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ACOUSTIC AND VIBRATION ANALYSIS OF FLUID INDUCED BLOWER AND PIPING UNWANTED MOTION

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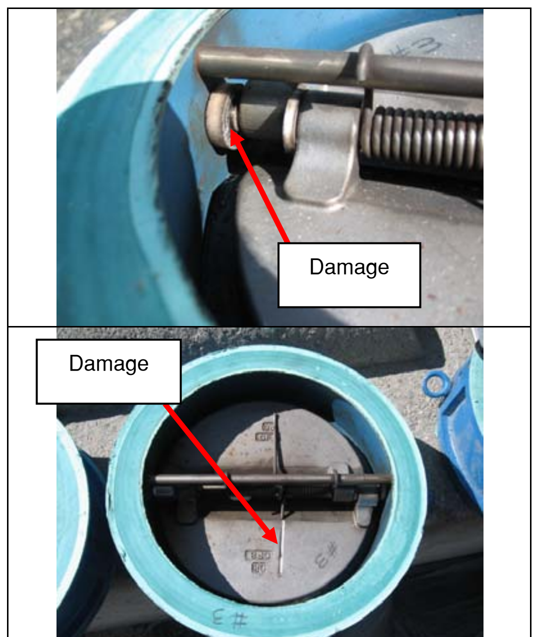
Abstract

This paper presents investigational findings and a discussion of recommendations relating to sound and vibration measurements performed in connection with a fluid induced vibration issue on two air blowers and attached exhaust piping at an industrial facility. These vibration and sound measurements were prompted by recent check valve failures for the air blower units, and unacceptable sound levels emanating from these units and affecting nearby residents. This data was acquired during steady state operating conditions of the blowers under normal operating conditions. An FFT data acquisition system, a piezoelectric microphone and three piezoelectric triaxial accelerometers were used to collect vibration measurements at each of the 70 locations on the blowers, motors, blower bases, and exhaust piping, while sound measurements were simultaneously acquired with the microphone. Piping and blower vibration readings were used to construct an operating deflection shape analysis of the blowers, foundations and attached piping system. The resulting vibration and sound analysis revealed that acoustic excitation of the piping system appeared to be the likely source of the high vibration, high sound pressure levels; piping cracks and check valve failures. Corrective actions were implemented that reduced the sound pressure levels, vibration levels, and reduced/eliminate the piping damage and valve failures.

INTRODUCTION

Metrix was retained by plant management to provide vibration and acoustic diagnostics expertise to analyze elevated noise and vibration levels of four air blowers. This project encompasses sound and vibration measurements of aeration blowers and attached exhaust piping for an MBBR wastewater treatment plant within a large industrial plant. This investigation was prompted by recent check valve component failures for these blower units, and unacceptable sound levels emanating from these blower units which potentially would affect nearby residents (photos 1 and 2). Plant maintenance workers were also hesitant to work around these units due to high noise and vibration causing nausea and other medical problems. Onsite vibration and sound measurements took place in March 2010. The vibration and sound data were acquired during steady state operating conditions of the blowers. During these measurements only two blowers were operating, blower “B” and blower “C”.



Photos 1 and 2: Check Valve Damage

Recent check valve failures and repair of cracked exhaust piping were causing forced outages of the blower units and severely disrupting the water treatment process. Economic impact of the potential cracked valves and piping while in service were causing waste treatment plant downtime and were estimated to be exceeding $100,000 for 2010. Potential complaints about objectionable noise levels from nearby residents were also a concern. By correctly identifying root

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cause(s) of the high vibration and noise levels, this cost was avoided and a positive relationship with nearby residents was maintained.

INSTRUMENTATION

Throughout the body of this paper, the following conventions will be used; regarding sensor naming and orientation. Blowers are "viewed" as being with the "X" direction, or axial direction, along the shafts of the motors and blowers. The positive "X" direction is along the blower driven shaft from blower C towards blower B. Likewise, the "Y" direction will be perpendicular to the "X" direction in the horizontal plane, see figures 1 and 2. The "Z" direction is vertical, perpendicular to both the "X" and "Y" directions; forming a "right hand" coordinates system for these vibration measurements. The triaxial accelerometer sensor directions follow this same coordinate system. For instance, a vibration measurement at location 1 in the vertical direction will be referred to as a vibration amplitude (velocity and displacement) "1Z" in the following vibration summary tables later in this paper.

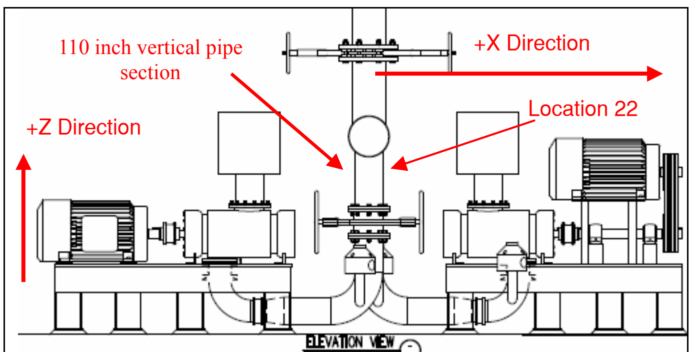

Figure 1: Vibration Measurement Coordinate System

A Data Physics Abacus data acquisition system and Data Physics Mobilyzer analysis software were used to collect vibration measurements at each of 70 key locations on the blowers, motors, blower bases, and exhaust piping. Three PCB triaxial accelerometers were used to capture vibration conditions at each selected location and a TMS 130P10 ICP microphone was used to capture sound readings adjacent to Blower C.

SOUND MEASUREMENTS AND OBSERVATIONS

Sound pressure levels were measured during the vibration measurements using a single microphone at 4 locations adjacent to blower C. Measured sound pressure levels typically reached 107-110 dBA with the highest sound component being at 3600 cpm (60Hz) of about 104 dBA [1,2], see figure 3. The predominate spectral components were 3600 cpm (60 Hz) and harmonics of 3600 cpm, namely 7200cpm, 10,800 cpm, 14,400 cpm… There was also a smaller spectral peak at 1800 cpm, which correlates with running speed of the blower. Since these blowers have two lobes on the higher speed drive shaft, pressure pulsation in the exhaust of the blowers should be occurring at 3600 cpm (3600 rpm = 2x 1800 rpm drive speed) which showed up in the sound spectra.

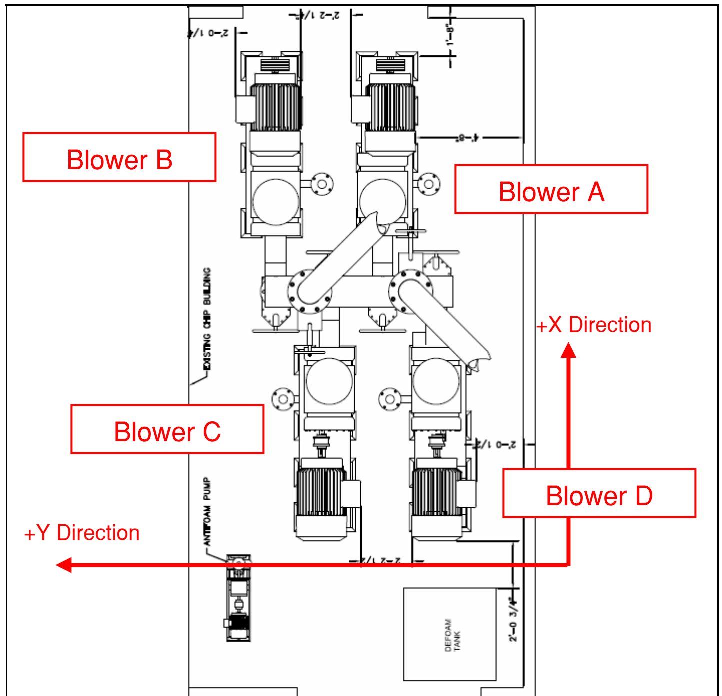

Figure 2: Vibration Measurement Coordinate System

At standard temperature and pressure, acoustic wavelength of the 3600 cpm component would be about 225 inches, thus too long to form a standing wave within the current piping within the building, see photos 3, 4 and 5. The longest piping length was approximately 180 inches for the vertical section of blower D piping, which ran from the floor to the ceiling where the piping exited the building. After the leaving the building, the piping took a horizontal turn to the wastewater tanks. However, the vertical piping runs of each blower from the floor to the crossover manifold pipe were about 110 inches, see figure 1. This run of pipe could feasibly accommodate a 1/2 wavelength standing wave at 3600 cpm and a full wavelength standing wave at 7200 cpm form within the pipe [3,4], see table A. When these blowers were designed, piping lengths that match critical acoustic wavelengths as shown in table A should have been avoided whenever possible.

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Photo 3: Blowers A and B and piping

Photos 4 and 5: Vertical piping from each blower, each being approximately 100 inches long

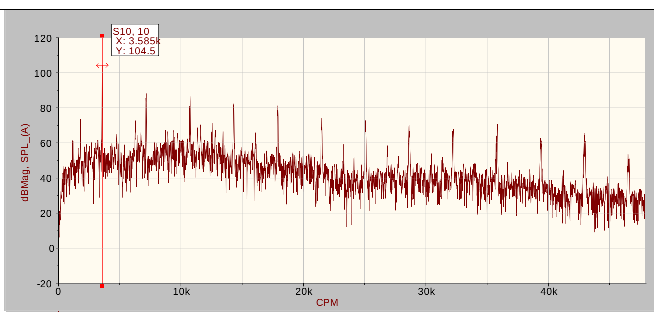

Figure 3: Sound Pressure Spectrum, Adjacent to Blower C, 111 dBA Overall

Since the blowers input shaft are turning at 1800 rpm and each shaft has 2 lobes, pressure pulsations from the exhaust of these blowers occurred at 3600 cpm or 60 Hz. This naturally explains the high sound level component at 3600 cpm. Unfortunately, this frequency also roughly matches the length of a ½ wavelength potential standing wave in the vertical pipe section of each blower exhaust. To suppress these pressure pulsations, a discharge (exhaust) silencer was considered to be a good initial solution; however examination of the blower configurations revealed that this would likely require significant modifications to the existing arrangement of blowers and increase installation cost.

Frequency cpm	Velocity in/sec	¼ λ in/cycle	½ λ in/cycle	λ in/cycle	1.25λ in/cycle	1.5λ in/cycle
1800	13500	112.50	225.00	450.00	562.50	675.00
3600	13500	56.25	112.50	225.00	281.25	337.50
5400	13500	37.50	75.00	150.00	187.50	225.00
7200	13500	28.13	56.25	112.50	140.63	168.75
9000	13500	22.50	45.00	90.00	112.50	135.00
10800	13500	18.75	37.50	75.00	93.75	112.50
12600	13500	16.07	32.14	64.29	80.36	96.43
14400	13500	14.06	28.13	56.25	70.31	84.38
16200	13500	12.50	25.00	50.00	62.50	75.00
18000	13500	11.25	22.50	45.00	56.25	67.50

Table A

VIBRATION MEASUREMENTS & OBSERVATIONS

To evaluate mechanical condition of each blower, vibration measurements were taken at each blower shaft bearing housing. Vibration levels at each blower bearing were measured in vertical, lateral (radial) and axial directions see tables 1 and 2. Two PCB triaxial accelerometers and Data Physics Abacus vibration data acquisition system were used to measure vibration on each blower. Representative vibration spectra from these measurements are shown in figures 4A through 4C. Almost all of the vibratory energy occurs at running speed (1800 cpm) of the blowers and harmonics (3600 cpm, 5400 cpm, 7200 cpm…) of this frequency. Vibration higher than 0.6 inches per second (ips) peak are considered by the blower OEM to be elevated vibration levels, and these readings are highlighted in yellow. Elevated vibration levels in multi-lobe blowers can be due to a variety of sources (root causes):

- Misalignment (coupling misalignment)
- Impellers/lobes rubbing
- Worn bearings and gears
- Unbalance
- Blower looseness
- Piping resonances (standing wave pressure pulsations)
- Foreign material buildup
- Casing strain

Vibration levels were consistently higher in the vertical direction compared to the axial and lateral directions, particularly for blower B. This appeared to be likely caused by downward exhaust pressure pulsations (forces) in the exhaust, which exits downward from beneath each blower [5], vibrating both the blower and piping vertically. This motion was plainly visible in the operating deflection shape of each blower, seen in later section of this paper. These measurements also showed that vibration frequency components at blower running speed (1800 rpm) and below were poorly correlated (low coherence)

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with piping vibration in this frequency range, see figures 5 and 6. These measurements also indicated that blower vibration at twice blower running speed (3600 rpm) and above did correlate well with piping vibration and acoustic energy components in this frequency range. In short, blower vibration at running speed, like blower unbalance…, was not responsible for the observed piping vibration.

Blower C Locations		Measurement Direction: Vertical Z ips Peak	Lateral Y (Radial) ips Peak	Axial X ips Peak
Coupling End, Drive Shaft Bearing	OA	**0.693**	**0.368**	**0.342**
Coupling End, Drive Shaft Bearing	1X	**0.018**	**0.091**	**0.045**
Blind End, Drive Shaft Bearing	OA	**0.624**	**0.604**	**0.463**
Blind End, Drive Shaft Bearing	1X	**0.033**	**0.193**	**0.043**
Coupling End, Idle Shaft Bearing	OA	**0.706**	**0.379**	**0.465**
Coupling End, Idle Shaft Bearing	1X	**0.031**	**0.091**	**0.026**
Blind End, Idle Shaft Bearing	OA	**0.464**	**0.565**	**0.546**
Blind End, Idle Shaft Bearing	1X	**0.044**	**0.190**	**0.032**

Notes:
OA = Overall Direct (unfiltered) vibration amplitude, inches per second (ips) peak
1X = Synchronous (once-per-revolution) filtered vibration amplitude, inches per second peak

Table 1

Blower B Locations		Measurement Direction: Vertical Z ips Peak	Lateral Y (Radial) ips Peak	Axial X ips Peak
Coupling End, Drive Shaft Bearing	OA	**0.845**	**0.509**	**0.558**
Coupling End, Drive Shaft Bearing	1X	**0.214**	**0.140**	**0.126**
Blind End, Drive Shaft Bearing	OA	**0.940**	**1.089**	**0.554**
Blind End, Drive Shaft Bearing	1X	**0.271**	**0.383**	**0.136**
Coupling End, Idle Shaft Bearing	OA	**0.814**	**0.463**	**0.709**
Coupling End, Idle Shaft Bearing	1X	**0.241**	**0.137**	**0.248**
Blind End, Idle Shaft Bearing	OA	**0.825**	**1.108**	**0.714**
Blind End, Idle Shaft Bearing	1X	**0.219**	**0.408**	**0.249**

Notes:
OA = Overall Direct (unfiltered) vibration amplitude, inches per second (ips) peak
1X = Synchronous (once-per-revolution) filtered vibration amplitude, inches per second peak

Table 2

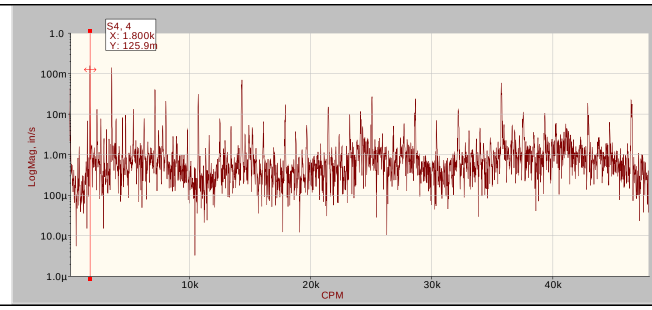

Figure 4A: Auto Power Spectrum, Blower B, Driven Shaft, Coupling End Bearing, Vibration Velocity in Axial Direction

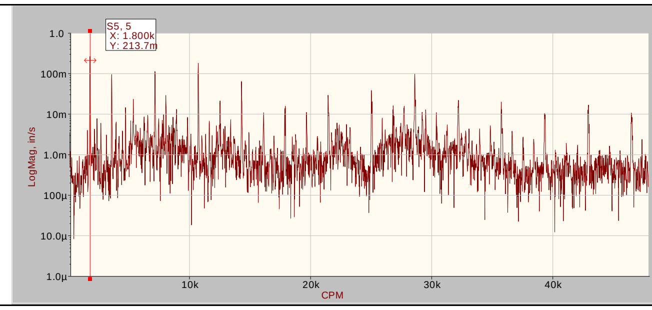

Figure 4B: Auto Power Spectrum, Blower B, Driven Shaft, Coupling End Bearing, Vibration Velocity in Vertical Direction

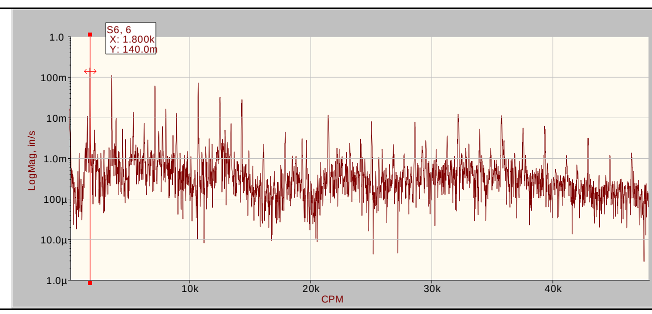

Figure 4C: Auto Power Spectrum, Blower B, Driven Shaft, Coupling End Bearing, Vibration Velocity in Lateral or Radial Direction

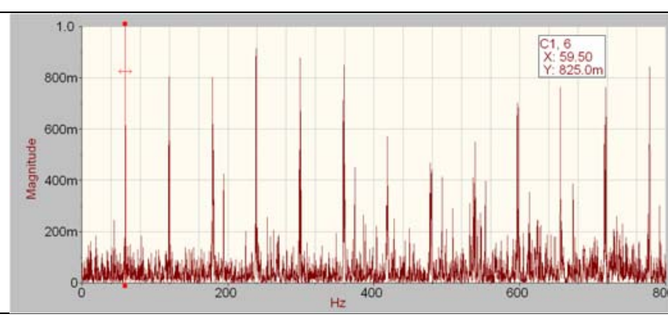

Figure 5: Coherence Spectra, Coherence Between Blower C at Reference Location (-Y) Direction and Piping Location 22 (-X) Direction

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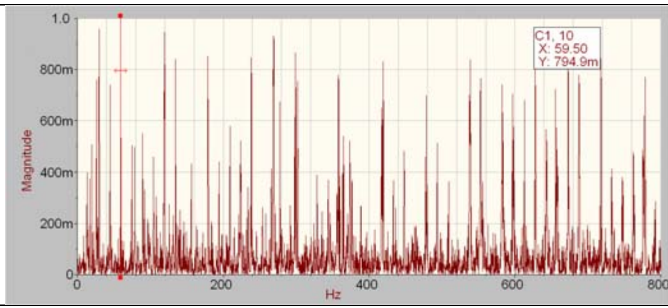


Figure 6: Coherence Spectra, Coherence Between Blower C at Reference Location (-Y) Direction and Microphone

OPERATING DEFLECTION SHAPE ANALYSIS

An operating deflection shape (ODS) is defined as the deflection pattern or shape of a mechanical structure (machine or machine components) at a particular frequency (typically at operating speed). This shape is determined by measuring dynamic deflections of a machine in response to operating forces at two or more locations on machine, while it is in operation. This methodology is used to show the relative motion of machine components due to existing forces. As well, the ODS gives an intuitive representation of the synchronous (1X) motion (and other frequencies) of the blower units and connected piping. When the frequency of the existing forces coincides with natural frequencies of the machine, it will resonate. When this occurs, the mode shape and ODS at that particular frequency will look similar.

To perform ODS measurements, a reference triaxial accelerometer was mounted to the "B" blower housing at a location and in an orientation where the response of the blower to existing forces can be readily measured and are significant. Typically, the location exhibiting the highest vibration level is selected as the reference location. In this instance, a blower housing outboard (blind side) bearing housing was selected (see photo 3). Three triaxial accelerometers and Data Physics Abacus vibration data acquisition system were used to measure vibration on each blower and piping. This first triaxial accelerometer was used as an amplitude and phase reference for other measured locations. This sensor remained fixed at this location throughout the ODS measurements. Two other triaxial (3 orthogonal axes) accelerometers were used for measuring the response of the two blowers "B" and "C" at other key sensor locations in a manner to sufficiently define the motion of the machines. Since radial vibration motion of the blower appeared to be a direct source of rotational vibration energy, the reference accelerometer was mounted to the higher speed shaft side, blend end of the blower housing and orientated with its axes in the radial (lateral, transverse) direction, axial direction, and vertical direction of the blower. The ODS reference vibration amplitude was measured using the radially orientated triaxial accelerometer channel. As mentioned, two triaxial accelerometers were used to acquire 3 dimensional vibration deflections at each of the 70 locations (XYZ directions; see sketch A and B below).

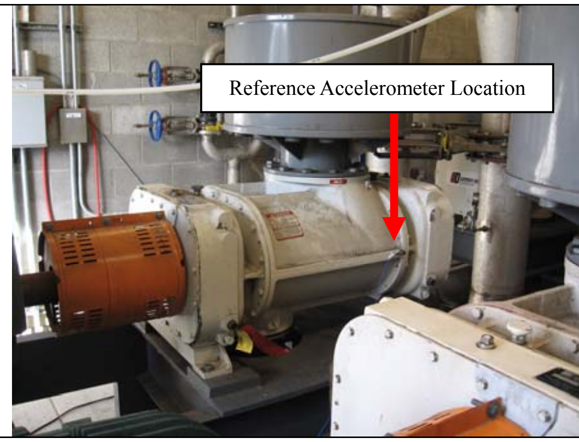
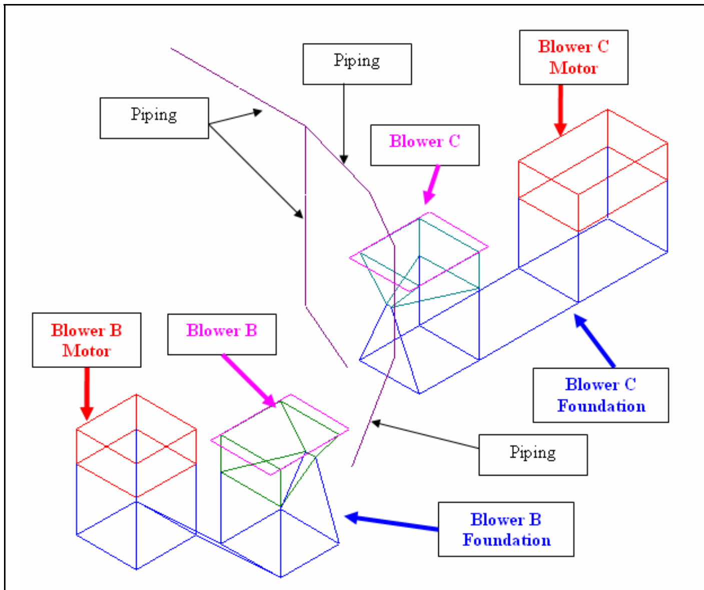


Photo 3: Reference Accelerometer Location on Blower C

Vibration data for operational deflection shape model was acquired by collecting "point-by-point" data on the two blowers and piping. This ODS data was collected while both blower units were running at steady state operating conditions. This ODS was performed to characterize the relative motion of blower components during "typical" vibration conditions of the blowers. A twenty four-channel Data Physics Abacus vibration data acquisition system was used to capture the vibration data from the two roving triaxial accelerometers and reference triaxial accelerometer. The vibration data acquisition system captures the frequency responses functions (FRF) of each roving triaxial accelerometer channel relative to the reference triaxial accelerometer signatures. The operating deflection shape vibration deflections were then computed from these measured FRF's.



Sketch A: ODS Geometry

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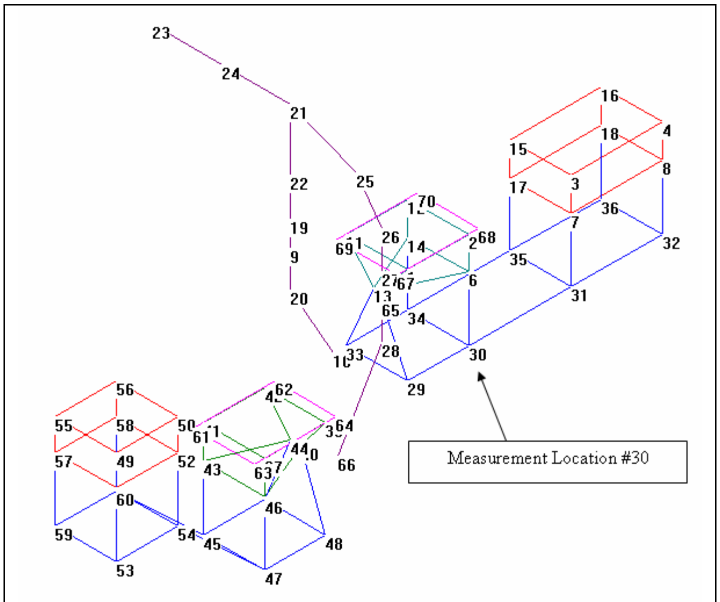

Sketch B: ODS Measurement Locations

To visualize the ODS deflections, a computer "model" of the geometry of the measured points was generated; see sketch A and B. ODS deflections for each measured location were applied to matching points in the computer model. Since the ODS deflections are very small compared to the dimensions on the machine, the amplitudes of the ODS are scaled and animated so that the motion was clearly visible on the computer screen. In other words, the ODS deflections in the model are greatly exaggerated to make them visually apparent; in short, the deflections are not display at the scale of the models itself. Hence, an ODS shows the relative motion and phase between locations on the blower components, not absolute "scaled" deflections of the machine. In the ODS figures, the black dotted lines are the undeformed ODS model shape. The solid lines are the animated ODS shape.

The operating deflection shape was assembled from vibration measurements acquired for 70 vibration measurement locations. Figures 7 through 14 illustrate the operating deflection shape at 60Hz, the frequency corresponding to the fundamental tone. This ODS shape clearly showed significant deflections occurring at piping sensor locations 9, 10, 19 through 28, and 65 with the motion being predominately vertically in the Z direction and axially in the X direction. This motion appears to be in reaction to pressure pulsations from the two running blowers; however dynamic pressure measurements would have to be performed to confirm this hypothesis. To arrest the piping motion, the piping could be constrained by pipe fixturing (snubber, pipe rolls, spring hangers…) or a tuned damper; however it was believed that these actions would probably not reduce noise levels appreciably. This solution would help reduce piping vibration, cracks and other potential fatigue damage (photo 4). The ODS also showed blowers B and C moving slightly vertically up and down (out-of-phase with each other) in what appears to be reaction to the exhaust pulsations at 3600 cpm and piping motion. Coherence measurements between blower and piping vibration showed good correlation (> 0.7) between blower and piping vibration at 3600 cpm. Sound measurements and blower vibration also showed good correlation (> 0.7) at 3600 cpm. These coherence levels (> 0.7) indicate a reasonable "cause and effect" relationship between sound and blower vibration and between piping and blower vibration (figure 15).

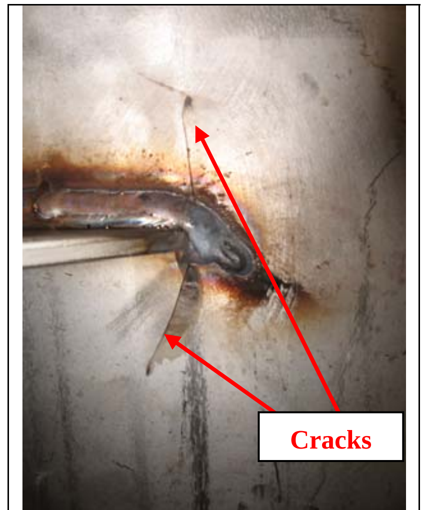

Photo 4: Pipe Crack

Structural modal measurements on the exhaust piping were attempted after acquisition of the ODS data. These measurements are useful in identifying whether structural natural frequencies coincide with vibration frequency components inherent with the blowers or piping resulting in a structural resonance. Resonant vibration typically amplifies the vibration response of a structure far beyond the design deflection levels predicted by computation based upon only static loading.

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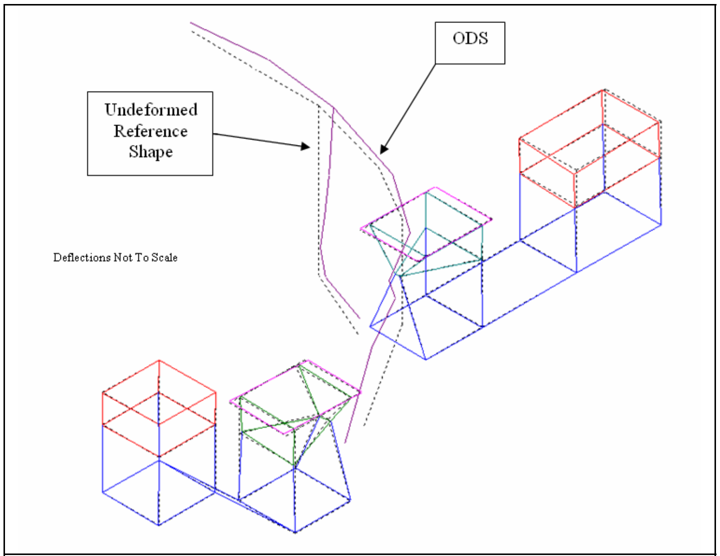

Figure 7: Operating Deflection Shape at 3600 cpm Isometric View

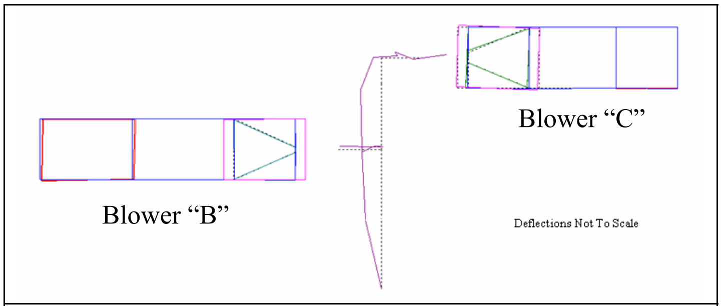

Figure 8: Operating Deflection Shape at 3600 cpm Top View

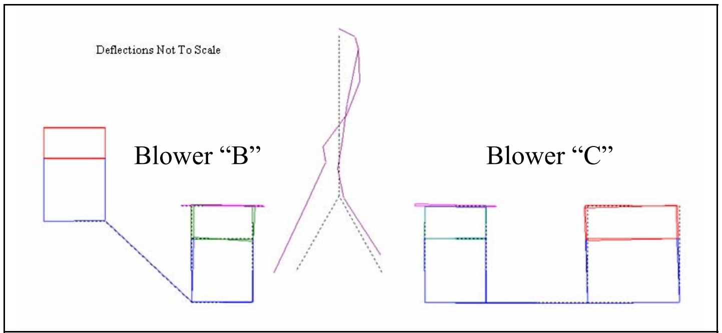

Figure 9: Operating Deflection Shape at 3600 cpm Lateral View (side view)

Since the blowers could not be shut down for more than 5 minutes without compromising the bacteria in the process, adequate modal measurements could not be performed to identify any natural frequencies of the piping or blowers. This was recommended as testing to be conducted during a future outage.

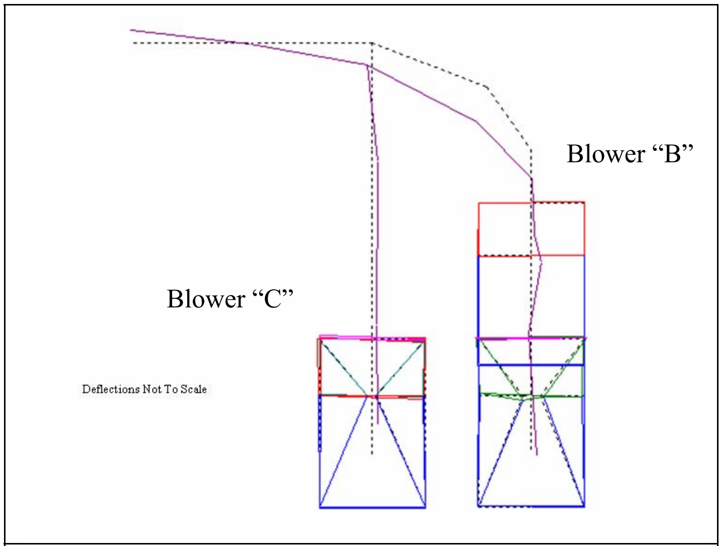

Figure 10: Operating Deflection Shape at 3600 cpm Axial View

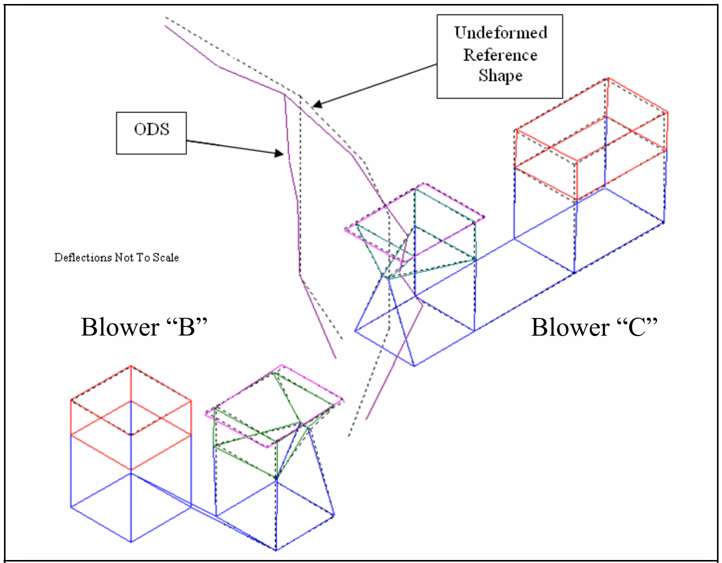

Figure 11: Operating Deflection Shape at 3600 cpm Isometric View

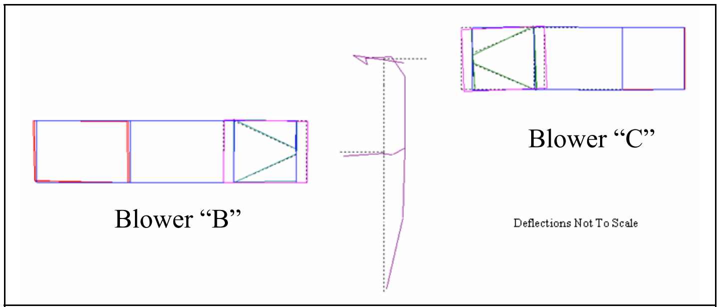

Figure 12: Operating Deflection Shape at 3600 cpm Top View

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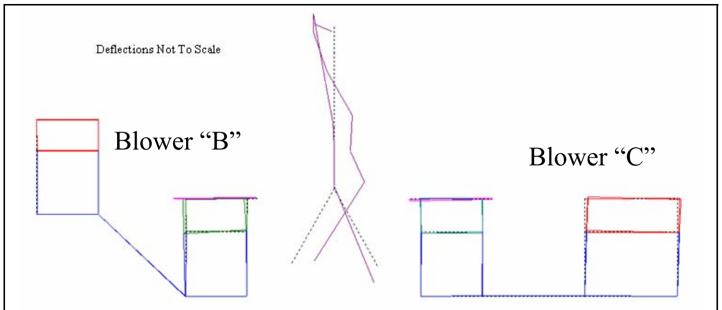

Figure 13: Operating Deflection Shape at 3600 cpm
Lateral View (side view)

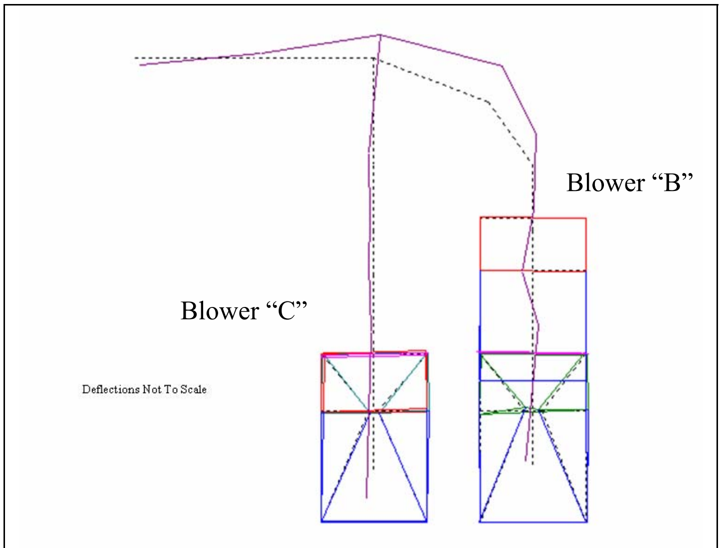

Figure 14: Operating Deflection Shape at 3600 cpm
Axial View

In late March, the best initial solution to both the vibration and noise problems appeared to be installing exhaust silencers on each blower. It was anticipated that reduced exhaust pulsation should occur which should also reduce piping vibration and exhaust generated noise. An acoustic standing wave due to blower exhaust pulsations appeared to be the root cause, but dynamic pressure measurements would be needed to confirm this conclusion.

Exhaust silencers were installed on each blower in April 2010, see photo 5. They had an immediate effect on sound pressure levels both inside the blower room and in the surrounding neighborhood. Sound pressure levels dropped at least 20dBA inside and outside of the blower room, making for a more pleasant environment for maintenance personnel. In-house PdM vibration measurements revealed that blower vibration levels were reduced by more than 50%.

In July 2010, Blower C experienced problems with silencer packing material escaping without any detectable mechanical damage to the silencer or its welds; see photos 6 and 7. Fortunately, this packing material was captured by a down stream screen. The silencer OEM is pursuing a root cause for the packing material issue and has deployed a differently designed silencer. Dynamic pressure measurements were recommended as further testing to be conducted during a future outage in order to measure pressure intensity of the acoustic standing wave and to provide insight into silencer packing issues.

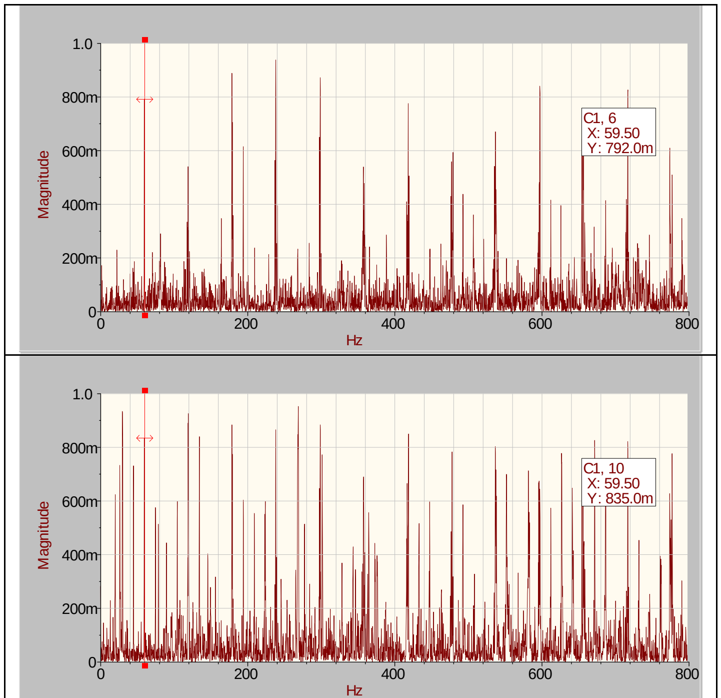

Figure 15: Coherence Spectra: Coherence Between Blower C at Reference Location (-Y) Direction and Piping Location 25 (-X) direction (top), Coherence Between Blower C at Reference Location (-Y) Direction and Microphone

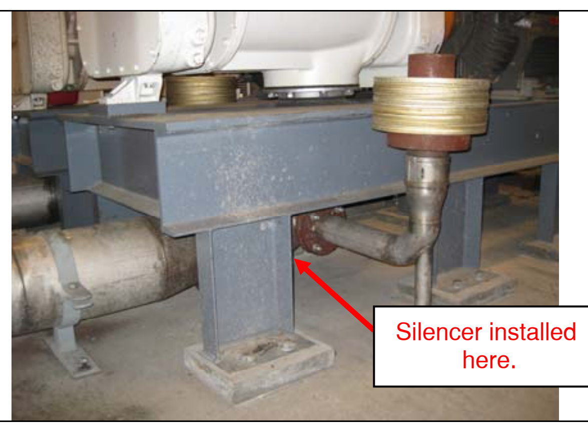

Photo 5: Location of Installed Silencers

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Photo 6: Silencer packing material retrieved from downstream screens prior to aeration tank

Photo 7: Disassembled silencer showing loss of packing material without physical damage to silencer

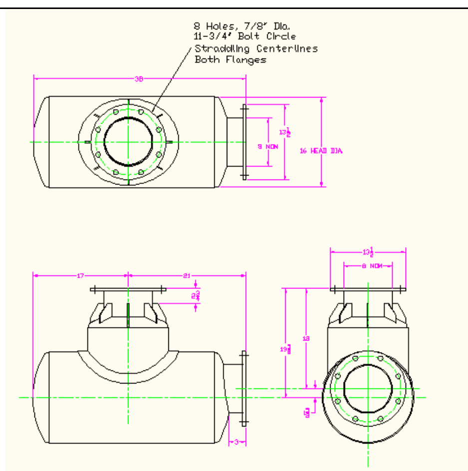

Figure 16: New silencer design

SUMMARY

The above case history clearly shows that integration of sound and vibration measures along with an operating deflection shape analysis and presentation of the vibration data proved to be an effective methodology for solving piping related issues.

An acoustic analysis of the sound measurements suggested that a potential acoustic standing wave might be present in a vertical section of each blower exhaust piping. This phenomenon suggested that exhaust silencers might be a potential solution. An operating deflection shape analysis of the vibration measurements clearly showed deflection patterns which also indicated that deploying exhaust silencers on the exhaust of each blower might reduce acoustic and vibration levels of the piping. Exhaust silencers were deployed and noise levels were significantly reduced, pipe cracking, check valve damage were also eliminated. In fact, acoustic sound pressure levels were reduced by 20 dBA; thus avoiding nearby resident's complaints concerning noise and making the immediate environment more pleasant for maintenance activities.

Original blower installation plans did not take into account the impact of acoustic standing waves occurring inside piping sections that were not proper sized. These plans did not also consider acoustic interaction of the blowers; which were to be run in 2 and 3 blower configurations. A cursorily review of piping section lengths and acoustical wavelengths would have identified suspect piping sections, highlighted the need for deploying silencers and resulted in piping design modifications before hardware was purchased and installed.

Economic impact of the potential cracked valves and piping while in service causing waste treatment plant downtime was estimated as exceeding $100,000 for 2010. Potential complaints about noise levels from nearby residents were also averted. By correctly identifying root cause(s) of the high vibration and noise levels, this cost was avoided.

REFERENCES

1. ISO-3746, "Acoustic – Determination of Sound Power Levels of Noise Sources Using Sound Pressure – Survey Method Using an Enveloping Measurement Surface Over a Reflecting Plane", 1995-08-15.

2. ISO-16032, "Acoustic – Measurement of Sound Pressure Level from Service Equipment in Buildings - Engineering Method", 2004.

3. Halliday, D., Resnick, R, "Physics: for Students of Science and Engineering", 1960, pg. 422-447.

4. Blevins, R.D., "Formulas for Natural Frequency and Mode Shape", 2001, pg. 340.

5. Yeon-Whan K, Young-Shin L., "Damage prevention design of branch pipe under pressure pulsation transmitted from main steam header", JMST, Volume 22, Number 4, Pg 647-652.

6. Bently, D. E., Hatch, C. T., "Fundamentals of Rotating Machinery Diagnostics", 2002, pg. 1-19.

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