of 130 Hz. The peak values depend on the fuel split between the secondary and tertiary fuel flows. The quaternary fuel reduces the 130 Hz mode but amplifies the 90 Hz. The target values are dynamics below 2 psi peak to peak and NO_x below 25 ppm. The variation between each combustion chamber and the differences in air flow between them necessitate individual tuning of each unit. Once this is done, retuning is needed only when the combustion chambers are dismantled.

In order to prevent spreading of instabilities from one fuel nozzle to another through the fuel system, the fuel nozzles were modified to become “softer” by installing an orifice upstream of the nozzle. This prevented the propagation of pressure fluctuations between different combustors through the common fuel pipe. The pressure drop across the swirler was increased to enhance mixing. This resulted in increased flame stability without significant penalty in combustion turbine efficiency.

Even though the combustors are constructed identically with the same nominal design of inlet and outlet pressures, some flow variation remains between combustor due to manufacturing tolerances. This can result in excessive tertiary fuel flow to suppress dynamics in the worse combustor, resulting in excess NO_x. The solution was to tune the system such that each combustor gets the amount of fuel to match the different airflow. An individual turning valve (see Figure 6-15) for each combustor was added in the secondary gas fuel lines downstream of the secondary/tertiary flow split valves. Tuning for an individual combustor is achieved by monitoring the pressure on each unit while changing the fuel flow rate. The initial tuning is typically done by GE engineers and requires 4 hours. The turndown of a DLN-2 combustor tuned to 9 ppm NO_x and CO was estimated to be about 70% load, compared with 40% load for the 25 ppm NO_x and 15 ppm CO system.

In DLN 2.0, combustion instabilities are known to manifest themselves principally at the following FFT frequency bands:

- Band 1: Less than 140 Hz,
- Band 2: 140 to 500 Hz,
- Band 3: Greater than 500 Hz.

Specifically, large resonant amplitude at specific frequencies indicates the following combustion instabilities:

- 10 Hz – Flameout,
- 70 Hz – Chug.
- 100 Hz – Flashback (axial instabilities).
- 175 Hz – Vibration (radial Instabilities).
- 200 Hz – Screech.

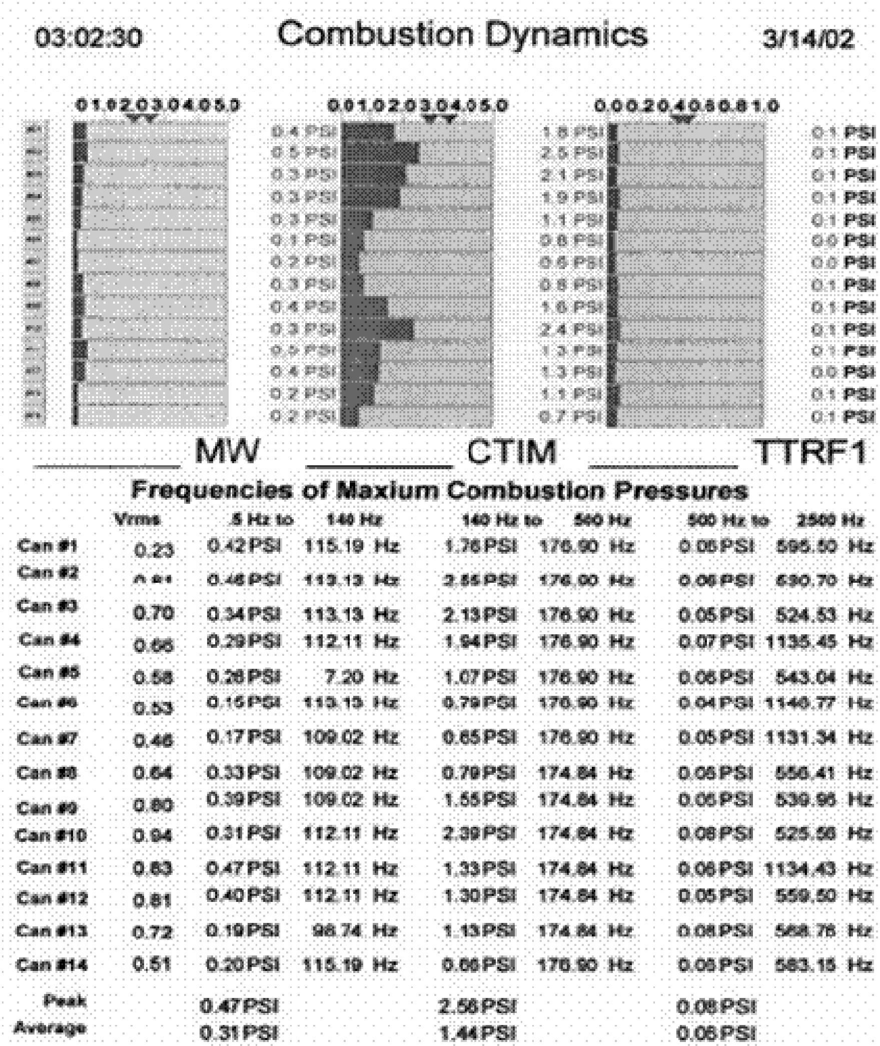
Figure 6-14 shows the scan obtained on March 14, 2002 at 03:02:20. The bar charts indicate the alert levels when amplitudes reach set points. Table 6-3 shows data on engine operation that should be monitored along with CD data in each combustor. For Band 1, the set points are at 2.4 and 2.6 psi. For Band 2, the set points are at 3.0 and 3.5 psi. For Band 3, the set point is at about 4 psi. Although Band 2 set limit is at 3.5 psi, combustor damage due to vibration does not occur until amplitudes of 4.5 to 6 psi are reached. Therefore, the set points are established to give the operators sufficient time to adjust fuel flows and bring amplitudes back to the normal operating range with no impact on hardware.

For this particular frequency scan, only vibrational instabilities were sufficiently high to approach set points in some of the combustors, specifically combustor number 2 and 10. Adjustments to the fuel flow for the affected cans are made only when the set points are reached. Operating data must be evaluated in conjunction with the FFT scans to obtain an overall evaluation of the changes that might have occurred to affect CD in a particular combustor and an accurate status of the combustion stability in the turbine. For the scan in Figure 6-14, Table 6-3 lists the corresponding engine operating data, such as compressor inlet temperature (42°F), turbine load (167.5 MW), total fuel flow (20.91 lb/sec), secondary split (77.38 percent the fuel flow to the combined secondary and tertiary fuel nozzles) and quaternary split (9 percent of the total fuel flow). Note that the secondary split was 77.38, which is slightly less than the even 80 percent split, indicative of equal gas flow to the four secondary nozzles. Based on operating experience it has been found that the combustors operate more stably with a secondary split in the range of 76 to 79.

Table 6-3
User Defined Display – 7FA (Reproduced Form Printout)

Signal	Description	Value	Units	Signal	Description	Value	Units
STATUS	Base Load			WI_PPR	WI Press Ratio	0.95	prs_R
DLN MODE	Premix			AA_PPR	Atom Air Press Ratio	0.97	prs_R
TTRF1	Combustion Ref Temp	2356	deg F	LF_PPR	Fuel Oil Press Ratio	0.96	prs_R
DWATT	Load	167.5	MW	AAMANPR	Atomizing Air Pressure	-7.6	psi
CSGV	IGV Angle	88.1	DGA	WICHKPR	Water Injection Pressure	-2.8	psi
FSRXSR	Primary Split (P/(P+S+T+Q)	0.00	%SPLT	LFCHKPR	Fuel Oil Pressure	-0.4	psi
FSRXPR	Secondary Split (S/(S+T))	77.38	%SPLT	FSGR	SRV Position Feedback	57.83	%
FSRQT4	Quaternary SpltQ/(P+S+T+Q)	9.00	%	FSRP	Fuel Split Primary	0.00	%
FXKSPMSB	Premix Split Bias	0.00	%SPLT	FSRS	Fuel Split Secondary	60.16	%
FQLM1	Liquid fuel Mass Flow	0.00	#/sec	FSGX	Fuel Gas Cont VIv LVDT (Premix)	63.95	%
WQ	Water Injection Flowrate	0.00	#/sec	FSGQ	Fuel Gas Cont (VIv LDTV (Qt)ua	13.86	%
CSRBH	DLN Bleed Heat Flow	3.00	%	CPR	Compressor Press Ratio	15.40	prs_R
CTD	Compressor Disch Temp	706	deg F	FQG	Fuel Gas Mass Flow	20.91	#/sec
CPD	Compressor Disch Press	213.3	psi	FSR	Fuel Stroke Reference	66.15	%
CPDABS	Comp Disch Absolute Press	228.2	psi	FXKSPMSB	Premix Split Bias	0.00	%SPLT
TTRX	Exhaust Temperature	1093	deg F	FXKSPMSBMX	Premix Split Bias Set (max)	1.25	%SPLT
TTXM	Mean Exhaust Temperature	1093	deg F	FXKSPMSBMN	Premix Split Bias Set (min)	-3.00	%SPLT
TTXSP1	Exhaust Spread 1	81	deg F	CMHUM	Fuel/ Air Ratio	0.0048	#H/#A
TTXSP2	Exhaust Spread 2	69	deg F	AFPAP	Atmospheric Press	30.4	in Hg
FTG	Fuel Gas Temp	56	deg F	CTIM	CT Inlet Temp	42	deg F

When the CD scan shows an unacceptable amplitude for any of the 14 combustors (i.e., pressure amplitude nears or exceeds the alert set points), an adjustment is made to the secondary fuel flow. This is possible by adjusting the manual controlled valve in the PM2 line to each combustor. Figure 6-15 shows the placement and manual adjustment made to the valve.



Frequencies of Maxium Combustion Pressures

	Vrms	.5 Hz to	140 Hz	140 Hz to	560 Hz	500 Hz to	2500 Hz
Can #1	0.23	0.42 PSI	115.19 Hz	1.76 PSI	176.90 Hz	0.05 PSI	595.50 Hz
Can #2	[illegible]	0.46 PSI	113.13 Hz	2.56 PSI	176.00 Hz	0.06 PSI	590.70 Hz
Can #3	0.70	0.34 PSI	113.13 Hz	2.13 PSI	176.90 Hz	0.05 PSI	524.53 Hz
Can #4	0.66	0.29 PSI	112.11 Hz	1.94 PSI	176.90 Hz	0.07 PSI	1135.45 Hz
Can #5	0.58	0.28 PSI	7.20 Hz	1.07 PSI	176.90 Hz	0.06 PSI	543.04 Hz
Can #6	0.53	0.15 PSI	113.13 Hz	0.79 PSI	176.90 Hz	0.04 PSI	1140.77 Hz
Can #7	0.46	0.17 PSI	109.02 Hz	0.65 PSI	176.90 Hz	0.05 PSI	1131.34 Hz
Can #8	0.64	0.33 PSI	109.02 Hz	0.79 PSI	174.84 Hz	0.06 PSI	556.41 Hz
Can #9	0.80	0.39 PSI	109.02 Hz	1.55 PSI	174.84 Hz	0.06 PSI	539.96 Hz
Can #10	0.94	0.31 PSI	112.11 Hz	2.39 PSI	174.84 Hz	0.08 PSI	525.56 Hz
Can #11	0.83	0.47 PSI	112.11 Hz	1.33 PSI	174.84 Hz	0.06 PSI	1134.43 Hz
Can #12	0.81	0.40 PSI	112.11 Hz	1.30 PSI	174.84 Hz	0.05 PSI	559.50 Hz
Can #13	0.72	0.19 PSI	98.74 Hz	1.13 PSI	174.84 Hz	0.08 PSI	568.76 Hz
Can #14	0.51	0.20 PSI	115.19 Hz	0.66 PSI	176.90 Hz	0.05 PSI	583.15 Hz
Peak		0.47 PSI		2.56 PSI		0.08 PSI	
Average		0.31 PSI		1.44 PSI		0.06 PSI	

Figure 6-14
DLN 2.0 Signal Display Showing Combustors 2 and 10 Approaching Alarm Limits

Figure 6-15
One of Fourteen Manual Values for Adjusting Secondary Fuel Flow to an Individual Combustor

The steps for tuning for dynamics are as follows:

- Take a reference dynamics scan of all combustors,
- Locate the combustor with the highest dynamics (pressure amplitude),
- Note the position of each combustor with high dynamics in relation to all the other affected combustors (dynamics in adjacent combustors can be affected by one mistuned combustor),
- Reduce secondary fuel to the affected combustor in 1 to 2% steps either up to 8% or to where a different combustor has the highest dynamics. A new dynamics scan should be taken with each step,
- Repeat the step above with each of the affected combustors, even new ones affected by the last change,
- If the dynamics continues to be excessive with a full 8% reduction, additional tuning can be tried by cutting fuel to the adjacent combustors. If no effect, return the secondary fuel to the adjacent combustors to the same level and reduce the secondary fuel to the affected combustor to a maximum of 10%.

Normal temperature spreads in fully premixed combustion should register levels below 80°F. Tuning of combustors is also necessary if the spreads are in excess of 100-140°F. In this situation, the fuel to the combustor responsible for the recorded increase in temperature spread should be reduced in 1-2% incremental steps. Each percent reduction in fuel should reduce the exit temperature by 10°F.

Tables 6-4 and 6-5 provide general tuning and OEM diagnostic guidelines for the DLN 2.0 combustors.

Table 6-4
DLN-2 Fuel Split Effect on Emissions and Dynamics

Symptom	Possible Causes
High NO_x	Tertiary split too high. Quaternary too low
High CO	Tertiary split too high
High 150-180 Hz Dynamics	Tertiary and quaternary splits too low
High 100-130 Hz Dynamics	Quaternary split too high. Tertiary too low

Table 6-5
Matrix of DLN-2 Operational Problems and Tuning Fixes

Problem / Symptom	Cause	Actions
High spread and/or flameout during the premix transfer	Secondary split too high	Decrease secondary split bias by one percent
	Premix transfer in temperature too low	Lock unit out of premixed mode, load unit to 50°F above normal premixed transfer temperature, unlock premixed mode. If successful contact engineer regarding permanent change to transfer temperature
	Blockage in secondary fuel passage to one or more combustors	Repeat to verify spread locations, check pigtails in suspect chamber(s) and /or flow check the passages
High spreads due to "hot spot"	Missing quaternary, flange orifice fuel nozzle defective or damaged missing fuel nozzle seals	
	Tuning valve left full closed blockage in secondary, or quaternary feed over aired or under fueled chamber	

Summary

Modern DLN combustors must operate within tight tolerances of fuel/air ratio, fuel/air mixing and heat release rate in order to deliver single digit NO_x performance with combustion stability and design power output. Changes in ambient temperature, humidity, and fuel composition can lead to the onset of combustion instabilities. Without immediate operator intervention and periodic combustion tuning, combustion instabilities can result in emergency shutdown with significant lengthier and costlier repair down times. In response to these operational risks, many new combustion turbines come equipped with piezoelectric pressure transducers to monitor dynamics in each combustor. In regions subject to broad changes in ambient conditions and fuel composition, the retrofit combustion dynamics monitoring systems (CDMS) to DLN turbines has been advocated. These monitoring systems, available from OEMs and independents, can provide evidence of the impending and unsafe pressure fluctuations as well as online diagnostics for preventive maintenance. The frequency and pressure amplitude signals from these CDMS coupled with engine emissions and supervisory data provide a wealth of information on the health of DLN combustors. EPRI continues to develop technology and methods for monitoring and controlling the onset of combustion dynamics and to map the operating regime for stable combustion in modern DLN combustors.

7
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