Simplified Analysis and Practical Mitigation Techniques for Industrial Boiler Thermoacoustic Vibration

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Abstract

Industrial boiler systems are prone to mechanical vibration which can lead to premature component failures, emission non-compliance, and greatly increased ambient noise generation. Driving mechanisms, thermoacoustic phenomena, and system geometrical interactions are complex. General methods for accurate predictive analysis of system vibration are historically unreliable. Rather, to effectively mitigate vibration in combustion systems, design best practices can be employed to minimize vibration potential. Should system vibration occur, this can be effectively coupled with comprehensive vibration data collection, prescriptive analysis methods, and system modeling techniques so that the most probable system drivers can be isolated and mitigation techniques can be quickly employed. An overview of combustion acoustic theory and design best practices with a strong emphasis on combustion as acoustic amplifier will be presented. Vibration data analysis, system modeling methods, and mitigation techniques will be presented. A case study of combustion system vibration will be discussed outlining vibration observations and effects, and how analysis of key design features and vibration data led to successful strategies for resolution.

Introduction

It is unknown when the first combustion vibration issue was encountered, but the most expensive is well known. During NASA's development of the F1 liquid rocket engine for the Apollo program in the late 1950's to early 1960's very severe combustion instabilities occurred on the first generation of tests resulting in complete failure of the rockets. As the failure mechanisms were largely unknown, test engineers were forced to use a trial and error approach which required several years to resolve. The engine was subjected to over 2000 full-scales tests through development and testing, a very time consuming and expensive process to resolve the instability and ensure failure modes could not be reproduced [15]. Ultimately, after significant effort, NASA engineers resolved the instability with the use of flow baffles inserted at the point of fuel/air injection. A successful strategy, later analysis would determine, due to the baffle's attenuation of transverse waves near the injection point.

Burner related rumbling or combustion driven oscillations has been observed dating back to at least an observation of singing flames by Dr Higgins in 1777. Researchers have proposed models to explain this phenomenon [2-4] since the early to mid-1900s.

The concept of a burner amplifying air and fuel pulsations was first described by Lord Rayleigh [7] in 1845 " … If heat be given to the air at the moment of greatest condensation [compression], or taken from it at the moment of greatest rarefaction, the vibration is encouraged"
Putnam [2] later restated this mathematically as the Rayleigh criteria:

$$\int h\, p\, dt > 0$$

Where: h = instantaneous rate of heat input;
p = instantaneous pressure (pressure – average pressure);
t = time

This can be stated as: the amplitude of pressure oscillations present in the system (p) will become amplified when heat input (h) is synchronously added to the system. Rather than providing guidance for resolving combustion vibration, this equation simply states the nature of flames as amplifiers.

Combustion vibration is a general term that refers to any mechanical vibration observed during firing modes of combustion systems caused by fluid pressure oscillations. Thermoacoustics refers to the relationship of fluid temperature, density, and pressure variations on acoustic waves within a system. Since the combustion process involves rapid changes in temperature, density, and pressure, all combustion vibration is inherently thermoacoustic.

Enclosed combustion systems are generally complex and include cold and hot gas sections, various aerodynamic flow regimes including mixing and recirculation, and rigid cavities for reflection of acoustic waves. Therefore, aerodynamics, combustion, and system geometry are all critical components of acoustic design review.

All forms of enclosed combustion processes can encounter forms of combustion instability and system vibration. This includes boilers (from residential to utility scale), process heaters, the gas turbine industry, and others. Combustion vibration issues encountered either in the field or during development have historically been resolved with the same approach as the example from NASA, through trial and error efforts. In general, experience in the industry led to basic design rules and resolution steps for different application types, however, accuracy and diagnostic confidence remained low. i.e. Methods to resolve a past combustion vibration problem, may not resolve a similar issue encountered, and a very different mitigation strategy may be required.

Vibration problems or instabilities can be caused by a variety of sources, either intrinsic to a component or the interaction/feedback with at least one other component device or boundary condition. The resultant vibration can range from annoyance to highly destructive.

Combustion instabilities can be classified by two general sources:

1. Intrinsic instabilities caused by a single component. This is referred to as a forced response and the source of the vibration is described as an acoustic driver. This can be sub-divided into:
    a. Burner sourced: Combustion acts as the acoustic driver.
    b. Non-burner sourced: Other energy sources including rotating equipment or the flow of fluids in piping, ducts, and enclosures can drive external mechanical vibrations.
2. System instabilities caused by the interaction or one or more components or boundary conditions. This is referred to a natural response or system resonance.

With industrial designs, isolation of acoustic drivers is generally difficult due to the complexity of interactions. Intuitive deductive logic generally cannot be used to solve combustion driven oscillations. For instance, identical units may not respond identically. The resultant vibrations may be physically located far from the source. As a result, incorrect conclusions are common without deeper analysis.

For designers of burner and furnace equipment, a well-known empirical relationship exists between furnace sizing and probability of system vibration. For a given heat input rate, as the furnace radiant section volume increases, the probability for vibration decreases; similarly, the probability increases as furnace volume is reduced. This can be explained (neglecting significant nuance) that with larger furnace volume there is more length-scale for acoustic waves to dissipate adding damping to the system for acoustic reflections. This trend in no way represents all other design features of acoustic concern; but does correlate well with the general industry trend related to vibration frequency relative to combustion equipment type. That is, process heaters which have large furnace radiant sections relative to heat input, tend to have far fewer vibration cases in industry compared to the boiler market. Similarly, combustors and specialty combustion equipment which have very high space heat release tend to have more significant challenges with vibration and pressure oscillation.

For technology sectors that include significant numbers of entire enclosed system design duplicates such as the gas turbine industry, the financial scale is very large compared to a single design and significant effort is employed during development to avoid acoustic modes both in terms of testing and model simulations.

A challenge in the boiler industry is that burner and furnace configurations vary greatly and may include unique design features. Similar applications may not have similar vibration trends. Further, testing at full-scale is not economical for large industrial applications, and partial-scale tests may not capture full-scale behavior

Vibration in Industry

Prior to discussing vibration in combustion systems, it is necessary to establish a baseline of industrial trends of the acceptability of vibration levels. Combustion vibration specifically refers to the mechanical vibration and deflection of system structures driven by acoustic pressure oscillations within enclosed chambers and ducts. The frequency range of concern for vibration is less than 500 Hz. The typical industry breakdown of this range is:

1. Low Frequency < 20 Hz: Often referred to as panting or breathing. Observers typically can see pulsation in the flame

2. Medium Frequency 20-150 Hz: Referred to as rumble. Vibration is typically observed as mechanical shaking, but typically cannot be directly observed in the flame.
3. High Frequency 150-500 Hz: Referred to as a whine or howl. Often only a noise/annoyance issue.

Excessive vibration can lead to:

1. Mechanical stress fatigue failures of either structural supports or devices mounted to supports.
2. Ambient noise generation: Larger structural surfaces can act like speakers to propagate sound due to low amplitude higher frequency vibrations.
3. Emission non-compliance: Acoustic pulsations in systems may drive burner stoichiometry between fuel rich and lean operational regimes resulting in net higher emissions of NO_x or CO compared to steady state.

Due to the nature of combustion processes all equipment generate some level of mechanical vibration. What vibration level will lead to pre-mature failure compared to annoyance has been a debated topic in industry for at least 65 years. No concrete Industry standards have been adopted that can be globally applied. Usually the debate is initiated regarding a specific installation based on individual perceptions of what is normal or excessive. In many cases, site specific specifications have been created and the allowable limit becomes that of achieving specification compliance. Vibration level limits for specific equipment such as fans, rotating shafts, burners, chamber walls, boiler casings, flow in piping have been specified, recommend and computed routinely.

Figure 1 provides recommended guidance levels for vibration as a function of frequency based on observation and experience. [19]

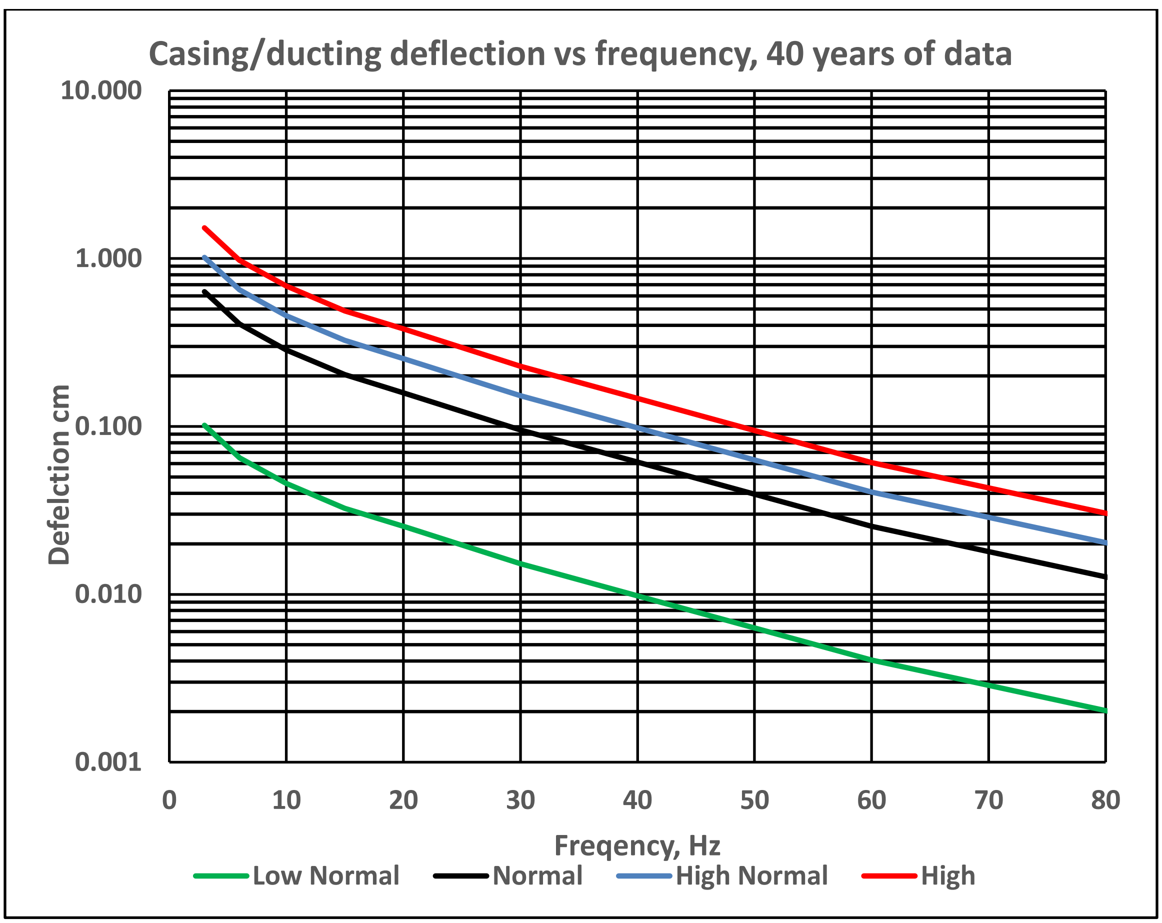

Figure 1 – Observed casing and ducting deflections based on site problems and expert perception

It is important to note that absolute deflection is proportional to mechanical stress and deflection is inversely proportional to structural stiffness. An overriding guide of allowable vibration is any mechanical damage due to over stressing structural components. Almost always deflection decreases with frequency as the shown in Figure 1, if frequency dependent at all. Synchronous and periodic frequencies of vibration may not exist at low frequencies less than 5 Hz and many times the existence of these low frequency vibrations may be random, not time dependent at all. Low, Normal, and High labels in Figure 1 are relative milestones within the dataset and are not meant to provide recommendations for acceptable levels on a specific application, although the "High" category may be loosely associated with cause for structural concerns.

If the observed vibration is not associated with structural damage over a time frame to cover fatigue limits, then the observed vibration may be nothing more than a perceived annoyance. To determine the time required to observe damage from fatigue refer to Figure 2 below [20].

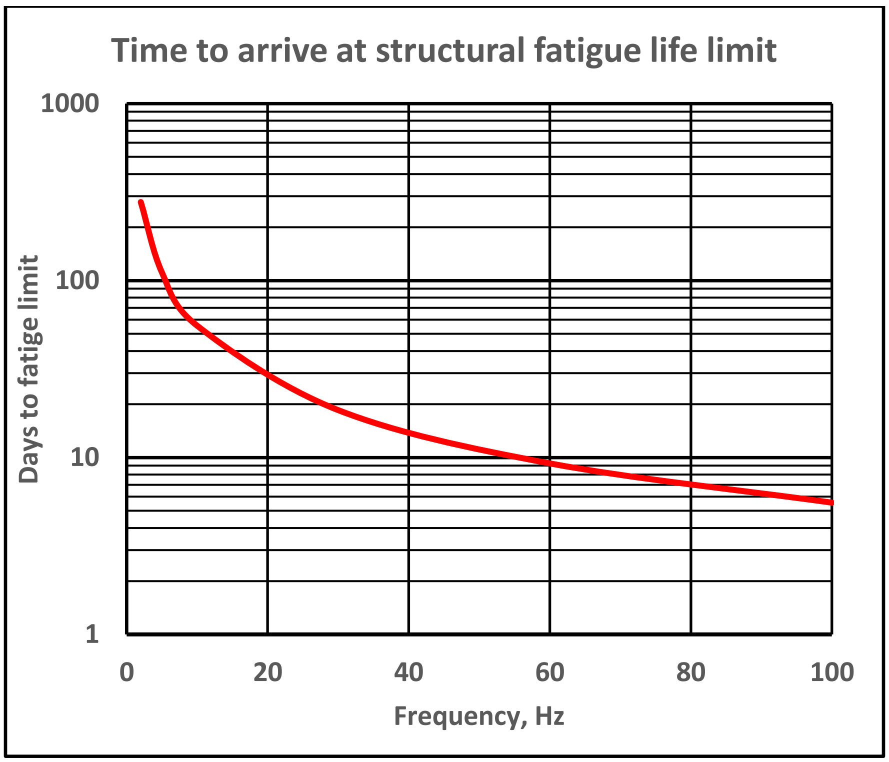

Figure 2 – Fatigue time limit vs vibration frequency [20]

Experience has shown that if the observed vibrations do not cause mechanical failure and are generally in the range of "Normal" as shown in Figure 1, then the mechanical vibration can be classified as normal and expected or simply a perceived annoyance. Mechanical vibration can be reduced by adding structural reinforcements to achieve reduced amplitude of deflections. Though an effective tool, stiffening alone may not resolve all vibration related issues in combustion systems, especially in the case of severe resonance issues.   For these cases, analysis methods are required to identify the sources of vibration and mitigation techniques must be employed.

Predictive Analysis

To predict thermoacoustic vibration it is desirable to model the underlying phenomena. A further development of the Rayleigh criteria was provided by Baade [4], who presented the instability process in the form of a feedback loop shown in Figure 3.

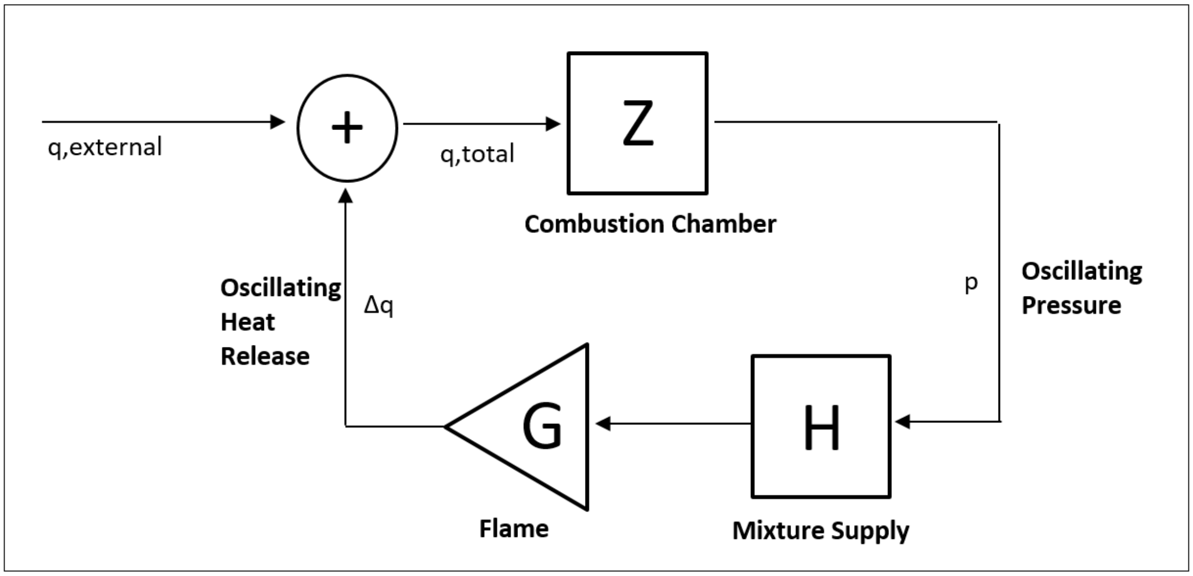

Figure 3 – Combustion Instability Feedback Loop Model [4]

This chart expresses the acoustic stability of the system as the product of transfer functions for the furnace chamber [Z], mixture supply (i.e. burner assembly) [H], and flame [G]. These functions, if known, would fully describe the acoustic properties of a system. However, these are complex functions of magnitude and phase both of which are frequency dependent.

The combustion chamber and mixture supply functions can be evaluated as an acoustic impedance. This is the frequency specific resistance of the system (ratio of pressure oscillation amplitude to volumetric flow oscillation). If geometry is known, impedance can be estimated using calculated acoustic modes of the system.

The flame transfer function, however, is more complex as along with frequency dependence the variation of both velocity and mixture rate must be considered.

This feedback loop model has been implemented on boiler systems [9-11]; however, these systems were very small capacity (0.5 MMBtu/hr fuel heat release) relative to industry scale and required specialized testing and diagnostic equipment while utilizing simple burner designs.

For predictive analysis, unless combustion technology is an exact duplicate, determining the flame transfer function based on design features alone is at best highly inaccurate.

In practice on the industrial scale, even for prescriptive analysis, using this method to successfully model the transfer functions has significant challenges. First, the industrial scale creates logistics challenges for effective testing equipment. For instance, an acoustic pulse generator is an important testing device required to obtain system transfer functions of Baade's model. To generate low frequencies pressure pulsations at sufficient amplitude for measurement accuracy, a loudspeaker sized for >1kW may be required! Sourcing portable or temporary test devices, thus, can become very impractical.

Second, many industrial applications use complex burner designs and custom features and/or design conditions such as waste fuels and variable burner configurations to achieve lower pollutant (e.g. NO_x and CO) generation. Thus, though this model can provide a better qualitative understanding of acoustic modes, it may not be a good diagnostic tool for troubleshooting on industrial-scale equipment.

Due to the challenging nature of accurate full system modeling, it is doubtful that a practical universal method exists to fully predict acoustic performance of industrial combustion equipment. Rather, best design practices can be employed to reduce risk of vibration. If vibration occurs, vibration data collection, data analysis, and prescriptive modeling techniques, can be utilized to identify most-probable system drivers and engineer practical mitigation strategies.

Vibration Data

Should a noise or vibration issue occur, the first step should always be equipment tuning. On many systems, either equipment adjustments or process condition variations can be made to alleviate operational vibrational modes. Industrial experience in the boiler market highlights that most vibration issues encountered can be mitigated by tuning. It is only where equipment tuning does not resolve vibration, or tuning requirements drive emissions into non-compliance that further mitigation strategies are required.

To avoid a trial and error approach and/or chasing observational perceptions of vibration that can be misleading, it is critical that the acoustic and vibration data of the system should be collected.

The critical items to collect are:

1. High-speed pressure transducer data: [Minimum 1000 Hz response time recommended]
2. Wall structure acceleration and deflection amplitude

Additional data to collect of potential value include:

3. Ambient sound measurement
4. High-speed video capture of the flame

Data measurements should include:

- Multiple test points on system: Minimum of furnace and windbox, and if available the fan air inlet and boiler convection/economizer sections
- Firing rate and process condition variation
- Fan-only data

Following data capture, data must be processed to reveal amplitudes, frequencies, and trends. High-speed data is typically viewed over a 2-5 second window. Such graphs are often referred to as Raw time-series data or more formally as signal oscillograms. Refer to Figure 4 for an example of oscillating time-series data.

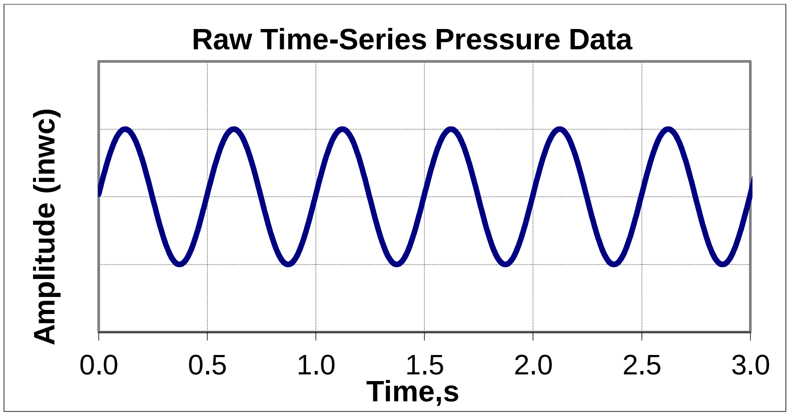

Figure 4 – Raw Time-Series Example

Time-series can be converted to frequency bands via mathematical processes such as a Fourier Transform [21]. Graphs of this are referred to as frequency spectrum data or more formally as power spectral density. Refer to Figure 5 for an example of processed frequency data using the time-series data of Figure 4.

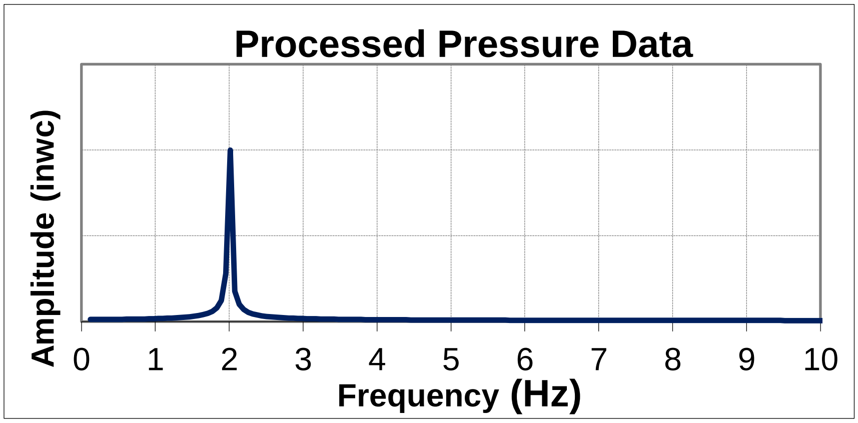

Figure 5 – Frequency Spectrum Example

Reviewing and comparing the time-series data and processed frequency data is a critical tool for vibration system analysis. Averaging filters for time-series data can be beneficial to isolate higher and lower frequency ranges. Additionally, it is very useful to evaluate and compare the high-speed pressure data and high-speed accelerometer data at varying process conditions. Both acoustic properties and structural stiffening properties of the system can be revealed.

System Modeling

Due to the scale and custom nature of many industrial combustion applications, this paper argues a simplified system modeling approach will lead to faster resolution of the majority of combustion vibration issues.

Vibration and noise in industrial boiler systems arises from the complex interaction of rotating equipment, combustion, structural design, and aerodynamic fluid flow. Though potential sources are numerous, the sequence of acoustic sound wave generation and resultant mechanical vibration can be simplified to a five (5) step process shown in Figure 6.

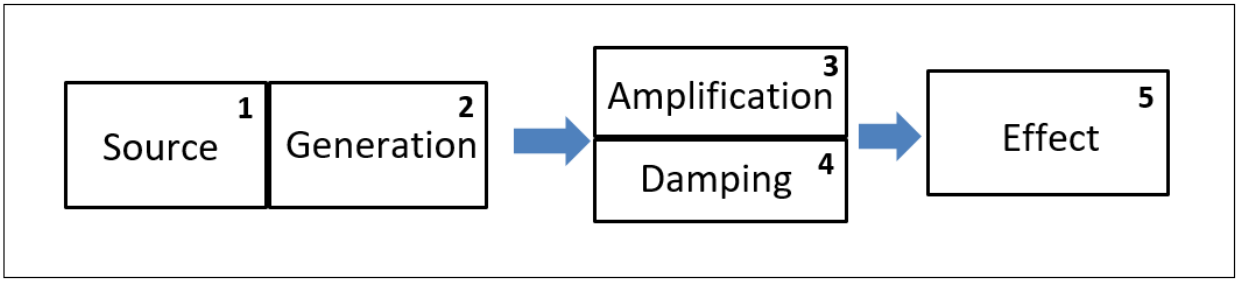

Figure 6 – Sound Generation Process (Simplified)

1. **Source**: The sources of energy added to the system. This includes fans, combustion, and various fluid flow (e.g. air, fuel, steam, water, and flue gas).
2. **Generation**: This is the conversion of energy from a source to acoustic energy.
3. **Amplification**: Geometry and structure dependent natural frequencies of the system that can provide positive feedback to amplify acoustic energy.
4. **Damping**: Geometry, structure, and flow path dependent properties of the system that determine the transmission loss and attenuation of acoustic energy.
5. **Effect**: The final output as a balance between generation, damping, resonant/amplifying sources, and system structural design.

Sources/Generation

Fans: Rotating equipment is a common source of vibration and a huge amount of information is available to reduce/eliminate vibration from power driven rotating equipment [16,17]. Generally, all fans generate some amount of cyclical noise/pulsations. Frequency peaks will occur at rotating frequency and blade pass frequency. For most fan systems, these peaks are low amplitude. Significant fan driven vibration can occur due to a number of phenomena caused by very specific flow conditions and design criteria; these include stall, rotating stall, and surge. These conditions are less common and tend to generate pressure amplitudes at low frequency (less than fan speed).

It is important to note that combustion systems may be quiet during purge conditions (fan-only), yet fan systems can still be the primary acoustic driver of vibration, meaning successful mitigation techniques may require fan system changes.

Combustion: Combustion may be a source of vibration or, more commonly, an amplifier of sound generated from other parts of the system.

Significant vibration from combustion acting as acoustic driver is due to unstable flow patterns and regimes. These combustion instabilities include overly intense or confined combustion, weakly anchored combustion, severe flame impingement, and poor mixing. For these cases, the flame equilibrium position is weak and small perturbations result in oscillating flame front as the flame anchor point moves. Small pockets of gas may ignite at once creating mini explosions. The pressure signature tends to be erratic with high pressure oscillation amplitudes over a broad spectrum (no significant frequency peaks). These conditions are typically mitigated with equipment design.

Amplification

Sound amplification can occur both from the combustion process and the acoustic modes of the system (resonance).

Combustion:
Due to the rapid pressure and temperature increase that occurs during combustion, all burners can be amplifiers of system noise. The primary design concept that determines combustion amplification strength is temperature uniformity. Since the speed of sound is a function of temperature, high temperature uniformity can better propagate acoustic waves, whereas there is more destructive interference if temperature is non-uniform. Two trends emerge from this concept:

1. Pre-mix and/or simulated pre-mix burners will be strong acoustic amplifiers.
2. Furnaces with small uniform cross sections will generally result in more combustion amplification.

The amplification effect of burners can be mitigated to a degree through burner design techniques such as staging, as this provides less uniform temperature profiles in the furnace. It is important to note that for every burner design, the combustion response to acoustic waves is frequency dependent. Staged burners have high impedance for higher frequency acoustic waves, but for lower frequencies impedance is significantly less similar to pre-mix burners, meaning they are equally susceptible to low-end resonant issues such as panting.

Resonance:
With all equipment, there exists natural frequencies of each part of the system. These can be related to length-scale, structure, or flow profiles. Resonance is a positive feedback cycle where small perturbations are produced at a frequency corresponding to a system natural frequency in a process of constructive interference. During each cycle, more acoustic energy (gain) is added compared acoustic energy dissipation (damping) resulting in a rapid and potentially large increase in acoustic energy. Stated mathematically, the amplification factor or Gain is greater than unity (1). The oscillation builds until it reaches a limit, referred to as the limit cycle, where the system response becomes non-linear and damping matches acoustic gain. Depending on system design and most importantly burner design type, limit cycle final output can vary between mild to very severe vibration.

In most parts of real systems, significant damping exists in the system such that the Gain is less than unity (1) for most natural frequencies. Resonant related vibration issues are defined by large high-amplitude pressure pulsations at a single frequency that matches a natural frequency of the system.

Resonance in combustion systems can occur due to three (3) principle phenomena:

Fluid Flow: Fluid flow can create pressure peaks through oscillation as fluids are diverted by system geometry such as flowing around tubes or through elbows. Oscillation is the natural resonance frequency of the changing flow field. Examples would include vortex shedding over convection tubes, or precessing vortex core of a burner or fan inlet. This resonance is geometry dependent and a function of velocity.

Standing Wave: This is the fundamental frequency (and harmonic nodes) of each principle length-scale measurement inside a furnace system. This occurs when the sound wavelength matches a length-scale dimension based on the speed of sound (c) through the media. The speed of sound is primarily a function of temperature.

The natural frequency for a given temperature is constant; however, the temperature of a furnace changes with firing rate, excess air, and flue gas recirculation (FGR), meaning there will be a small shift in natural frequency based on firing rate and tuning conditions.

Spring-Mass Related: This is the natural frequency of a mass (or weight) attached to spring. In boiler systems, spring-mass related resonance occurs in two forms:

Helmholtz Resonance: This relates to volumetric sections of a furnace system resonating. Cavities can act as a mass, while high velocity sections can act as a spring. Helmholtz resonance is primarily a function of internal dimensions. Frequency only has weak dependence on temperature/density and is not altered by flow or velocity. Thus, the peak frequency is generally constant and not a function of load. Helmholtz resonance is low frequency and most commonly associated with "panting" of combustion flames.
Typical sources would be the air inlet ducting (spring) and windbox (mass), or the furnace stack (spring) and furnace chamber (mass).

Structural Natural Frequency: This is the natural frequency of mechanical structures. This is related to the mass of structures and their respective stiffness. Stiffness refers to the amount of force required to produce a defined deflection.
This natural frequency is related only to the dimensions and support of a structure, and thus is constant frequency.

Damping

Acoustic energy dissipation or damping is a critical component of system design. Putnam stresses in detail the importance of damping [2] to avoid modes of instability. Including sufficient damping in systems will drive Gain less than unity, preventing the resonant feedback cycle from building energy each cycle. The vast difference in observed vibration level from a small increase in damping is often surprising to observers. Severe vibration may be eliminated completely through the partial closing of a flow damper.

The primary damping source in combustion systems is from pressure drop from the flows of combustion air and flue gases. This includes flow through burner registers, furnace convection sections, economizers, and breaching. Damping for these devices, therefore, is highest at higher heat input ranges and is minimum at low fire. Due to this, a resonant mode (low frequency oscillation panting) is a common issue at lower firing rates, and rarely seen at higher firing rates.

Fan differential pressure is an additional source of damping. Flow control dampers, necessary for the control of air, FGR, and flue gas process conditions, provide a source of variable damping.

Effect

The combination of acoustic sources, amplification, and damping generates a complex acoustic signature within the combustion system. The degree these pressure pulsations result in mechanical vibration and deflection is determined by mechanical stiffness.

For a given magnitude of pressure oscillations, a stout structural design may have no noticeable vibration, whereas significant vibration could be perceived on a weaker structure. A common field observation of this effect is that vibration amplitude tends to be highest on cantilevered structures or sections connected by expansion joints. For many vibration issues, adding stiffening braces can significantly reduce mechanical deflection.

Mechanical stiffness can be expressed as a natural frequency obtained from the structure mass and stiffness. Lower natural structural frequency represents less stiffening where pressure pulsations will result in greater mechanical deflection.

Vibration Data Analysis

As described previously, instabilities can be either intrinsic to a component or due to the interaction/feedback with at least one other component device or boundary condition. Thus, combustion systems with vibration can be classified by two general sources with the following characteristics as shown in Figure 7.

	Category 1	**Category 2**
Time-Series Data (Vibration Signature)	**High repeatability**	**Mostly Erratic**
Processed Frequency Data	Frequency Peaks Sharp and High Amplitude	Frequency Peaks Broad and Lower Amplitude
Acoustic Driver	System entering into acoustic resonance	Energy producer is driver: Fan or Combustion or Both
Mitigation Strategy	Review system natural frequencies, and target accordingly	Review burner and fan design. Acquire fan-only data. Test design changes.

Figure 7 – System Vibration Categorization

Category 1 - System Resonance: Vibration in these systems is caused by a resonance. A constructive feedback cycle occurs where multiple natural frequencies overlap that causes a large build in pressure oscillation amplitude. These systems are characterized by a narrow range of operation, or certain operating conditions, where vibration amplitude dramatically increases. Outside this load range, vibration may be very minimal. Resonant frequency peaks will be high amplitude and comparable in magnitude to time-series data. Mitigation strategies typically focus on changes to system geometry or structure to alter system natural frequencies, or acoustically isolating system components via damping techniques.

Category 2 - System Instabilities: Vibration in these systems are driven by the components forcing energy into the system such as combustion or the forced draft fan. For these systems, the general trend is that vibration occurs across a wide range of firing rates and the vibration amplitude builds gradually with firing rate. Resonant frequency peaks may be present, but these peaks are minor compared to time-series pulsation amplitude. Mitigation strategies typically focus on changes to the acoustic driver (combustion changes or changes to fan system).

Real systems may exhibit characteristics of both categories, however, identifying the system type and corresponding mitigation strategy generally provides a much higher chance of resolution.

Thus, to fully diagnose vibration in combustion systems that experience vibration, it is necessary to understand acoustic drivers, damping, and resonance. Though much information can be gained from vibration data and system analysis, for real combustion systems, this relationship is very complex and system analysis is more likely to reveal sources of higher or lower probability rather than generating firm conclusions.

Best Practices

Based upon considerable industry experience [19] and a deep review of acoustic system analysis, design best practices can be recommended. If implemented, vibration remains a potential risk, but equipment can avoid certain vibrational modes and more tools for mitigation are available for field tuning.

Burner:

Londerville [1] provides several burner design criteria recommendations to avoid gas-fuel burner pulsation.

1. Design with a single flame holder: Multiple stabilizing zones, either intentional or inadvertent can cause burner instability due to oscillation of the flame front.
2. Always provide gaseous fuel admission such that ignition zones are defined by mixing limits rather than thermal ignition: This is required to prevent zones of flammable mixture igniting via "mini" explosions.
3. If burners include expanding refractory throats (quarls), avoid refractory length to burner diameter ratios of 0.28-0.35. Refractory lengths should either be shorter or longer. This recommendation is made as the flame recirculation length is a function of this ratio. A non-linear response exists within this ratio that should be avoided.

Additionally, Londerville provides a recommendation for traditional burner technology to avoid rapid mix of fuel and air in combination with uniform Air/Fuel ratios. This is recommended to avoid high, even thermal gradients across the flame front, since temperature uniformity across the throat area may encourage oscillatory flame front motion normal to air pulsations. It should be noted, however, that with later low NO_x burner developments, this recommendation is often purposely not followed as a method to achieve ultra-low NO_x levels. For this technology class, acoustic risk increases, and additional mitigation techniques are typically required.

Eisinger and Sullivan [8] provide guidance on burner geometric design criteria to avoid. This is expressed in terms of ratio of combustion air burner barrel to furnace length. The analysis reviewed the system Rijke and Sondhauss natural frequencies of a number of past field installations and noted that as these natural frequencies approached the standing wave axial (length-scale) natural frequency of the furnace, a threshold was identified where vibration occurred.

Furnace:

A trend in the boiler market undoubtedly due to economic forces is that larger boiler units once constructed as stout field-erect designs are now modular package units. Designs are significantly more portable, but less stout with reduced furnace radiant volume. A result is that the occurrence of boiler vibration has increased. With this trend, including design features to assist tuning is very important.

The biggest tuning mitigation device within boiler scope are automated stack dampers. Often these components are installed only if stacks are tall or are common to multiple boilers. To better protect systems from vibration, it is recommended to install draft dampers on all boilers that include FGR or low NO_x heavily staged burners. Dampers should be placed at the lowest point in the stack if possible.

The benefit of a stack damper is that it adds damping to a boiler system between the furnace and stack. Without this device, when flue gas flow is low (i.e. minimal system damping), low frequency pulsation (panting) via Helmholtz resonance has a high probability of occurring. With burner designs that include FGR or staging, flammability limits are reduced, and combustion has weaker impedance to low frequency pulsations, meaning panting is much more common. Closing the stack damper at lower firing rates can eliminate Helmholtz resonance for nearly all combustion systems.

Fan Systems:

Significant resources exist for the design of fans and fans systems [16-17]. These resources along with fan vendor information, provide guidance to avoid flow conditions such as surge or stall. Assuming fan systems follow these guidelines, there are several additional recommendations to assist with vibration mitigation.

1. Include variable frequency drives (VFD) on large systems. VFDs provide ability to change fan speed which alters the fan rotating speed and blade pass natural frequencies. This will greatly increase the tuning ability of the fan system so that resonance frequencies of the system that correspond to fan speed can be avoided. Additionally, many fan issues (e.g. rotating stall), occur at lower airflow rates, VFD modulation of fan speed can avoid these potential issues.
2. On larger boiler systems, always include one modulating damper. It is possible to fully control air flow of the operating range with a variable frequency drive. However, it is recommended to always include a damper in the system. With VFD control only, the VFD is at minimum speed at low flow rates meaning pressure differential across the fan is minimal with minimal damping. Including a flow control damper provides the ability to throttle the damper as needed to increase damping to the system.
3. For systems that contain ultra-low NO_x technology, include a discharge damper between the fan and windbox. Burners of this design type can be strong acoustic amplifiers. Discharge dampers will better isolate the fan from the furnace.

System Structural Design:

Many mechanical structures in combustion systems are designed by maximum deflection. In these cases, the force is calculated using the expected static pressure and cross-sectional area. The force results in deflection and structures are braced to meet a deflection requirement. The downside to this design criteria is that structures that have low expected static pressure could meet the bracing requirement but have a low natural frequency and be susceptible to oscillating pressure forces.

It is recommended to design the stiffening of all system structures to a minimum of 30 Hz.

Case Study

A case study is provided to highlight how the analysis methods and system review approach presented in this paper may be utilized to resolve long-standing and challenging vibration problems with industrial combustion systems.

An industrial client experienced severe vibration on three (3) field-erect boilers over a period of 20 years. Each unit contained four (4) forced-draft burners firing both natural gas and a low-BTU gas fuel. A section view is shown in Figure 8. The low-BTU supply pressure to the units varied significantly. During normal operation, boiler units experienced mechanical vibration at a less than desirable but tolerable level, concentrated on the burner common windbox and the supply ducting of the low BTU fuel.

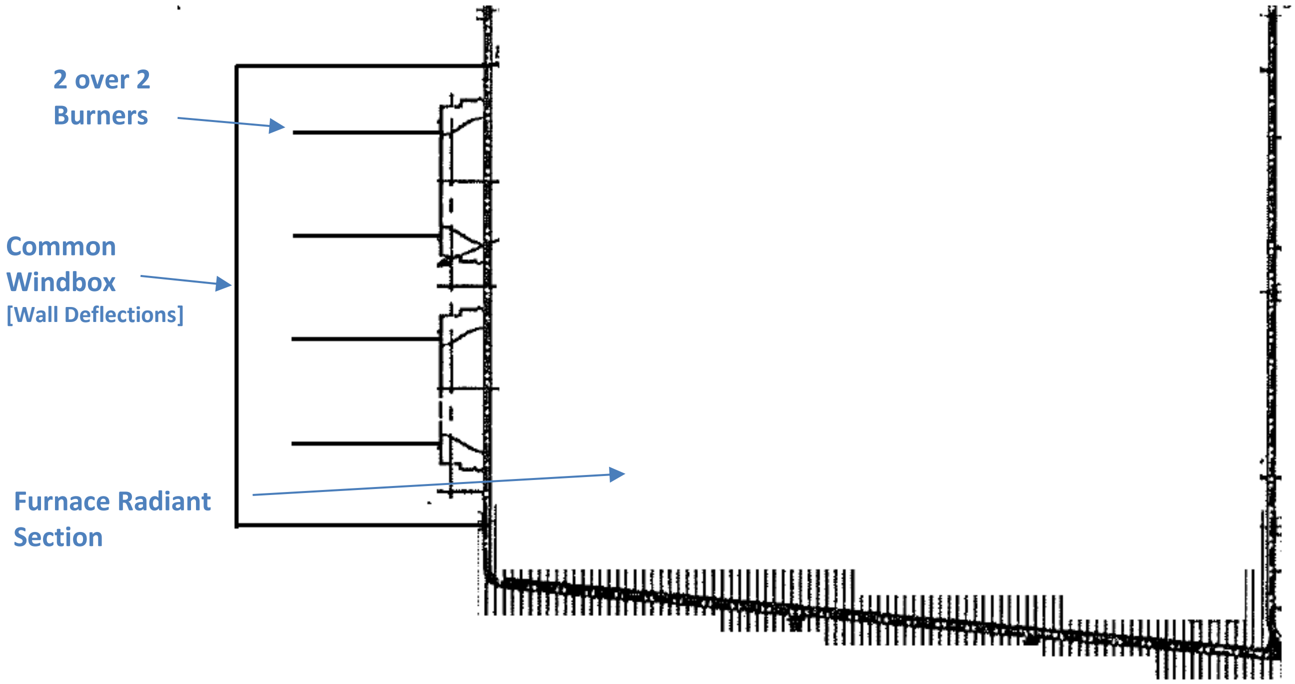

Figure 8 – Large Field-Erect Furnace Cross-section

However, during certain process swings when the Low BTU Fuel supply pressure increased, very severe vibration occurred. Vibration levels damaged ducts and bracing requiring frequent repair. An example of a vibration event is shown with site trend data in Figure 9. This Figure shows how the rapid increase in supply pressure results in very severe acceleration amplitude of the windbox frontplate.

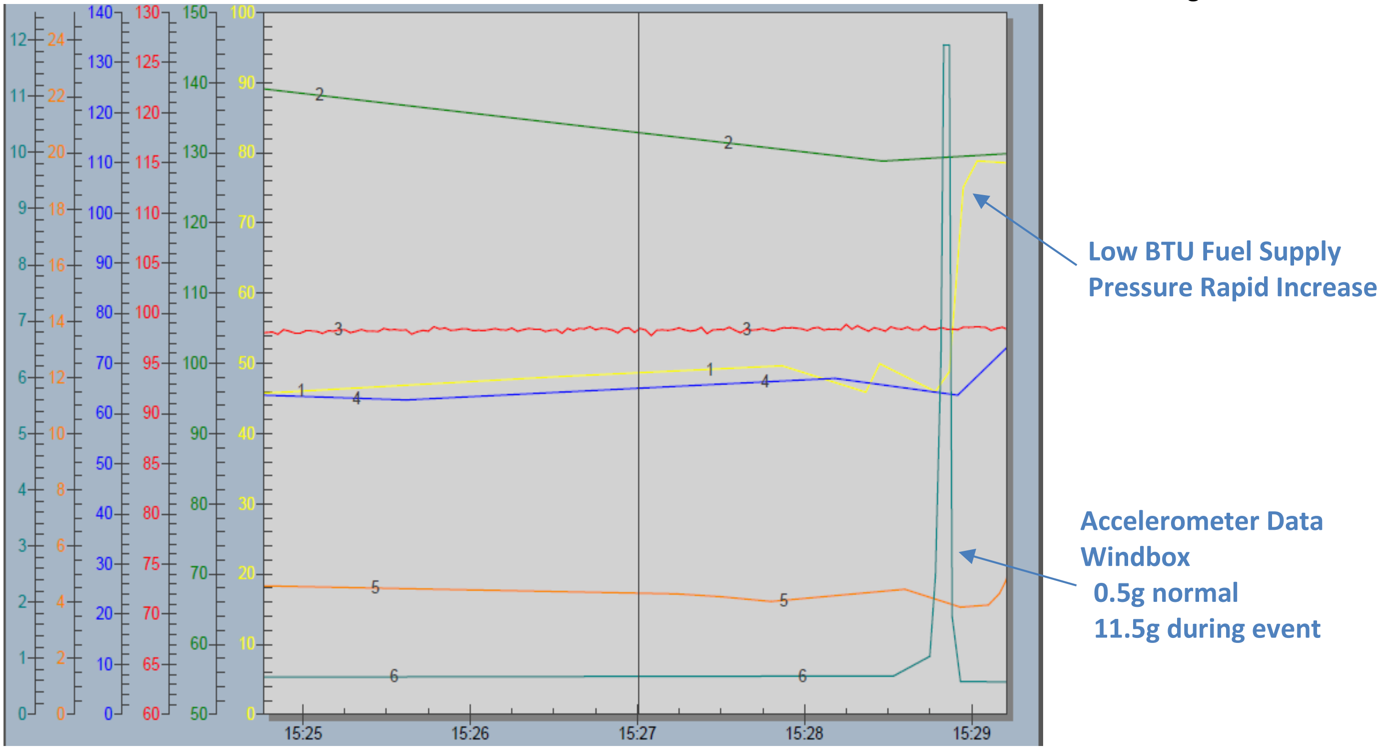

Figure 9 – Site Trend Data during Vibration Event

Early attempts to address the vibration heavily focused on modification to combustion equipment. A rough outline of modifications includes:

Equipment Status	**Result**
Original Combustion Equipment	Severe Vibration on all firing modes
Burner modifications to fuel injection pattern	Reduced vibration on some firing modes; significant vibration on other firing cases
Burner modifications to swirler	No significant change in vibration
Full burner replacements (*new burner vendor*)	No significant change in vibration
Burner modifications to fuel injection pattern	No significant change in vibration

This outline represents a simplified overview of 10+ years of efforts to improve vibration. This overview greatly simplifies the efforts of burner vendors, consultants, and site personnel. Site engineers identified the vibration as a combustion design problem. Since initial burner modifications resulted in significant improvement, this reinforced this concept. After the first burner vendor could not fully resolve vibration events, the site took the major step to fully replace burners with an alternate design. Though there were some minor improvements, overall this change did not reduce frequency of vibration events.

After many years without improvement, site personnel believed the issue could not be resolved and further efforts focused on operation reliability and avoiding further damage through steps such as initiating burner trips during high vibration, replacing auxiliary components with more rugged versions, and scheduling frequent maintenance for inspection and repair.

Vibration Data and Analysis:

Preliminarily, site personnel requested assistance upgrading mechanical design features to be more rugged for increased life between maintenance cycles. However, after a description of the vibration challenges the site faced, it was clear vibration analysis was warranted. A site visit was made, and vibration data was collected. This included windbox and furnace high-speed pressure data, windbox frontplate and low BTU supply ducting mechanical accelerometer data, and high-speed video. Initiating a vibration event was not possible over the testing period; data capture was limited to system load variation.

High speed pressure data [normalized] is shown in Figures 10 and 12 for the Windbox and Furnace radiant sections, respectively. The corresponding processed frequency spectrum data obtained via a Fast Fourier Transform (FFT) is shown in Figures 11 and 13.

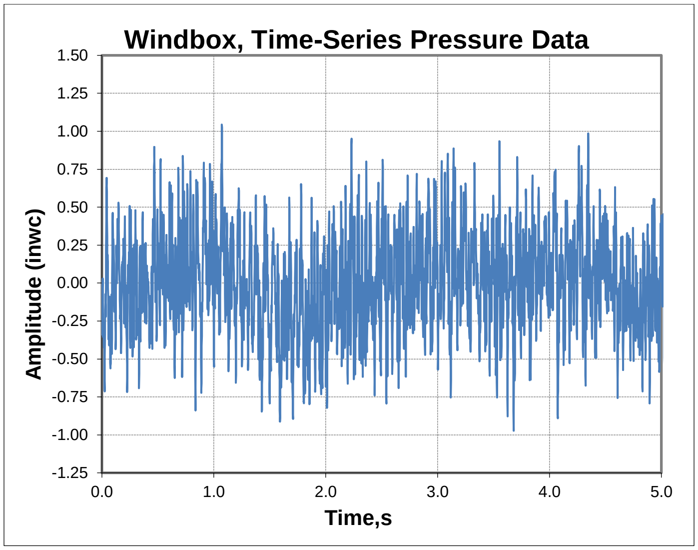

Figure 10 – Windbox Pressure-Raw (Case Study)

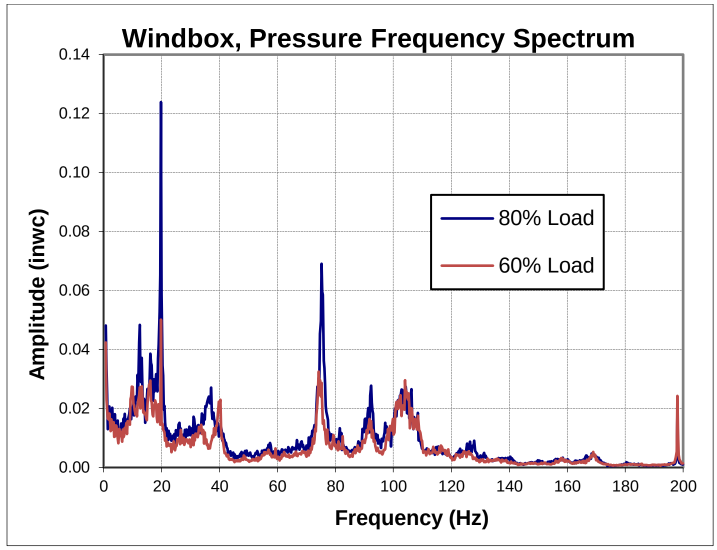

Figure 11 – Windbox Pressure-Process (Case Study)

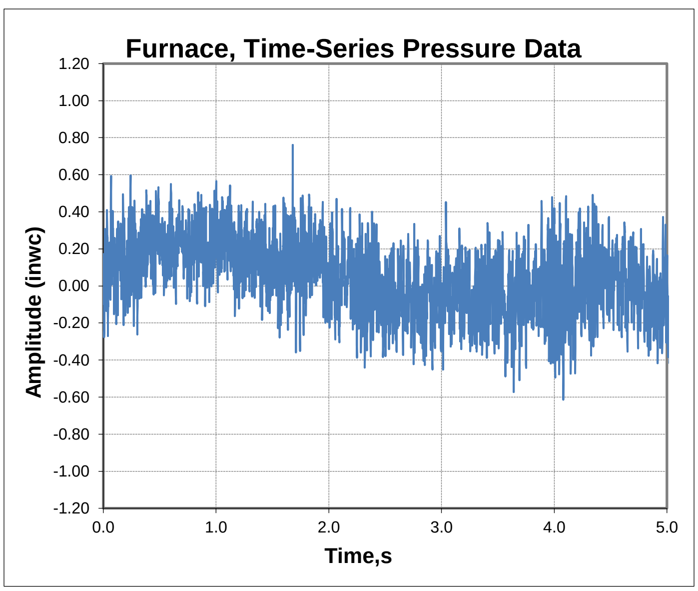

Figure 12 – Furnace Pressure-Raw (Case Study)

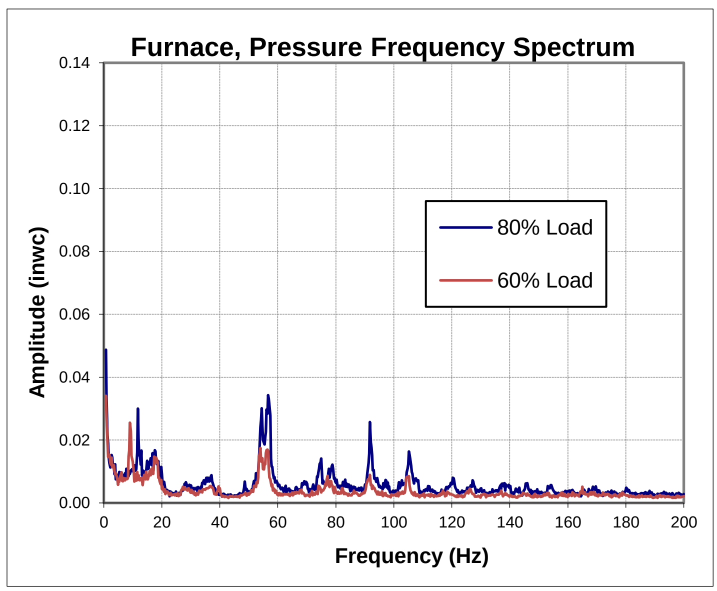

Figure 13 – Furnace Pressure-Processed (Case Study)

Accelerometer data is shown on Figures 14-17:

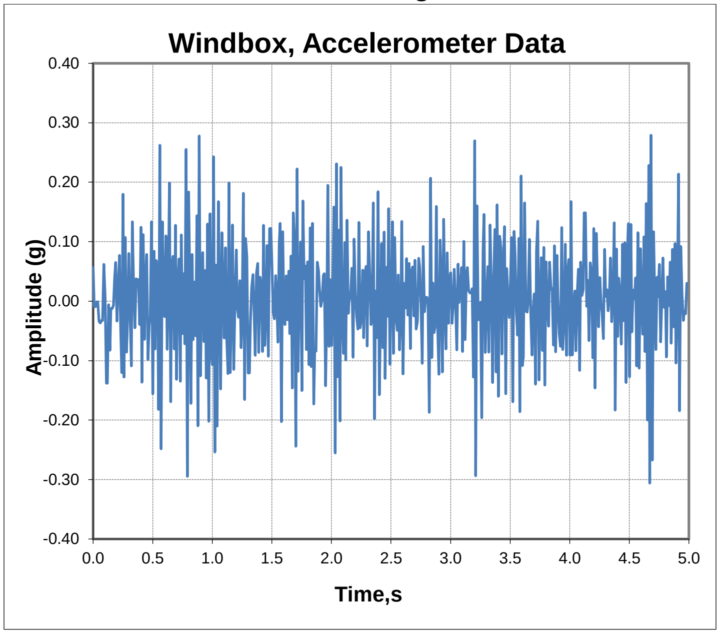

Figure 14 – Windbox Frontplate Acceleration-Raw

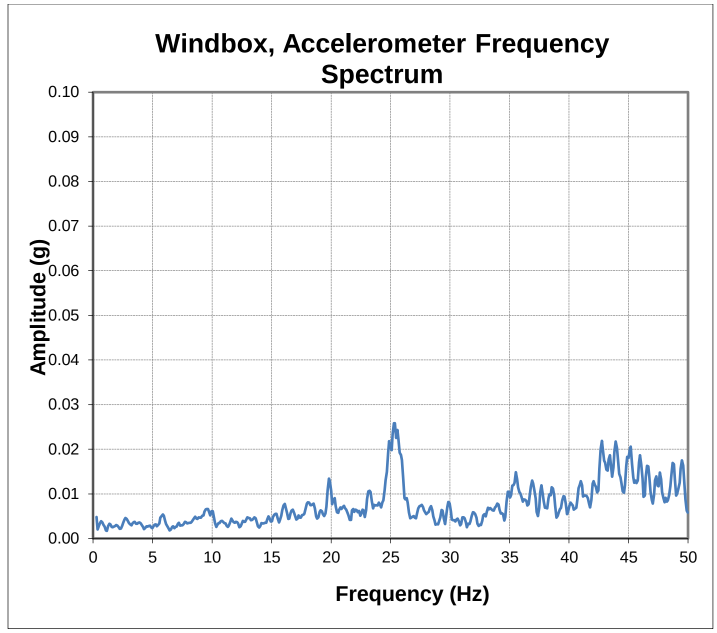

Figure 15 – Windbox Frontplate Acceleration-Processed

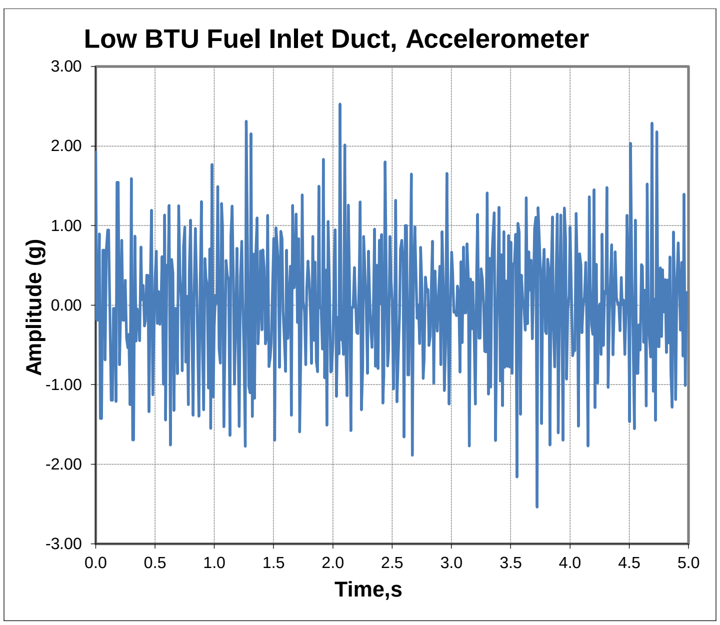

Figure 16 – Low BTU Fuel Supply Duct Acceleration-Raw

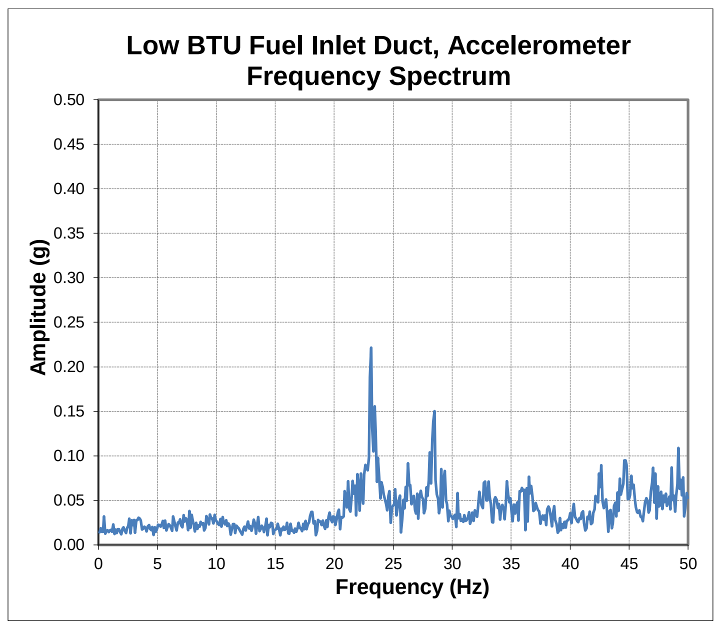

Figure 17 – Low BTU Fuel Supply Duct Acceleration-Processed

Figures 10 and 12 - Observations of raw time-series pressure data:

- Windbox pressure oscillation amplitude is greater than Furnace

Figures 11 and 13 - Observations of processed frequency spectrum data:

- Frequency peaks do not shift with firing rate and therefore are not flow or velocity dependent.
- Windbox data shows sharp frequency peaks at 20 Hz and 75 Hz
- Furnace data does not contain frequency peaks of high amplitude.

Figures 10 through 13 - Comparison of raw time-series data to processed data:

- The Windbox 20 Hz frequency peak is significant. The peak is very sharp and the amplitude (0.125 inw.c.) is on a comparable scale compared to the time-series oscillation of ±0.5 inw.c.
- The Furnace frequency data does not appear to show frequency peaks of any significant consequence.

Figures 14 through 17 - Observations from accelerometer data.

- Windbox frontplate time-series acceleration data is approximately ±0.12g
- The low BTU fuel ducting has much higher vibration compared to the Windbox. The approximate vibration amplitude is ±1.0g ang ±0.15g.
- The Windbox frequency data shows a weak (low amplitude) peak at 25 Hz.
- The low BTU fuel ducting data shows a sharp frequency peak at 23Hz

When reviewing vibration data, an important analysis step is filtering which frequency peaks are real and which may be noise. All vibration data reveals some frequency peaks. Reviewing the peak amplitude compared to time-series data is a good method for this analysis; meaning to classify a frequency peak as significant, the processed data peak amplitude should be of similar scale to the amplitude oscillation of the time-series data. From this review it can be determined that the 20 Hz pressure amplitude in the windbox is a real frequency peak, whereas a comparable peak does not exist within the furnace.

It is clear the system has an Acoustic Resonance (Category 1). As is demonstrated by the Site Trend Data, during a vibration event the system mechanical vibration amplitude increases rapidly by a factor of greater than 20. Under certain process conditions related to high fuel flow, this large system resonance emerges. Thus, the first steps for mitigation is to determine the most likely acoustic driver and perform system natural frequency analysis for resonance.

Pressure frequency spectrum data shows the biggest peak occurs at 20 Hz. This peak is measured in the windbox and attenuates as it passes to the furnace chamber. It is known that this frequency peak appears at variable firing rates, and it is higher amplitude at higher flow rates.

The forced draft fan is very large with a rotating speed of 1200 RPM, equivalent to 20 Hz. Since the 20 Hz is noted in the Windbox, but not the furnace, it is very likely the forced draft fan is the primary driver.

Compared to typical industrial fan systems, the fan rotating speed pulsations were unusually high amplitude. Analysis revealed this could potentially be caused by a precessing vortex core (PVC; an unstable inlet swirl) on the fan inlet. Additionally, the system contained minimal damping between the fan discharge and furnace system. The fan system contains only an inlet damper (no outlet discharge damper) for airflow control.

Observations from site personnel and trend data strongly suggest the primary process change that leads to a vibration resonance event is tied to a surge of low-BTU gas flow.

The increased flow had the following impact on the system:

- Increased discharge velocity.
- Increased burner net swirl. Since the fuel injection scroll was a highly swirled zone, increased velocity caused higher net swirl impacting flame shape.
- Decreased air/fuel ratio (stoichiometry) for combustion. The flow increases faster than the combustion air can compensate.
- The furnace radiant temperature increases.

The natural frequencies of the system were reviewed. This analysis identified the furnace length as a strong candidate for resonance. Based on an estimated furnace temperature (and corresponding speed of sound), the standing wave natural frequency is 40 Hz. This is a node of 20 Hz.

For the fan speed pulsations to couple with the furnace, the pulsations must pass through the combustion process. Flames that are more intense OR unstable tend to be stronger amplifiers, while flames that are more staged tend to be weaker amplifiers.

Analysis and data observation strongly indicate the increased flow drives combustion into a mode of stronger amplification. This could cause a global change in the flame by one of two possible scenarios:

1. The flame becomes more strongly anchored. This is due to increased swirl which intensifies combustion.
2. The flame becomes unstable: This could occur if the low-BTU flow increases so rapidly that the burner registers operate sub-stoichiometric (insufficient air to burn all fuel).

During normal operation, combustion acted as a weak amplifier and the furnace standing wave did not acoustically couple with the 20 Hz acoustic driver. The low-BTU duct was weakly braced with a low natural frequency. Since the driving pressure oscillations were high amplitude under normal conditions, the result was weak vibration on the common windbox, and much larger vibration on the fuel supply ducting.

During high vibration events, due to either combustion and/or the increase in furnace temperature, a furnace standing wave acoustically coupled with the 20 Hz driver resulting in a very strong resonance. For the weakly braced fuel supply ducting, vibration levels were so extreme that ducting failure could occur rapidly, and site personnel were forced to trip the unit to avoid damage.

Resolution:

Several mitigation methods were suggested, with the primary target identified as the lack of damping between the fan discharge and furnace. Independent of the final sequence that initiates the resonance, data and system analysis provides guidance that by dissipating the fan rotating-speed pulsations, the resonance can be avoided.

To meet this need:

1. A modulating flow control damper was added to the fan discharge ducting in a location identified to best damp the low frequency pulsations.
2. The low-BTU fuel supply duct was redesigned to both stiffen the structure and better isolate the ducting mechanically from the windbox structure.

The result was a significant improvement in system reliability and maintenance with no reports of excessive vibration following these retrofits.

Case Study Summary:

At first glance, the system analysis and resolution for this case study may seem obvious. However, it is important to note the challenges of this installation. From a site perspective, operators and engineers identified this as a combustion issue requiring combustion equipment changes to resolve. No significant vibration existed during fan-only operation, and therefore changes to the fan system did not seem necessary. The vibration problems on the unit had persisted over 20 years through two (2) burner designs of different vendors and following several vibration consultant evaluations. Site personnel had grown accustomed to the vibration as an integral component of their equipment operation.

A highlight of the benefit of the analysis methodology presented was that successful resolution did not require identifying the exact forcing function driving resonance. Rather, analysis of the vibration data and system layout revealed mitigation opportunities of the greatest probability of success.

This case study also highlights the potential fallacy of addressing vibration issues by following direct observations. That is, site personnel observed vibration only during combustion modes and the vibration was concentrated on the windbox and burner. Therefore, it was believed changes to these components were required; whereas, the successful mitigation strategy added a component (the modulating damper) to a section of the system that experienced very minimal vibration.

Conclusion and Recommendation

Combustion thermoacoustic vibration remains a potential operational challenge that may be encountered on all combustion systems. Vibration can delay commissioning of equipment and lead to premature component failures.

Despite many years of research and the advent of modern computing techniques, accurate predictive techniques are not yet available to identify all vibration modes of modern industrial combustion systems. This is due to the chaotic and complex nature of the combustion effect on thermal and pressure gradients, coupled with most industrial combustion applications including large-scale and customized designs.

Research and lessons learned from field experience have identified design best practices that can be employed to minimize vibration potential, and better equip system designs with mitigation adjustment devices and tools.

When vibration occurs, the necessity to collect extensive vibration data is outlined including recommendations for testing equipment and system locations, along with simplified analysis methods for time-series and frequency band data. Common acoustic drivers and resonant sources are discussed. Since accurate modeling of system forcing functions can be challenging, it is argued a simplified analysis method is better suited for fast resolution. With this method vibration data and system analysis are used to sort vibration modes into general categories to better refine areas of interest and identify components and/or system features of greatest concern. By targeting these areas, mitigation techniques can be employed that provide higher probability of resolution.

A case study is provided to highlight how the simplified analysis method can be used in practice. This study demonstrates that challenging vibration problems can be resolved with non-intuitive mitigation techniques based upon a framework of data collection, data analysis, and system modeling.

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