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Challenges of unanticipated power plant startup noise

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ABSTRACT

Most of today's power plant developers account for noise in the conceptual stages of the project and follow through with detailed engineering designs to assure the plant's final noise goals are achieved. Unfortunately, even some of the best engineering plans can't always predict the 'unanticipated noise' that is unique to each power plant. Unanticipated issues are first seen during the power plant's initial startup and are often the dominant noise sources within the community. Causes of unexpected noise problems include faulty materials used in silencing equipment, noise abatement that's been left out during construction, dynamic flow and burner instability issues, design errors, and vendors supplying equipment that is significantly noisier than they guaranteed. In any case, unanticipated noise issues are not generally associated with errors in the power plant's overall acoustical design but with the idiosyncrasies unique to the individual plant specific equipment or on-the-fly design or construction changes intended to 'improve function' but result in undesired noise. Ultimately, the unanticipated noise issues help form the new knowledge base necessary to reduce future problems.

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1. INTRODUCTION

Like people, power plants have only one chance to make a good first impression – and a noisy power plant is seldom welcomed to a community as a good new neighbor. Unfortunately noise issues usually arrive early in the facility's life – showing up during the initial startup and commissioning phases of the power plant. Furthermore, once the community is sensitized to a noise problem, any retrofitted noise abatement usually requires nearly complete elimination of the offending sound for the community to be fully satisfied. In any case, the silencing necessary to achieve acceptable results is often more substantial than had the problem been identified and mitigated to some degree before the community was exposed to it. Clearly minimizing as many of these initial startup issues is a major goal of the acoustical engineer.

While there are still occasional catastrophic noise problems due to a developer's or owner's lack of due diligence at the conceptual development stage of the project, most noise issues found in acoustically well-designed plants are related to individual pieces of equipment that didn't meet the noise control engineer's requirements, or a vendor's or manufacturer's design expectations.

This paper provides a series of brief case histories for some of the more typical noise problems encountered over the past decade. While many of the details have been omitted to disguise the plant and/or protect the guilty, the information provided is generally adequate to understand the problem and its resolution.

2. BRIEF CASE STUDIES

Power plant noise problems generally can be categorized into the following:

1.) Inadequate silencers – either nonexistent or that do not perform
2.) Unanticipated equipment or system dynamics that results in excessive noise or vibration
3.) New equipment where no sound data previously existed
4.) Mechanical 'improvements' to equipment or a system that neglected noise impacts

Although the case studies presented may focus on a specific item or piece of equipment, the cases are intended to be used as a basis of understanding more generalized problems.

A. Gas-turbine air-inlet silencers

High pitch sound was observed by neighbors in the community when a new three (3) unit peaking facility was first brought on-line utilizing nominal 150 MW gas-turbines. Plant personnel believed it was coming from the gas-turbine's air-inlet filter house.

The consultant saw a similar issue at another plant utilizing a different gas-turbine manufacturer's nominal 100 MW machines just a few weeks prior. Plant personnel were correct – it's fairly easy to diagnose a problem related to a poor performing gas-turbine air-inlet silencing system. It's predominantly tonal sound generated in the most audible 1000-2000 Hz frequency range. The sound source was quickly shown to be related to the gas-turbine's first stage compressor blade row by correlating Fourier Transform narrow band analysis with the calculated blade passing frequency harmonics. The blade passing tones are seen at integer multiples of the number of first stage compressor blades multiplied by the rotational frequency of the compressor. An example of the tones, as seen near the filter house and at 400 feet away, is shown in Figure 1. This facility, however, had committed to achieve octave band sound level criteria and the compressor tones were causing the upper frequency octave bands to exceed the requirements by as much as 20 dB.

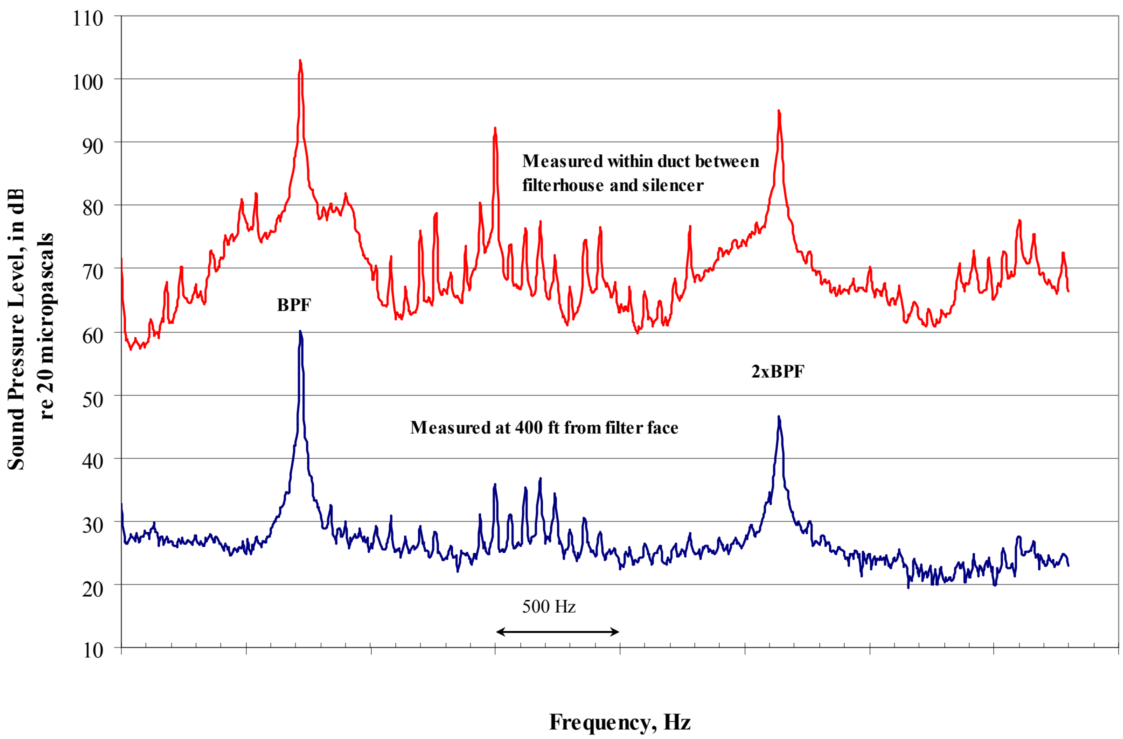

Figure 1: Tonal noise from a problematic gas-turbine air-inlet silencer.

Qualitatively, it was conjectured that the inlet silencer did not provide sufficient sound attenuation to reduce the compressor tones to acceptable sound levels. But the silencer was the same physical parallel baffle design that had been used in dozens of other successful applications. The gas-turbine manufacturer and the silencer vendor verified the silencer's length, baffle thickness, flow passage gaps, materials, etc. were made to the gas-turbine manufacturer's specifications. Fingers were pointed by both the gas-turbine manufacturer and silencer vendor. The gas-turbine manufacturer 'suggested' it had to be the silencer vendor's fault, while the silencer vendor argued they made the silencers to the manufacturer's specifications so it must be that the gas-turbines were making more noise than they had in previous applications.

At both plants, ***each*** gas-turbine manufacturer (different companies) stressed that they:

- 'Do not guarantee specific equipment sound levels'
- 'Only guarantee spatially averaged sound levels'
- 'Do not guarantee octave band sound levels'
- 'Met all near-field and far-field spatial average A-weighted guarantees'

Effectively, any problem was the plant owner's to fix even though the design sound power level data provided by the manufacturer(s) was as much as 20 dB(A) lower than the actual field tested inlet system sound levels indicated.

Subsequent testing with in-duct operational measurements and loudspeaker testing showed the silencer's dynamic insertion loss was about 25 dB(A) less than its theoretical design would indicate. Further investigation into the silencer's materials showed the fiberglass cloth used to wrap the silencer's mineral wool fill had a higher flow resistivity than is normally expected from similar batches of cloth material. While the cloth met the gas-turbine manufacturer's mechanical specifications for density, material, weave, etc., the gas-turbine manufacturers had failed to include a requirement of flow resistivity (the most important component of the acoustical performance of silencer materials). While the physical specs worked most of the time, they clearly weren't sufficient to assure the silencers worked as anticipated.

Several design modifications were considered, including removing and repacking the existing silencer and retrofitting an additional silencer into a small section of empty ducting just downstream of the inlet filter house. The empty ducting was about half the length of the silencer duct section. In both cases, the least expensive and quickest option was retrofitting the additional silencer in the existing empty duct section. A retrofit silencer, designed by the consultant for one of the projects and by a silencer vendor for the other, both provided about 20 dB(A) of the additional attenuation as expected. The sound reduction, however, was at the expense of slightly increased pressure drop and a corresponding reduction in the gas-turbine's efficiency. The small performance loss was considered acceptable by the plant owners for the simple-cycle applications but likely would have been a concern in a combined-cycle application.

B. Gas-turbine air-inlet ducting

In another simple-cycle gas-turbine power plant, compressor tones were radiated by the air-inlet ducting. Near-field measurements clearly indicated the tones were coming from a portion of ducting located between the gas-turbine enclosure and the air-inlet silencer. This was the first installation of a new model combustion turbine. The vendor responded quickly by adding mass and damping to the duct wall which resolved the problem.

C. Gas-turbine exhaust infrasound

A simple-cycle gas-turbine facility was designed by a large Engineering Procurement and Construction (EPC) firm. The noise requirements included meeting a C-weighted sound level requirement at the plant's property line. The EPC firm wrote the exhaust stack silencer specifications and asked vendors to guarantee the sound levels from their stacks. Their exhaust silencer specification included the 'estimated' un-silenced octave band exhaust sound power level spectra as supplied by the gas-turbine manufacturer. The vendors offered silencers based on the octave band sound power level data provided in the EPC firm's specification and 'guaranteed' the design as requested.

Many silencer vendors will not release details of their 'proprietary silencer design' resulting in the EPC firm purchasing a 'black box' guaranteed to meet their specification. Usually, the silencer is awarded to the low cost bidder. As a note, gas-turbine manufacturers generally supply only non-guaranteed sound power level data for use in exhaust silencer designs. The data supplied by gas-turbine manufacturers is in the 31.5 Hz to 8000 Hz octave bands and is provided 'for information only'.

When the facility was completed, the consultant was called to measure and assess the property line sound levels. The plant was found to miss the C-weighted property line requirements by as much as 8 dB(C) in some directions. Sound pressure at and below the 31.5 Hz octave band dominated the measured C-weighted sound.

While gas-turbine manufacturers' sound power level amplitudes are suspect in many cases, the frequency range they provide (31.5-8000 Hz) is typically adequate for A-weighted silencer designs. However, data provided in the 31.5-8000 Hz frequency range is rarely adequate for silencers required to meet C-weighted specifications. While typical A-weighted requirements are inherently controlled by mid-range frequency sound, a gas-turbine's C-weighted sound is controlled by very low, mostly inaudible frequencies (infrasound). Gas-turbines inherently produce high levels of low frequency sound in their combustion and exhaust process. Often, the exhaust's sound spectrum maximizes well below the 31.5 Hz octave band. Acoustical consultants who've designed exhaust silencers are aware of this – as are most experienced silencer vendors.

Subsequent testing within the exhaust ducting showed the un-silenced gas-turbine exhaust sound power level was more than 10 dB higher than that provided in the gas-turbine manufacturer's 'estimated' 31.5 Hz octave band sound power level data. Furthermore, the 16 Hz octave band and frequencies below contained substantial sound energy – also contributing to the C-weighted sound.

The power plant temporarily resolved the problem by purchasing a noise easement from adjacent property owners. Ultimately, a new silencer was designed and paid for by the power plant.

D. Gas-turbine combustion dynamics

A combined-cycle gas-turbine based power plant located in a rural area was used as an intermediate load facility. Its typical operation included starting every weekday morning at about 4 AM, brought to base load and continued operating at base load until about 10 PM. Neighbor(s) living near the facility were awakened by 'rumble' in the morning when the facility started. The low frequency 'rumble,' as presented in Figure 2, only occurred during startup and lasted only a few minutes before conditions stabilized and generally disappeared – until the next day's early morning startup. Because of the short duration and transient nature of the problem, it probably would have been acceptable to residents had it occurred at a more reasonable time – like the middle of the afternoon.

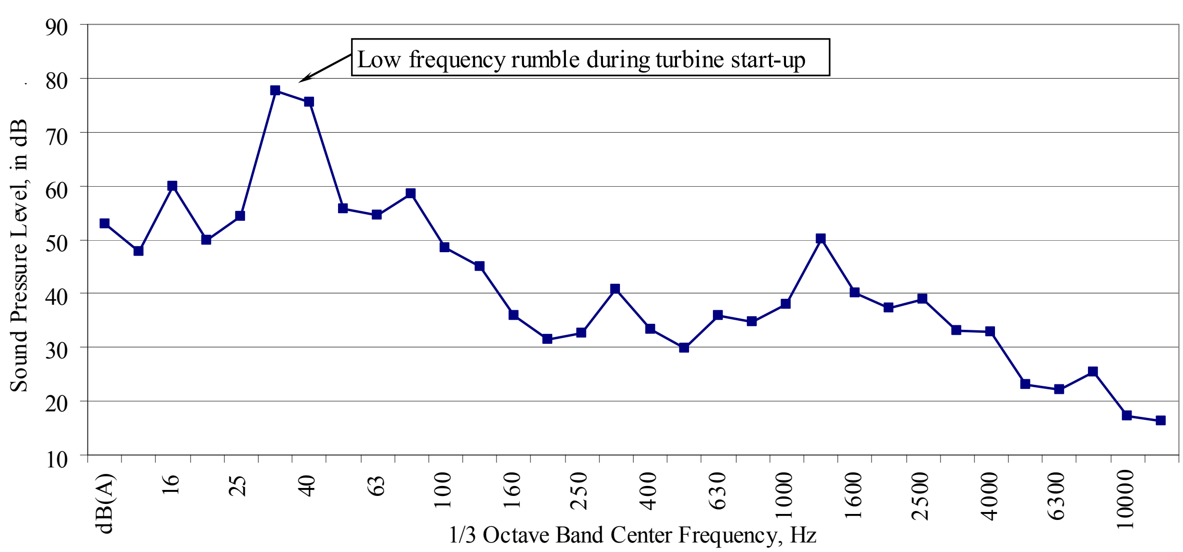

Figure 2: Low frequency rumble of combustion turbine at startup.

It was conjectured to be caused by combustion instability since as the conditions changed (air and fuel flow) and power was increased, the problem resolved itself. It was suggested that the high levels of low-frequency energy observed as a transient during startup could be treated by re-tuning the gas-turbine's startup process and corresponding conditions such as inlet guide vane position, fuel flow, etc. The gas-turbine manufacturer was not responsive at first to the request but eventually (and reluctantly) modified the operating startup schedule.

E. The transonic turbine (tonal harmonics)

Issues associated with new equipment occur periodically and it is nearly impossible to anticipate where potential problems will occur; the only given with 'new' equipment (either new to the marketplace or new to the acoustical engineer designing the facility) is that more than likely something unexpected will happen and require correction. Even if manufacturers can properly estimate the sound power level generated by the equipment, issues such as unique sound qualities generally are not known until the equipment is placed in service.

In the case of a simple-cycle gas-turbine facility utilizing a high efficiency turbine with transonic tip speeds of the rotating components, the octave band sound power levels, as provided by the gas-turbine manufacturer, were quite accurate. The corresponding measured community A-weighted sound levels were also very close to the predicted community levels. However, the quality of sound produced was unique in that it produced low-frequency tonal buzz saw like harmonics as seen in Figure 3. The sound was unique enough that the facility could be identified by residents living far away from plant even though it was mostly below the community's residual ambient sound. The facility sounded like distant propeller aircraft.

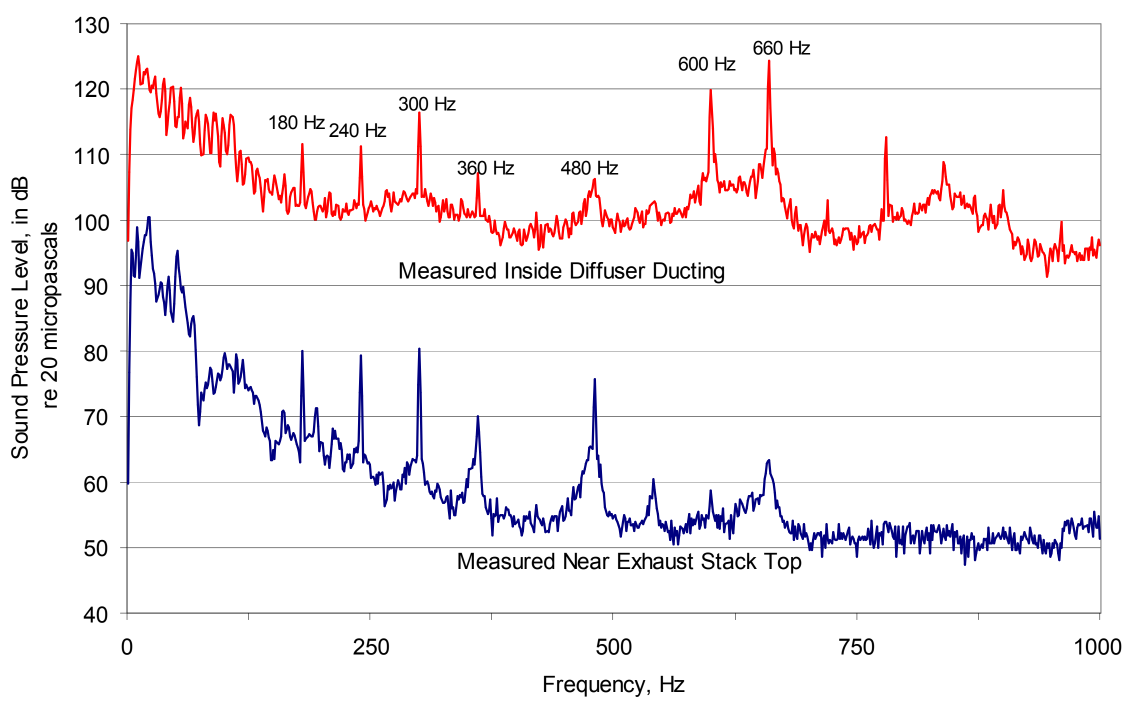

Figure 3: Tonal qualities of advanced turbine designs.

The difficultly in assessing this noise problem was that the tonal frequencies observed at 60 Hz harmonics are also harmonics seen in most every other piece of power plant equipment. Until measurements were made within the exhaust ducting which conclusively pinpointed the problem, several plant components – including the generators – were thought to be the cause.

The suggested solution involved adding exhaust silencing to reduce the severity of the tonal sound and reduce the tonal footprint within the community.

F. Gas pressure reduction piping and metering station

The gas pressure reduction and metering station piping were found to be the major noise source in an otherwise very quiet plant. The piping and valves were more than two-hundred feet from the major plant equipment but were placed very close to a property line – with an A-weighted property line noise ordinance. Gas metering requires long piping sections that can be very efficient in radiating noise. The radiated sound is mostly high frequency noise (1000 Hz and above). Specifying heavy wall piping and low noise valves alone did not result in the desired results. Luckily, when gas metering noise problems are found, the sound is at frequencies where acoustical pipe lagging and sound barrier walls are quite effective. This problem was solved with a three sided sound barrier wall located between the gas pressure reduction/metering station and the property line.

G. Air-cooled condenser steam bypass operation

In combined-cycle power plants with air-cooled condensers (ACC), steam bypass operation can lead to significant noise problems; not only from an occupational health and safety perspective, but also as a community noise issue. Often community noise requirements do not differentiate between or allow for additional noise during plant startup operating conditions – leading to a significant challenge for the steam bypass design.

The steam-turbine and the air-cooled condenser are connected through a main steam duct and several risers, which leads the steam to the ACC's heat exchangers. The air-cooled condenser and the steam-turbine are often separated by 100 feet or more, and the main steam duct is typically in the range from 15 feet to 22 feet in diameter – which leads to a significant surface area with very efficient noise radiation properties.

The steam bypass system is used during initial plant capability demonstration testing of the bypass system, during some plant startup scenarios, and emergency events where the steam-turbine trips and the high energy steam is bypassed to the condenser. The main noise components of the bypass system are the steam bypass valve and a diffuser. The diffuser typically protrudes into the main steam duct. Both the bypass valve and the diffuser assist in reducing the potential energy of the high pressure and temperature steam. The diffuser is often a multistage pressure reducing design, which can be of a proprietary design or a conventional drilled-hole design. The main ACC steam duct is maintained at a very low pressure while the upstream steam pressure is very high. The large pressure drop across the bypass valve and diffuser results in a substantial amount fluid dynamic noise. The internally generated acoustic energy is then transmitted through the ACC duct work. Depending on steam conditions and diffuser design, the sound pressure level measured one (1) meter from the main steam duct was as high as 110 dB(A).

On several occasions, the 'low noise' technology used in diffuser designs, often offered by valve vendors, did not meet the design requirements, and resulted in significant rework of the ACC ducting.

To define the noise problem, sound pressure level measurements and surface vibration measurements were conducted along the length of the ACC duct and risers in order to accurately determine the sound radiation and sound decay along the length of the ducting.

The detailed noise and vibration surveys were used to evaluate the required noise reduction and the extent of the acoustic mitigation. In this case, a noise reduction of 15-20 dB was required. To reduce the duct sound radiation, an elaborate acoustic lagging system was designed which minimized the structural coupling between the ACC duct wall and the external metal cover sheet.

H. High-pressure bypass to air-cooled condenser (case 2)

Bypass of high-pressure (HP) steam directly to an air-cooled condenser during restart of combined-cycle power plants have been problematic. The pressure drop of HP steam to the partial vacuum of the ACC dissipates a great deal of energy during a plant restart. Early in the design stage, considerable effort went into avoiding this noise problem; a low-noise HP steam bypass valve was procured to meet a limit of 85 dB(A) at 1.5 m, a silencer was placed immediately downstream of the valve, and the main steam duct, from the steam-turbine outlet to the ACC, was covered with a shroud. In theory, these should have prevented a noise problem. The HP steam flowed trough the valve, a steam silencer, then into the very large diameter steam duct from the steam-turbine exhaust to the ACC.

During a cold plant restart, operators in the control room, which was located adjacent to the HP bypass valve, described the noise as sounding like a Saturn V rocket taking off. After the startup procedure was changed to reduce the noise, the indoor HP bypass valve was measured to be 108 dB(A), which was 28 dB higher than the guaranteed limit. During HP bypass, a level of 58 dB(A) was measured at the nearest residence. This level caused concern about possible shutdown in response to nuisance complaints. After fill from the silencer was found in the ACC tube banks, all the silencer's fill was removed to prevent plugging the tube banks. Since the noise controls were carefully designed, specified, and installed, the noise problem was unexpected.

The process of retrofit noise control for the HP bypass valve is a near classic example of how to not do noise control engineering. First, it was discovered that the silencer design was based on an incorrect upstream pressure. For months, the valve vendor measured levels and eventually installed a seat diffuser in the valve, which reduced the A-weighted level near the valve by about 3 dB(A). The vendor had been told by their noise control engineer that the seat diffuser would not appreciably reduce noise, but it was far cheaper than a meaningful solution and at least demonstrated some action. Then, a conical diffuser was installed in the bypass duct downstream of the gutted silencer. The diffuser increased noise due to high flow velocity design issues. The diffuser met its specification. Unfortunately, the diffuser vendor had been given a performance specification, but not the details of the application. The diffuser was removed. A second diffuser was installed at the end of the bypass line so that its entire length was in the main steam duct leading to the ACC. For well over a year, the HP bypass valve vendor talked, but did not respond. Noise was then controlled only after pressure from higher management was applied to the valve vendor. The plant met its 53 dB(A) limit at the nearest residence only after three concentric diffusers were installed at the valve outlet. Controlling noise was made more difficult, and took longer than necessary, because there was a systematic lack of communication.

I. Induced draft fan blade tone

A coal-fired power plant received strong complaints about a tone (better described as a ‘growl’). The tone was intermittently audible in the community, but could not be heard near the plant. While tones typically exist in Induced Draft (ID) fans, predicting whether a significant tone will occur in the community is often not feasible. When it does, it’s an unexpected noise problem.

The tone was approximately 170 Hz, which corresponded to the blade passage frequency of both the Forced Draft (FD) fan, and the ID fan. Near-field measurements determined that the tone was generated by the ID fan, but the transmission paths were uncertain. A check was made to identify any other tones at 170 Hz but none were found. The dominate transmission path was identified by a process of elimination. Since the tone was not audible at locations within 500 feet of the fan casing, it was concluded that the fan casing and short exhaust duct were not a significant transmission path. The stack’s external walls were made of concrete, so it was also eliminated. It was hypothesized that the cause of the intermittent tone was related to variable wind speed and direction. Thus, the most likely candidate was the sound emitted at the stack top.

The exhaust ducts between the fan casing and the stack were very wide and tall, but also very short in length. Due to capital cost and their effect on flow, silencers in the exhaust ducts were determined to be an undesirable option. A series of scale model tests were run on possible modifications to the cut-off bars including v-shaped, and variable gap and notch designs. The best cut-off bar design was installed. While appreciably reducing the tonal noise, it was still not quiet enough. Eventually, a silencer was installed in each of the flues within the exhaust stack, solving the noise problem.

J. Weeping steam drain vent

Noisy steam drain vents were identified during the design stages of a combined-cycle power plant. Appropriate vent silencers were specified and installed. Unfortunately, one vent was missed during the design. The vent was for a tank that collected drains from various systems which had been erroneously deemed to be insignificant.

Designing a small vent silencer that reduced noise to an acceptable level was easy – installing it on a cluttered roof – difficult. To ease installation, the silencer was required to be fabricated in two pieces of roughly the same weight and size that could be reassembled on the roof. Construction chose to use rigging inside the plant to lift each piece, rather than lift over the edge of the plant. The silencer worked as anticipated and resolved the noise problem.

K. Blowdown tank atmospheric steam vent

As the acoustical consultant drove down the road leading to a combined-cycle gas-turbine plant, a problem was immediately identified – even though he was still about a half mile away. High frequency broadband sound filled the car. Listening to the car’s radio became nearly impossible upon entering the plant property. The steam venting near the exhaust end of the generation building was the first clue; it came from the intermittent blowdown tank. Asking the plant personnel about the noise they said that they ‘hardly noticed it’. Sound levels measured about 50 feet from the blowdown tank atmospheric vent were well in excess of 110 dB(A). In the community, the blowdown tank noise was clearly audible and strongly impacted the community A-weighted sound level as seen in Figure 4. This had occurred for months prior to the consultant’s scheduled noise compliance testing.

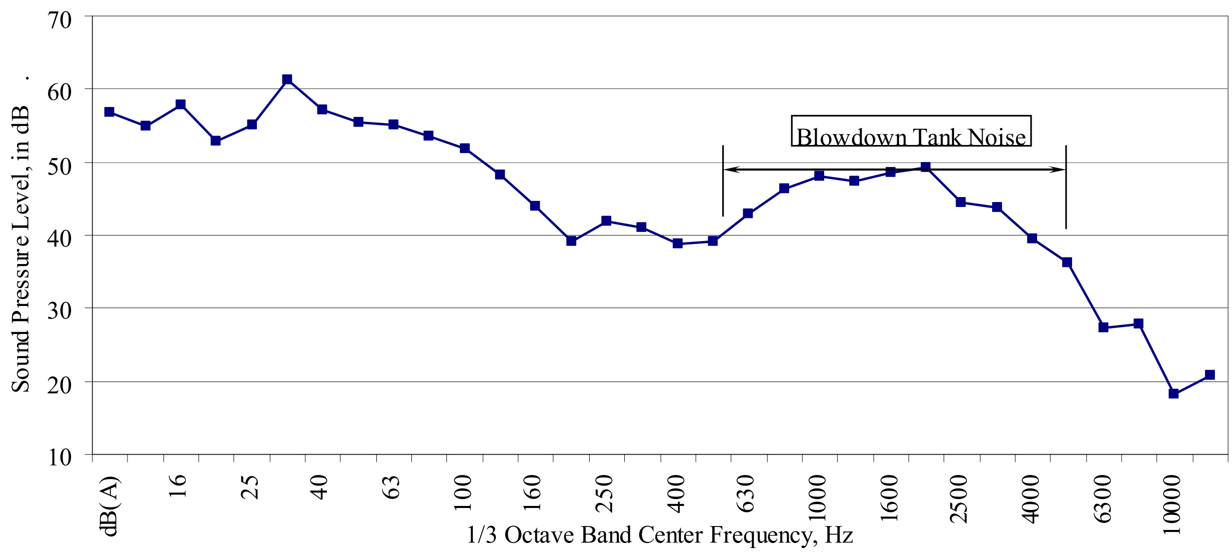

Figure 4: Blowdown tank noise measured within the community without a silencer.

While a silencer had been specified and purchased for the blowdown tank, it had not been installed because of a dimensional interference problem with some of the other equipment and generation building. Plant personnel didn't understand the silencer's importance to the overall noise plant control, so they placed the silencer in a storage area.

When the EPC contractor and plant owner/management were made aware of the magnitude of the problem, they worked to eliminate the interference issues, retrieved the silencer from storage, and installed it. Problem solved.

L. Non-existent 'low-noise' valve in duct burner skid

In many situations, equipment is specified to include valves with low-noise trim on various high-pressure gas or steam applications. In this case, a low-noise valve had been specified for use on a duct burner skid in a combined-cycle cogeneration facility. The valve should have achieved a guaranteed sound level of 85 dB(A) at 1 meter from its surface but instead produced about 95 dB(A). Furthermore, the sound generated by the valve had a piercing high frequency component that could be clearly heard at some plant property line locations where a clear line of sight to the duct burner skid existed.

When the problem was identified, it was assumed the valve had not met the sound guarantees. However, the plant mechanical engineers later acknowledged that they had removed the low-noise valve and replaced it with a 'standard' valve. Apparently, the valve's low noise trim didn't allow or provide the same level of control as the standard valve they had used in similar plants – even though on paper, and as specified, both should have worked equally as well.

A sound barrier wall was designed to partially enclose the duct burner skid and minimize its property line sound contribution. Since neighbors had not complained, the plant owner decided to live with the higher sound levels and never installed the recommended noise abatement.

3. CONCLUSIONS

Through a combination of art and science, experienced acoustical engineers have reached a good level of accuracy when predicting power plant noise control designs. However, power plants have many complexities and idiosyncrasies that make each facility unique in some way. Even when similar major equipment are used in a plant design, small differences and resulting noise problems can occur from some of the least likely and seemingly insignificant equipment.

While the case studies presented may focus on specifics, they also apply to more generalized cases. For instance, the low noise valve trim used in the duct burner example is not unique to duct burner applications. Anywhere where low noise valves are specified may result in potential control issues. Also, if noise control solutions are found to impede equipment operation in any way, on-site engineers (non-acoustical) may 'fix' the problems by eliminating the noise abatement – often without informing the noise control engineer.

Experience is the best way to avoid many noise problems but often even the most experienced acoustical engineers and consultants run into new or unforeseen situations that result in surprising new challenges.