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THERMOACOUSTIC VIBRATIONS IN INDUSTRIAL FURNACES AND BOILERS

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ABSTRACT

Industrial boilers and furnaces occasionally suffer low-frequency vibrations generated by a dynamic feedback process between the burner (or burners) and acoustic modes in adjacent gas-filled cavities in the main combustion chamber or connecting ductwork. This occurs when pressure pulses associated with acoustic resonances propagate to the burner so that they are in phase with combustion rate fluctuations caused by turbulence and reaction dynamics. When these pressure pulses become sufficiently phase-synchronized with fluctuations in heat release from the flame, the forces that normally dissipate the pressure waves are overwhelmed and an amplifying feedback loop is created. In the literature, such oscillations are referred to as thermoacoustic oscillations or 'rumble,' and their basic physics have been the subject of numerous investigations for well over a century. Unfortunately, rumble amplitudes can be large enough to negatively impact thermal efficiency and emissions, and the associated mechanical vibrations they cause can even lead to structural damage. The potential for rumble poses a significant challenge to combustion engineers because it is often very difficult to predict and can be associated with a large number of design and operating factors such as fuel quality, burner swirl and staging, induction and draft fan characteristics, ducting design and combustion cavity shape. The underlying relationships involved are sufficiently complex that it is possible for two apparently identical boilers or furnaces to exhibit completely different rumble tendencies. In this study, we review the published information currently available about the causes and suppression of burner rumble and suggest possible opportunities for improving its prediction, diagnosis, and active control.

Objectives

The goal of this study is to summarize what is currently known about the causes of thermoacoustic vibrations in industrial natural-gas-fired furnaces and boilers and to identify opportunities for enhancing diagnosis and remediation. To accomplish this, our approach includes:

- A brief review of recent literature on thermoacoustic vibrations;
- A summary of how such vibrations are typically manifested in industrial applications;
- Preliminary analysis of experimental field data from a recent case study;
- Plans for future work.

Limitations and Scope

The experimental data included in this study was generated without the benefit of controlled experiments on industrial-scale units. Instead, field measurements were collected from a commercial boiler to illustrate an actual case of industrial relevance, its quantitative characteristics, and how it compared with expected patterns. Future experimental field tests are being planned for this and other units that will provide additional data to improve basic understanding of the problem and support improved designs and/or operating strategies. We specifically do not address the problem of flow-induced vibration here, which can also occur in industrial furnaces and boilers.

INTRODUCTION

The source of thermoacoustic vibrations (rumble) in industrial furnaces and boilers is distinct from flow-induced vibration and is more closely related to the 'singing flame' phenomenon in heated tubes, which has been studied in various forms since the 18th century [Richardson (1922)]. Thermoacoustic vibrations always require the presence of two key features: 1) a large temperature gradient, and 2) a gas-filled cavity capable of supporting Helmholtz resonances [Chanaud (1994), Balasubramanian & Sujith (2008), Eisinger & Sullivan (2002, 2008)]. The underlying physics of this phenomenon was originally explained by Raleigh [(1878)], who noted that the key requirement for sustained vibrations is met when the rate of change in pressure in the vicinity of a flame or hot surface becomes directly correlated with the rate of heat input (either via reaction or transfer from a hot surface). More recently, Nicoud and Poinsot [(2005)] suggested that the requirement for vibrations in combustion systems is also met when the rate of temperature rise becomes directly correlated with the reaction rate. In either case, the key requirement is to have synchronized phasing between the rate of heat input and the rate of gas compression/expansion.

In some of the most definitive studies to date, Eisinger and Sullivan [(2002, 2008)] showed that large temperature differentials between the cold burner inlet air and the hot combustion gases can lead to such phase synchronization in industrial-scale furnaces and boilers. Other key factors are the lengths of the burner cavity and the hot gas zones, which provide the necessary spatial domains and boundary conditions for the development of standing acoustic (pressure) waves that can be excited by input heat energy. Two common standing-wave patterns that originate in the burner are referred to as either the Rijke or Sondhauss modes, which are named after their original experimental discoverers [Balasubramanian (2008), Rott (1983), Eisinger & Sullivan (2002, 2008)]. Synchronization occurs when there is constructive interference between these standing waves and downstream resonances in the post-burner hot zone. When this happens, thermal energy from combustion is converted into acoustic energy. Under normal operation, furnaces and boilers typically have large temperature differentials between the hotter product gases and the cooler reactant air, thus there is a large thermal energy source to sustain these oscillations once they begin.

Eisinger and Sullivan presented a map that delineates operating and design regions where synchronization is likely in furnaces and boilers. As shown in **Figure 1**, two dimensionless parameters, representing the key geometric and temperature factors, respectively, are used to determine whether any given system falls into the stable (non-vibrating) or unstable (vibrating) region. Typical load isotherms are plotted relative to these regions in **Figure 1** for a wide range of system geometries. As can be seen in the figure, thermoacoustic instability tends to increase with load for many different geometric configurations.

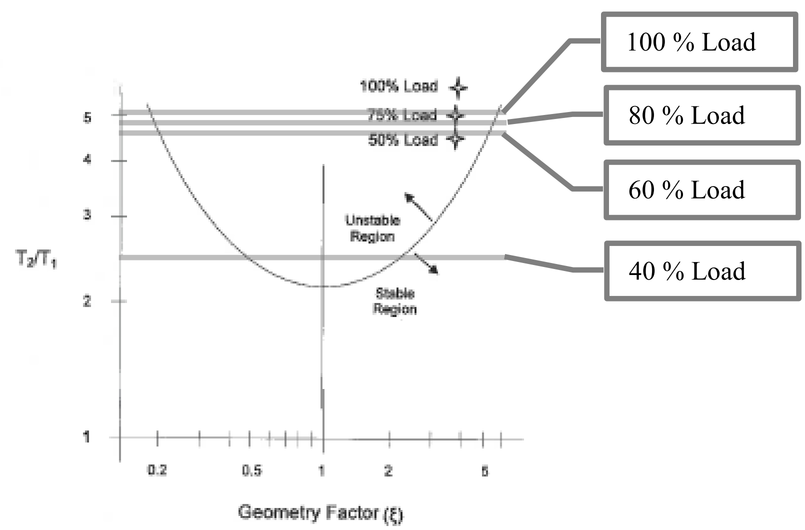

Figure 1. Thermoacoustic vibration stability curve [Eisinger and Sullivan (2008)] with typical load isotherms.

Often the key geometric factors responsible for thermoacoustic vibrations in a specific unit are difficult to identify because of the complex interaction of firebox and duct design characteristics that can be involved in the standing wave patterns. Rumble also often appears during specific types of operating transients or load states (typically at intermediate or part load states), so it may not always be observed until after a unit has been initially constructed and tested. This intermittent nature can also make rumble difficult to characterize and correlate with operational events. It is not uncommon for two "identical" boilers to be just different enough such that one unit will exhibit vibration while the other will not at apparently identical operating conditions. Much work has been done to characterize thermoacoustic vibration in combustion systems, but slight variations in geometry and structural rigidity can still allow unanticipated vibration to occur.

Fuel variations can also be involved in the appearance of rumble. Many industrial boilers have a single burner that is designed to fire a variety of fuels, including natural gas, #6 or #2 oil, syngas, biogas or waste gas depending on which fuel is most readily available at the lowest cost. Although natural gas is currently the lowest cost fuel, its composition can vary significantly depending on the source, and non-price driven supply issues can also result in fueling shifts. Whatever the cause, the specific combustion properties of different fuels can have a significant impact on combustion and heat release and thus promote or suppress the onset of rumble.

When rumble occurs, it is sometimes possible to make operational adjustments (e.g., changing burner settings or avoiding certain firing rates) that directly improve the burner or acoustic parameters themselves or that suppress the interactive feedback. In many cases, however, operating adjustments are not sufficient to solve the problem, and it may only be possible to take measures that reduce the resulting vibrational damage (e.g., adding mechanical stiffeners to vibrating components) rather than removing the original source of the problem. Ideally, if our understanding of the rumble phenomenon is sufficient, it should be preferable and less costly to remove the underlying causes by more effectively avoiding certain design features and/or developing and implementing more effective active control measures that suppress feedback. For the latter, it might even be possible to implement such controls post factum in existing units. In the following sections, we summarize the information currently available in the literature about both of these possible approaches. We also believe it is important to consider methods that might be used for diagnosing rumble in individual cases to better identify root causes and thereby reduce the cost and time associated with resolving the problem. We illustrate possible approaches for the latter with example field data collected from a boiler experiencing rumble.

BACKGROUND

Thermoacoustic vibrations can occur in a wide range of combustion systems operating in both fuel-lean, or fuel-rich conditions, as long as the conditions necessary for constructive interference between heat input and acoustics are met. Fuel-lean conditions are often chosen to minimize NO_x emissions in combustion turbines, and these tend to shift the main heat-release zone upstream into the burner throat while also reducing the temperature of the post-burner gases. Both of these changes affect the relative locations of the heat release and acoustic (pressure) waves. Excessively lean operation can also lead to flame lifting, which leads to transient shifts in the flame location and length that cause jumps back and forth across the stability boundary in **Figure 1**. Fuel-rich conditions coupled with air staging are often chosen for industrial boilers to minimize NO_x emissions, but these can also lead to longer and cooler flames, shifts in sound speed, and corresponding shifts in acoustic wave lengths.

Gas Turbine Lean Combustion

Although combustion turbines typically employ premixed combustion rather than diffusion flame combustion, relevant insight into the mechanisms of thermoacoustic instabilities for industrial furnaces and boilers can be realized. Bellows et al. (2006) studied vibration in a gas-turbine combustor simulator. Lean, premixed combustion was studied in a turbulent, swirl-stabilized flame. They mapped the response over a range of input driving frequencies and amplitudes. The driving frequency was varied through the resonant combustor frequency. They concluded that combustion dynamics was controlled by the thermoacoustics rather than gas dynamics.

Yao et al. (2012) predicted rumble as pressure fluctuations in the combustion chambers of aeroengines and combustion gas turbines by integrating computational fluid dynamics with a one-dimensional linear stability analysis. They predicted modes of pressure oscillations, frequencies, and growth rates. Linear acoustic theory was used to predict the propagation of acoustic waves both upstream and downstream of the combustion chamber. Experiments were conducted to show that combustion oscillations would occur at the predicted frequency.

Rodriquez-Martinez et al. (2006) studied instabilities in a partially premixed swirl burner and furnace test rig. The testing also included the effect of the inlet air duct geometry on the instabilities. Low-frequency (30 to 100 Hz) instabilities were characterized. Unstable pressure amplitudes greater than 5% of the mean pressure were observed. They concluded that the instabilities result from excitation of the longitudinal modes of the air supply ducting which formed the dominant acoustic geometry of their test rig. They note that their test rig exhibits similar behavior to that observed in pulse combustors, dump combustors and gas turbines where the flame dynamics are strongly coupled to upstream modulations in air flow. This could also be true for industrial natural-gas furnaces where furnace dynamics can be influenced by the dynamics of the forced-draft fan, control dampers or duct geometry. The length of the inlet duct had a significant impact on the pressure and velocity oscillation modes in the combustor. Generally, the velocity lagged the pressure by one-quarter cycle which they note is indicative of standing waves.

Acharya et al. (2013) analyzed premixed swirl flames subject to helical flow disturbances. The study was motivated by combustion instabilities and combustion noise. Combustion instabilities occur when unsteady heat release couples with one or more of the combustor's acoustic modes. High-amplitude pressure and velocity oscillations can occur. The analysis shows that helical modes can influence the flame "wrinkling" amplitude and heat release fluctuations differently. The instability parameters exhibit variations in sensitivity to swirl number and other flame parameters. This paper provides a good theoretical analysis of combustion instability.

Huang and Yang (2009) provide a comprehensive review of the dynamics and stability of lean-premixed swirl-stabilized combustion in gas turbines. The review discusses both mechanical design and operation.

Pulse Combustion in Gas Turbines

Whitelaw (1997) studied oscillations in gas-turbine combustors. The fuel and air feeds were pulsed at frequencies ranging from 75 to 200 Hz with the purpose of reducing NO_x emissions. Oscillation of the flow of natural gas led to increases in the noise levels as large as 6 dB. NO_x reductions in the range of 8 to 17% were obtained.

Benelli et al (1992) performed modeling and numerical simulation of pulse combustors. A Helmholtz-type mechanical valve pulsing combustor was simulated. The model captured the unstable feedback loop in sequential steps of valve opening, injection of reactants, turbulent mixing and combustion, etc. Simulation results are presented.

Natural Gas Industrial Burner Combustion

Sams and Jordan (1996, 1997) investigated natural gas burners for firetube indirect heaters. They show with test data that when the burner nozzle velocity is sufficiently greater than the firetube velocity, the low-frequency rumble decreases. Field data was used to construct relationships between the burner noise level and the gas volume expansion ratio, burner air-to-fuel ratio, mixture flow rate, natural gas orifice velocity, burner area, and the number of burners. They developed a correlation to predict burner noise. Burner rumble will occur when the velocity head ratio of the burner nozzle to combustion chamber is outside the range of 7 to 50.

Iordache and Catalina (2013) performed sound level measurements on two small (770 kW) commercial building heating boilers over the entire boiler operating cycle. Somewhat unexpectedly they found that the highest sound levels were recorded during the ventilation period of the boiler cycle before gas injection rather than at the maximum thermal load period. This indicates that air flow dynamics can be a plausible source of vibration and noise.

Kodis, Webster and Dirks (2002) reported an instance where a dirty air heater was the cause for furnace rumble.

Goldring (2007) used a computational fluid dynamics model to illustrate flame detachment as a function of combustion parameters. The modeling was used to show that the planned burner modifications, which included new staged flame stabilizers, low-NO_x gas lances and low-NO_x atomizers would produce stable flames over the load range and with a wide range of flue gas recirculation (FGR). Combustion instabilities in industrial furnaces could arise in low-NO_x burners as the combustion limit is approached due to fuel-rich conditions.

Eckstein et al (2004) studied the role of entropy waves in low-frequency oscillations for a diffusion burner.

Potential Corrective Measures

There are two accepted approaches to mitigate vibration in industrial boilers or combustion turbines. The equipment can be mechanically altered to reduce or eliminate the vibration. Adaptive or passive control techniques can be used to alter the combustion process to eliminate or reduce vibration.

Eisinger and Sullivan (2002, 2008) suggest simple changes to the inlet configuration of the burner to address furnace rumble by slightly altering the operation of the burner relative to the stability line and moving the operation into the no-vibration region.

Berkau et al. (1990) use natural gas co-firing to stabilize a deeply staged, low-NO_x coal burner. By introducing natural gas into the primary air line with the coal, the combustion instability is reduced over the load range and better attachment is achieved. By eliminating the fluctuations due to ignition and extinction at the burner front in a deeply staged burner, furnace rumble is eliminated.

Lifshits et al. (1994, 1995) eliminate combustion instability in natural-gas industrial burners by more uniformly distributing the natural gas and primary air. This is not a premixed burner. They distribute the gas through small holes along radial slots in the primary zone. Since the gas and air are mixed radially, local oscillations of the flame front occur at different frequencies and therefore do not synchronize, which dampens vibration and resonance problems. This approach was specifically invented to address furnace rumble due to staging the combustion in low-NO_x burners to reach ultra-low NO_x emissions.

Hersh and Walker (1997) developed a fluid mechanical model of the Helmholz resonator for application to jet engines. The model "predicts impedance as a function of the amplitude and frequency of the incident sound pressure field and resonator geometry." The purpose of this work was to quieten jet engines.

Han (2008) used a tunable Helmholtz resonator to dampen oscillations. Both the cavity volume (piston) and throat plate (exchangeable) could be adjusted to alter the resonant frequency. This work provides a nice presentation of calculations of Helmholtz resonance frequency. The Helmholtz resonator was affixed to the side of a cylindrical duct at the end of which was a horn that could be tuned to a range of frequencies. This is an example of adaptive tuning.

Candel (1992) employs active control in a pulse combustor. It is shown that in the presence of pressure waves, large-scale motions drive the instability. The paper describes the processes that drive the flame dynamics such as hydrodynamic instability, vortex roll-up, vortex interactions, front and reacting stream pulsations, periodic extinctions and reignitions, and self-acceleration. The paper reviews the basic principles and the initial demonstrations of active instability control of pulse combustors. The application of adaptive control methods is discussed in more detail.

RESULTS FROM A RECENT B&W BOILER CASE STUDY

Recently, Babcock &Wilcox (B&W) personnel have observed thermoacoustic vibrations first-hand on several industrial boilers. To better understand the problem in a specific case, field measurements were collected from a small natural gas-fired industrial boiler that was experiencing vibrations of an apparently thermoacoustic origin. Multiple types of instrumentation were used to measure the vibrations including: microphones; high temperature accelerometers; IEPE (Integrated Electronics, PiezoElectric) accelerometers; and dynamic pressure transducers. These sensors were installed throughout the flow path as shown in the plan view in **Figure 2**, beginning at the air inlet, then the furnace, followed by the generating bank and the boiler outlet/economizer.

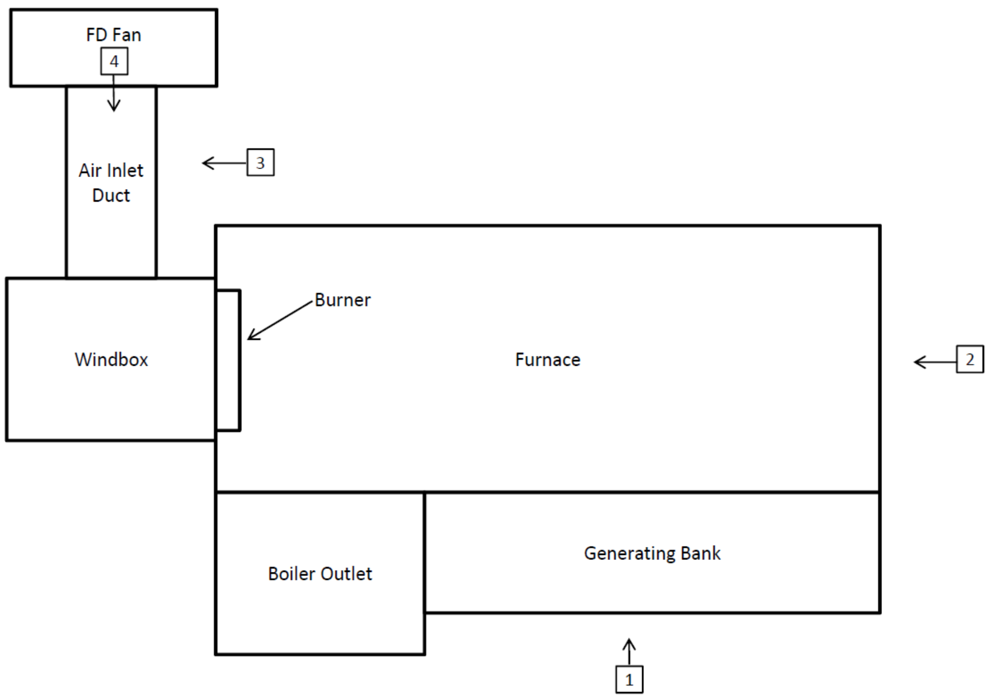

Figure 2. Simplified boiler arrangement and layout of field instrumentation.

During the measurements, the boiler was operated in the load range where significant vibrations were known to occur. Performance parameters of the boiler, such as air mass flowrate and density, were adjusted to determine what impact each of the parameters has on the boiler pulsation. The only adjustment that appeared to impact the boiler pulsation was increasing or decreasing the firing rate beyond the pulsation range.

The characteristic frequency spectrum measured during the furnace pulsation appeared to be a low-frequency, high-amplitude peak with a sawtooth shape, indicating nonlinearity in the signal. While the amplitude was significantly reduced outside of the pulsation range, the peak frequency remained the same regardless of firing rate.

The field data were further analyzed to determine the source of the thermoacoustic vibration, and to determine which parameters impact the vibration. The signals were compared and studied to identify whether geometric characteristics or combustion characteristics, or a combination of both, have the most impact on the vibration. Of specific interest is the development of a design specification that would optimize the design of industrial boilers and reduce the potential for thermoacoustic vibration in the future.

ANALYSIS OF THE B&W FIELD DATA

As a preliminary step in analyzing the B&W field case data, we analyzed the basic time-series characteristics of the microphone time series to resolve the general trends they revealed about the vibrations. These analyses included:

- Signal oscillograms — Depiction of the qualitative patterns in the raw signals
- Power spectral density functions — Measure of dominant frequency content
- Temporal irreversibility (T_3) functions — Measure of temporal asymmetry in signal oscillations
- Cross-correlation functions — Measure of linear correlation over finite timescales

Future analyses are planned that will include more sophisticated information-theoretic functions such as transfer entropy to quantify the degree and sense of information flows between measurement locations.

ANALYTICAL RESULTS FOR THE FIELD DATA

Figure 3 illustrates the temporal patterns detected by the four acoustic transducers (microphones) at the locations depicted in **Figure 2** for a test in which the boiler unit transitioned from rumble to normal operation and back to rumble during a series of load changes. We examine three 20-s segments of the test, as denoted on the figure, where the burner firing rate was nominally the same, with the only difference being the spontaneous transition out of and back into rumble. On large scales, the signal envelopes are quite variable, representing overall sound-intensity changes in the boiler unit. Additionally, the long-term drift in the envelope, as seen from 0 to 120 s, is visible and represents a long-term controlled change in firing rate. At about 120 s onward, the firing rate is nominally the same, and changes largely reflect uncontrolled, spontaneous system responses.

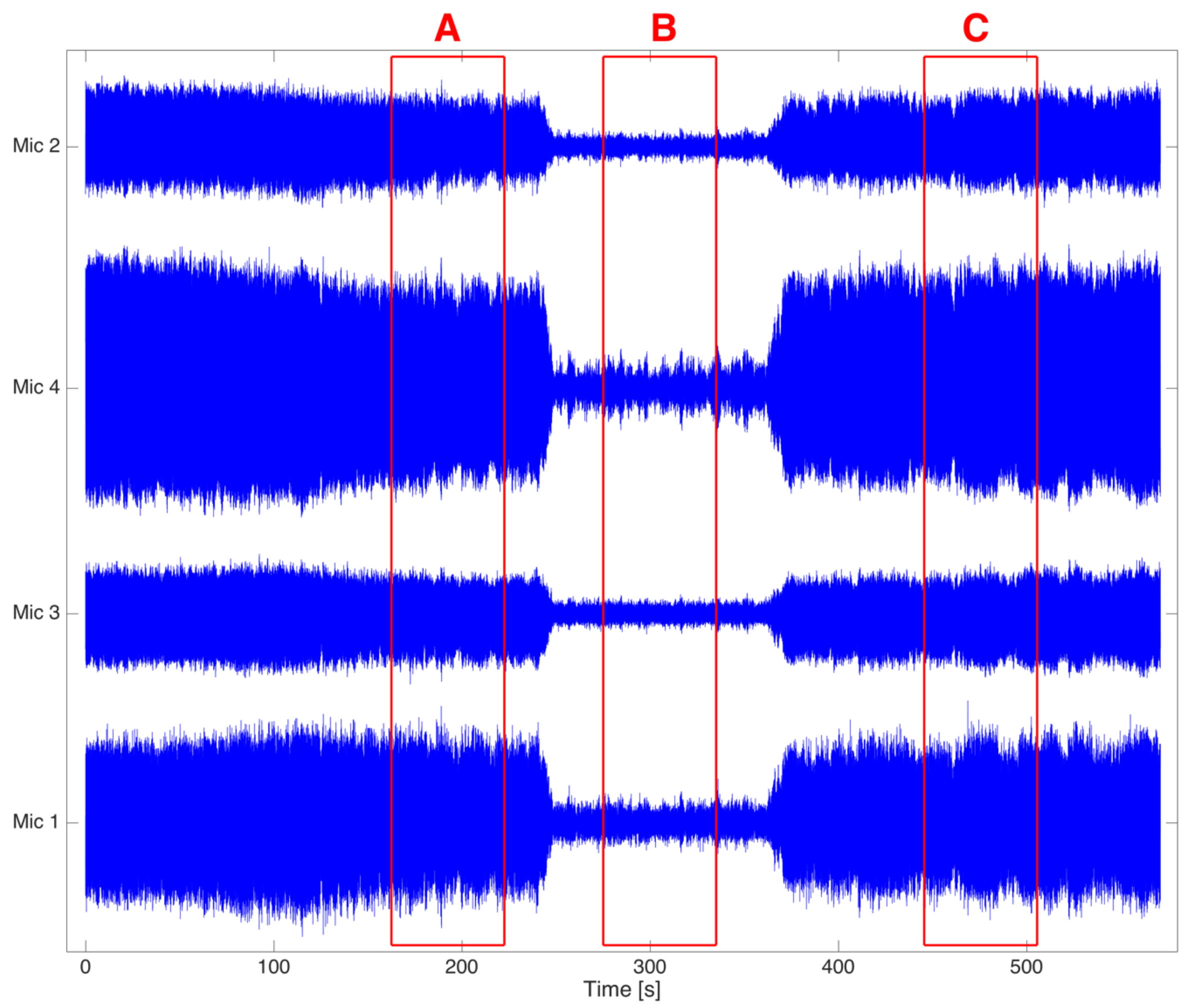

Figure 3. Complete time records for the microphone signals, from top to bottom: firebox (2); FD fan area (4); inlet duct (3); generating bank (1). All time series are plotted on the same scale. 60-s segments used in later analysis are marked by the red boxes and lettered for reference. Microphone numbers are from Figure 2.

On small timescales, however, distinct differences in the acoustic signal at each location are noted, as depicted in **Figure 4**. In the rumble condition segments, a dominant oscillation of about 10 cycles per second is observed, with higher-frequency content superimposed to varying degrees, depending on measurement location. The signals have very similar characteristics in both rumble segments, even after having spent several minutes in "normal" operation in the interim. The normal-condition signals do contain a discernible large-scale oscillation frequency of about 10 cycles per second, but the higher-frequency signal content is over equal or greater magnitude. The boiler vibrations, as measured by the acoustic sensors, do appear to resonate at a natural system frequency.

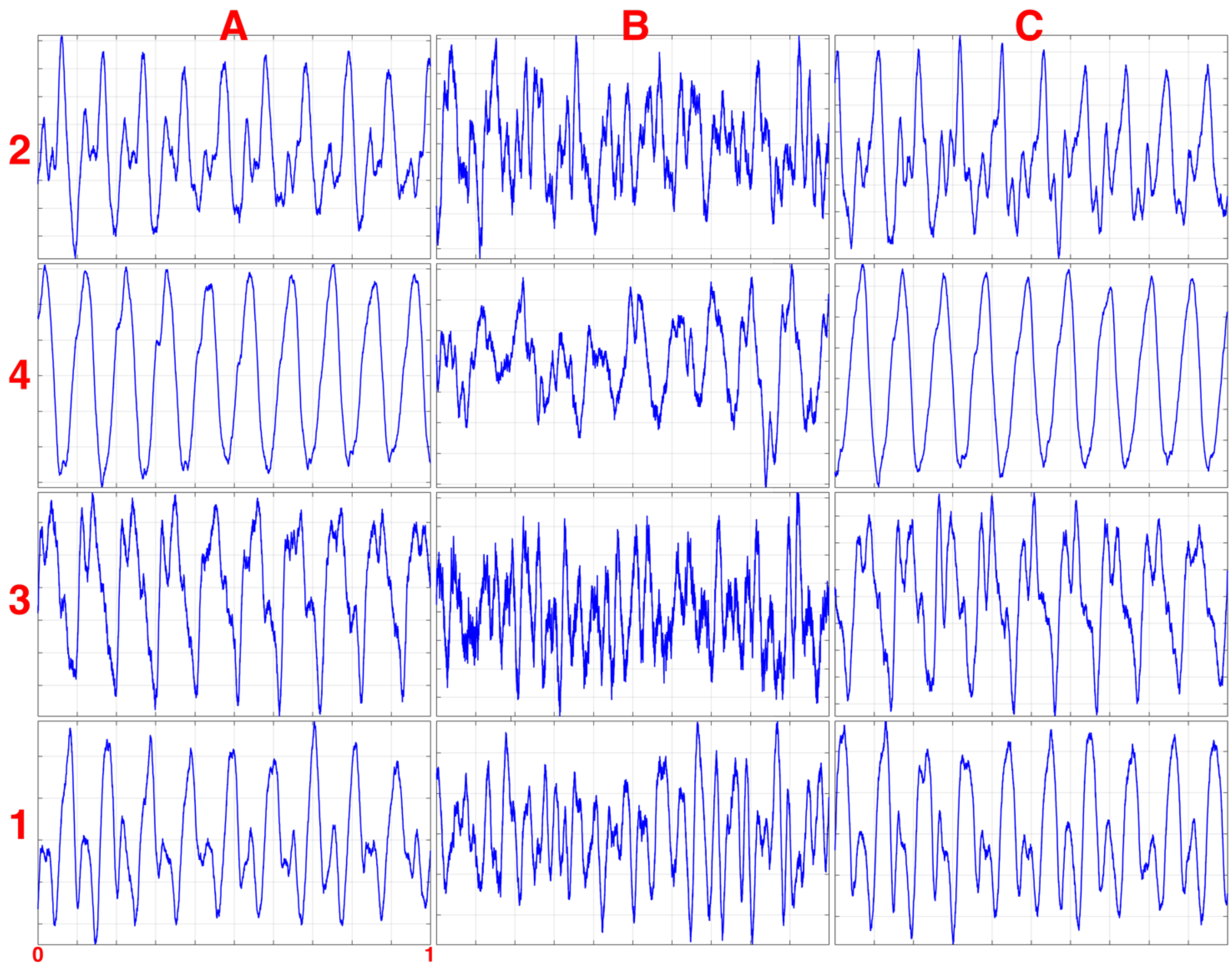

Figure 4. Time-series segments for the microphones at three conditions: rumble (A), normal (B), rumble (C). All segments encompass 1 s of time on the abscissa and are scaled to the segment signal range on the ordinate.

These observations regarding the relative frequency content of the signals is reflected in the power spectral density (PSD) functions, as shown in **Figure 5**. In this figure, the plots' axes are shown on the same scale, with $\log_{10}$ of power on the ordinate and frequency ranging from 0–100 Hz on the abscissa, with the 10 Hz frequency highlighted. For all signals, there is a distinct dominance of the 10 Hz oscillation. In the rumble cases, sharp harmonics of 10 Hz are pronounced, suggestive of the coherence in the signals at the primary harmonic. In the normal case, the harmonics are not significant, but other broadband frequency content, not necessarily at multiples of 10 Hz, is present within 1–2 decades of the 10 Hz content, suggesting a more complex signal dominated by higher-dimensional causes than observed in the resonance seen in the rumble condition.

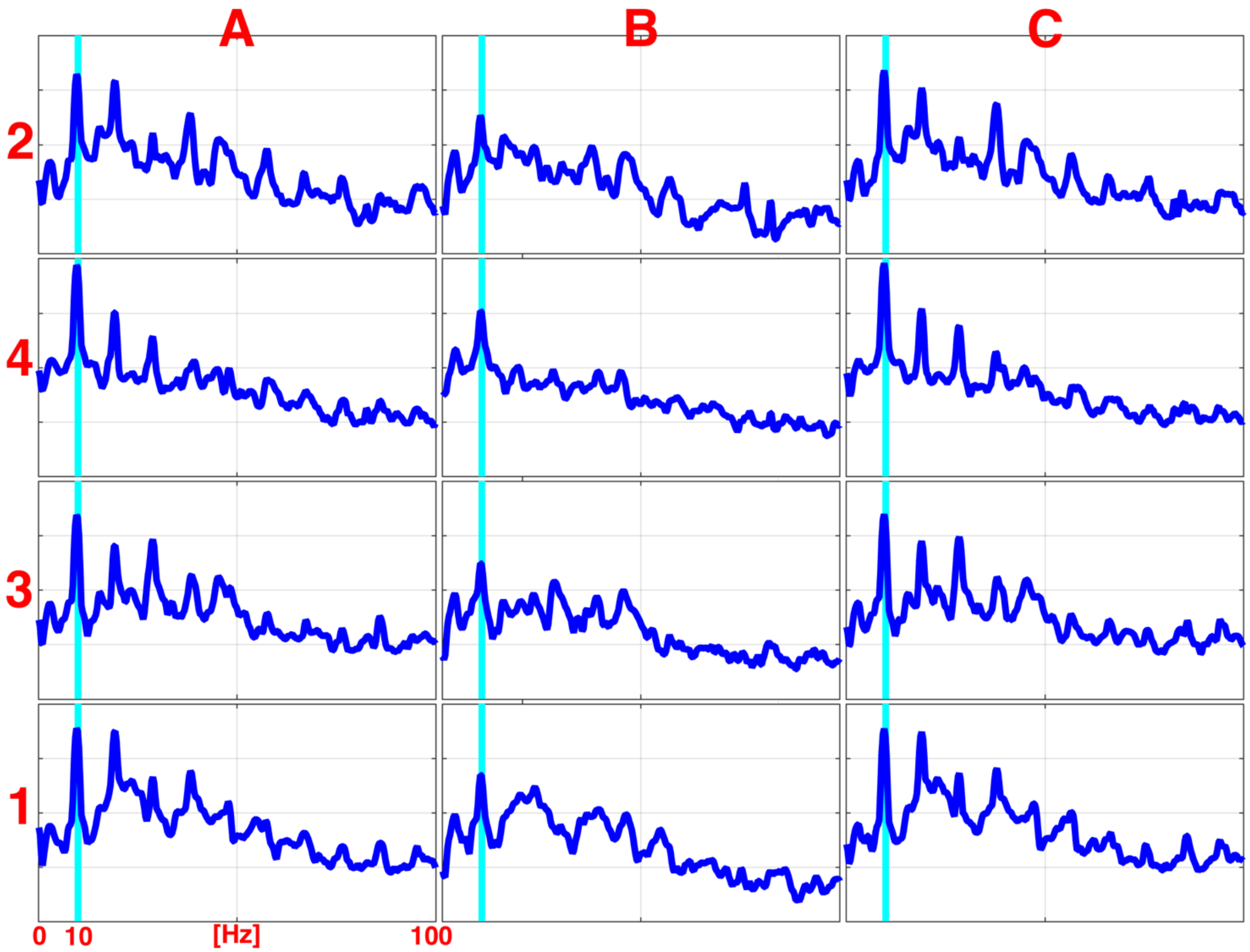

Figure 5. Power spectral density functions for the microphones at three conditions: rumble (A), normal (B), rumble (C). All spectra plotted on same $\log_{10}$ scale, from 0–100 Hz, with 10 Hz highlighted.

A question remains regarding the physical cause behind the 10 Hz natural frequency. This is significant because it does not seem to correspond to the Rijke or Sondhauss modes suggested by Eisinger and Sullivan (2002, 2008) as being the primary modes expected to be associated with thermoacoustic vibrations. In the cases they studied for similar boiler dimensions, the expected frequencies were typically in the range of 30–50 Hz. Thus there appears to be some other factor involved that is not accounted for by previous work in the literature. One possible explanation for the 10 Hz oscillation is that there could be some type of constructive interference between the forced-draft air fan and the furnace cavity. The fan has a maximum speed of around 1800 RPM, or 30 revolutions per second, and this speed varies depending on the firing rate. In the conditions measured here, the frequency range caused by the axial blade oscillations is thought to be around 250 Hz. To some extent, the fan oscillation frequency, coupled to the overall volume of the supply system and firebox, could explain the observations, but we are investigating to within the limits of available design and operating data to verify this assumption.

Another possibility is that the expected relationship between the pre- and post-burner resonance zones (i.e., between the air inlet and furnace cavities) that leads to the dominant frequency oscillations may not apply in this particular case. More specifically, it may be that the Rijke and Sondhauss modes are not dominant here, but instead there is another set of acoustic modes that are interacting with the heat release from the burner to drive a lower frequency resonance. This might occur if the characteristic wave length in the air inlet is actually different than suggested by Eisinger and Sullivan, resulting in the creation of a lower-frequency beating between the air inlet and furnace zones. As with the fan coupling, we are also investigating these other modes as possible root causes.

One of the more sophisticated analytical tools we are using to help understand the 10 Hz oscillations is time irreversibility. As is slightly visible in **Figure 4**, the large-amplitude acoustic waveforms were sawtooth shaped. For instance, for the fan-area transducer (microphone 4), the waveform has a slow rise to the cycle maximum and then a rapid fall to the cycle minimum; the opposite is seen for the air-duct transducer (microphone 3). Time asymmetry is important because Gaussian sources, or static transforms thereof, are temporally reversible, meaning that there is a statistical difference in time-sensitive metrics depending on whether the signal is examined in the forward or reverse sense. A metric capturing this difference, the T_3 function, is displayed in **Figure 6**. Each plot ordinate contains the same scale — the T_3 metric is naturally normalized — and the abscissa displays timescales from 0–0.2 s, much like with correlation functions. This time range captures roughly two cycles of the fundamental oscillation of the microphone signals. We note that opposite trending patterns occur in microphones 3 and 4. Additionally, the normal, non-rumble mode T_3 metrics are close to 0, suggesting a different cause than in the rumble conditions. Irreversibility is a hallmark of nonlinearity, and in many combustion systems, flame-oscillations are known to be temporally asymmetric. Thus the observation of pronounced irreversibility suggests that the flame is a key component of the vibrations, and thus the root cause is thermoacoustic vs. flow-based resonance.

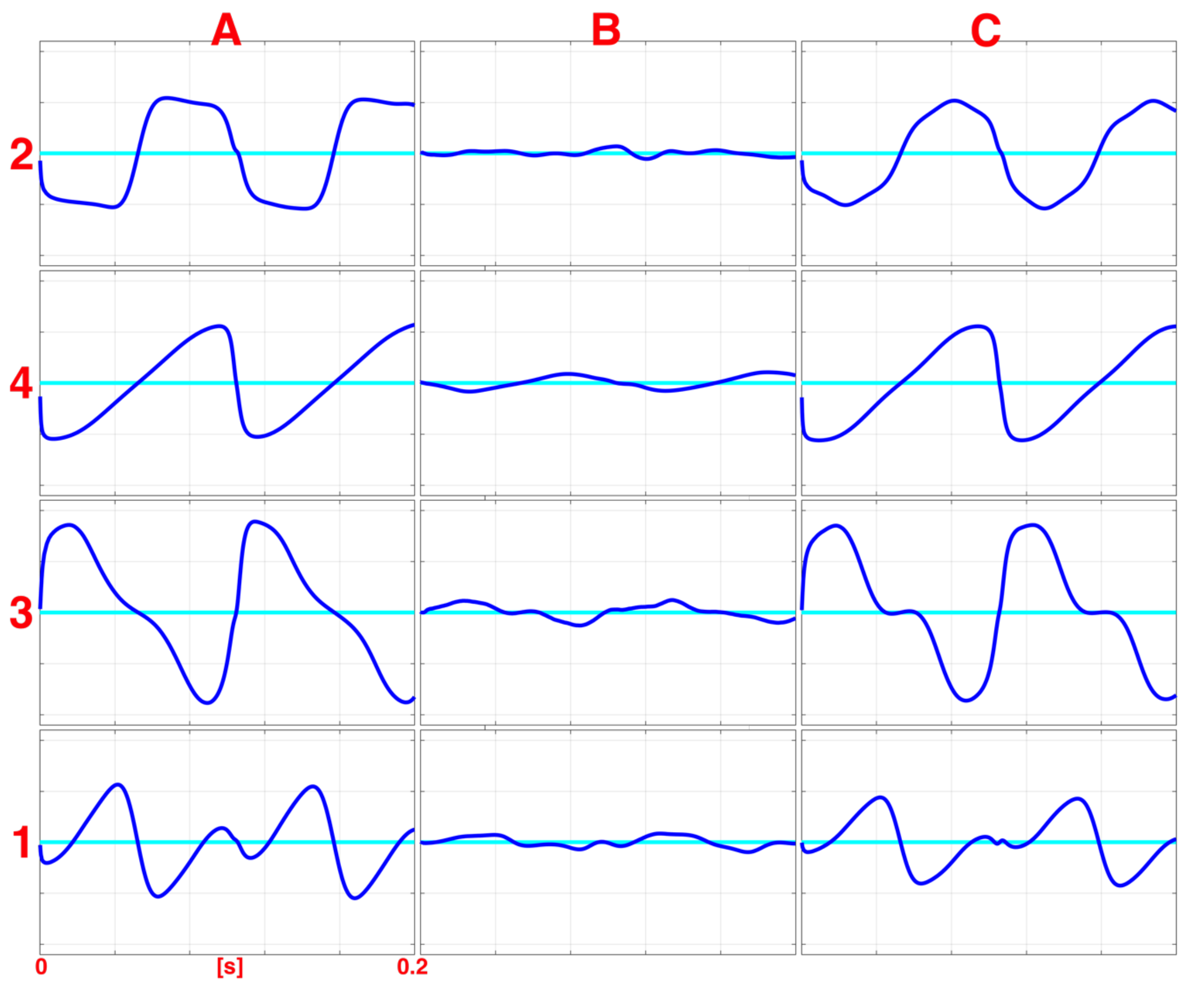

Figure 6. T_3 (irreversibility) functions for the microphones at three conditions: rumble (A), normal (B), rumble (C). All functions plotted on same scale, from 0–0.2 s lag, with T_3=0 highlighted.

Another useful characteristic of the B&W rumble case is revealed by cross-correlation among the acoustic signals. Cross-correlation functions quantify the linear correlation, or covariance, of two signals over a range of timescales. Cross-correlation is frequently employed in flow systems to measure timescales for spatial propagation of waves or flows. The cross-correlation functions for unique pairs of microphone signals is portrayed in **Figure 7**, where the pairs are given as microphone Y with respect to (wrt) X. Positive correlation of Y wrt X means that Y lags or follows X at the given time delay. A few general features of the cross-correlation functions may be noted. First, during rumble, there is stronger correlation compared with non-rumble between pairs of microphone signals, owing to the stronger coherence of each signal. Second, certain relationships are apparent in the locations of the correlation peaks. For instance, signal 1 lags signal 2 (top-most row), meaning the oscillation waveform happens in signal 2 before 1 (or at the firebox back wall before the generating bank), and signals 3 (inlet duct) and 4 (fan area) precede 2 (firebox back wall), suggesting a propagation of information from the fan or burner area downstream. However, because of the long-term cyclic persistence observed in the correlation functions, it is difficult to produce an exact mapping of information (pressure) propagation.

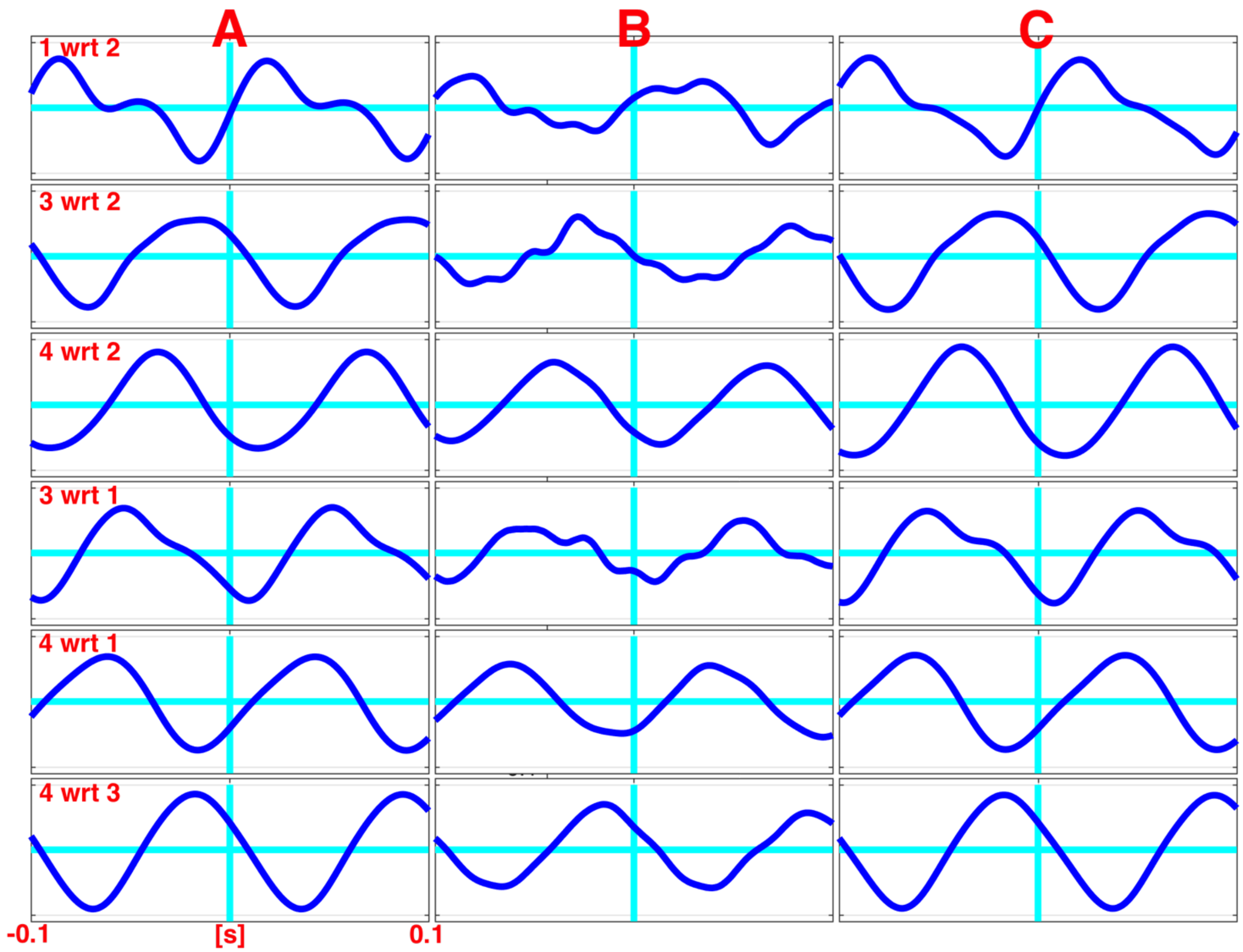

Figure 7. Cross-correlation functions for the microphones at three conditions: rumble (A), normal (B), rumble (C). All functions plotted on same scale, from –0.1–+0.1 s lag, with 0-axes highlighted. Positive correlation of Y wrt X means that Y lags or follows X at the given time delay.

In future work, we plan to employ more general, information-theoretic measures such as cross-mutual information or transfer entropy to define in a nonlinear sense of the information flows between sensors. The first pass of analysis suggests that information flows from the base of the flame area outward, which may be consistent with a thermoacoustic mechanism.

CONCLUSIONS

Vibration in industrial boilers is a complicated combination of fluid flow, combustion and system geometry. Much progress has been made to understand the root causes of vibration, but due to the complexity of the forcing functions of the vibration, it remains difficult to predict during the design phase, or address in an operating system.

In a limited study of boiler field measurements, we have observed pronounced resonant oscillations in microphone signals collected at several points around a single-burner gas-fired unit. The dominant frequency of these oscillations suggests that some process other than the fundamental thermoacoustic modes predicted by the Rijke and Sondhauss models is the primary root cause of the vibrations. Nevertheless, the presence of time asymmetry strongly implies that driving energy for the vibrations comes from the burner rather than flow instabilities (i.e., the vibrations are thermoacoustic rather than flow induced). To a limited extent, we are able to detect correlation patterns between measurement points that indicate how the main furnace cavity and burner are spatially coupled.

Future work will focus on verifying these results with more extensive data from other sensors as well as employing more sophisticated information-theoretic correlation measures to pinpoint the spatial sources of oscillations. Additionally, more rigorous analysis of the geometric and thermal driving sources can reconcile the observed vibrational frequency with analytical solutions. This will enable more selective design changes or operational controls to eliminate rumble in operating industrial furnaces and boilers.

ABBREVIATIONS

CFD	Computational Fluid Dynamics
FGR	Flue Gas Recirculation
IEPE	Integrated Electronics, PiezoElectric
NASA	National Aeronautics and Space Administration
PSD	Power Spectral Density
T_3	Temporal Irreversibility Function

SYMBOLS

L = Characteristic acoustic length of furnace cavity [m]
l_1 = Characteristic acoustic length of burner air inlet [m]
T_1 = Furnace (post-burner) temperature [K]
T_2 = Burner air inlet temperature [K]
$\xi = (L-l_1)/l_1$ = Characteristic geometric parameter for Rijke and Sondhauss resonance modes [-]

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