AVEC, Inc
Acoustical and Vibration Engineering Consultants

3154 State Street, Suite 2230
Blacksburg, VA 24060
E-Mail: info@avec-engineering.com
Phone: 540-961-AVEC

Quieting of Continuous Miner Scrubber Fan Noise

Report: ***NIH12-Final Progress Report***

Grant/Contract #: 1R43OH010282-01

Performance Period: September 15, 2012 through December 15, 2013
Report Completion: December 10, 2013

Principal Investigator:
Kyle Schwartz, M.Sc.
AVEC Inc.
3154 State St, Suite 2230
Blacksburg VA, 24060
kschwartz@avec-engineering.com

Co-Investigators:
Patricio Ravetta, Ph.D.
Ricardo Burdisso, Ph.D.
Tyler Englerth

December, 2013

Notifications:

Funding for this project was made possible by the cooperative agreement award number above from the Centers for Disease Control and Prevention. The views expressed in this document do not necessarily reflect the official policies of the Department of Health and Human Services, nor does mention of trade names, commercial practices, or organizations imply endorsement by the U.S. Government.

Table of Contents

List of Terms and Abbreviations

CM	Continuous Miner
CMM	Coordinate Measurement Machine
BPF	Blade Pass Frequency
REL	Recommended Exposure Limit
PEL	Permissible Exposure Limit
NIOSH	National Institute for Occupational Safety and Health
OSHA	Occupational Safety and Health Administration
dB	Decibel
dBA	Decibel accounting for human hearing (A-weighting)
RPM	Revolutions per minute
NIHL	Noise Induced Hearing Loss
ADPAC	Advanced Ducted Propfan Analysis Code (CFD) Software
CFD	Computational Fluid Dynamics
CFM	Cubic Feet per Minute
SCFM	Standard Cubic Feet per Minute
HP	Horsepower
PR	Pressure Ratio
TE	Trailing edge
LE	Leading edge
Cd	Drag coefficient
CL	Lift coefficient
Vinf	Freestream Velocity
CR	Counter Rotating (modes)
M	Mach number (or circumferential mode order number)
Cr	Rotor average chord
Hz	Hertz

ABSTRACT

In the mining industry, Noise Induced Hearing Loss (NIHL) affects 90% of miners by age 50 [Camargo et al., 2008]. This is a consequence of exposure to noise above the Permissible Exposure Level (PEL) or Recommended Exposure Limit (REL) as defined by NIOSH, i.e. a dose of 100% or an 8 hour time weighted average of 90 dBA sound pressure level at the miner's ear. In the particular case of Continuous Miners (CMs), NIOSH [3] estimates that up to 86% of the CM operators are exposed to levels above the PEL. Also, excessive noise levels affect communication inside mines and create a hazard when operators cannot hear alarms. Although most mining related injuries and illnesses have been reduced in recent years, hearing loss is an exception. Reducing the noise from the CMs would significantly improve the working environment and therefore reduce the potential of NIHL.

AVEC's approach to achieve noise reduction in the CM is to target one of the major noise sources, the scrubber fan system used to remove harmful coal dust from the air. ***The goal of this project is to completely redesign the fan (rotor and stator) of the CM scrubber system's for reduced noise while keeping the required performance.*** The approach is to use modern computation tools and technologies, including some innovative technologies utilized for designing quiet propulsors for unmanned aerial vehicles such as ducted fans. The primary objective of this design effort is to reduce noise, and maintain the duct collection performance of the scrubber fan. Whereas, current scrubber fan designs generally only consider efficiency as the main design parameter. The design for noise and aerodynamic efficiency is an approach that AVEC has successfully implemented previously in different military and commercial projects where noise is a major concern, e.g. unmanned air vehicles.

A redesign of the rotor and stator pair was performed using modern tools and technologies. Significant noise reductions were made by selecting an appropriate rotor and stator blade count, increasing the spacing between the rotor and stator, reducing the stator chord length, and designing new airfoils for both the rotor and stator. In particular, the stator vanes are well designed airfoils as compared to the curved flat plate currently implemented in the CMs. Analytical/numerical studies were used at this stage for aerodynamic performance and noise predictions, i.e. no experiments or measurements. ***Based on these studies, the new advanced fan design produces 20 dB less noise and continues to produce the same flow rate and produce the same pressure rise (head) required for dust collection.*** Although, the entire 20 dB of noise reduction will not be observed due to other noise sources becoming dominant, such as the scrubber fan inflow distortion noise. These other noise sources will be addressed in the Phase II.

Section 1

Significant (Key) Findings

The specific aim of the project is to reduce the noise produced by continuous mining machines with focus on redesigning the fan, the major noise source. The current scrubber fan design has remained mostly unchanged for 20 years or more and was designed without considering noise as a factor. Some of the major sources of noise on the scrubber fan were found to be a result of the very short distance between the rotor and stator, the number of blades, and unnecessarily long and poorly designed stator vanes. Multiple inefficiencies were also found to exist in the baseline system. For example, a significant design flaw of the demister and ductwork geometry cause significant flow blockage resulting and a highly distorted inflow at the scrubber fan inlet. Additionally, the current acoustic liner treatment is not designed properly and is not effective. These system inefficiencies will be addressed in Phase II of this project.

Trade studies were performed to assess the effect of various fan parameters on the overall noise of the fan. Rapid design iterations were performed adjusting fan parameters that were determined to have the highest impact on the fan noise, i.e. seeking the quietest design in the trade studies. Significant noise reductions were made by selecting an appropriate rotor and stator blade count, increasing the spacing between the rotor and stator, reducing the stator chord length, and implementing an airfoil shape on the stator vanes, as opposed to the curved flat plate the baseline fan uses. Studies on the aerodynamic performance of the advanced fan was performed which showed that the current design continues to produce the same flow rate and produce the same pressure rise (head). Aerodynamic efficiency is mostly conserved for the advanced fan with only an overall decrease on the order of 2-3%, e.g. from 86% to 83%.

The current advanced fan design was found to be significantly quieter than the baseline configuration. Noise predictions show that the advanced fan is up to 20 dB quieter than the baseline fan. It is likely that not all of this noise reduction will be possible due to the existence of flow distortion in the baseline continuous miner ductwork. Preliminary estimations show that the overall noise could be decreased by at least 3 dB by simply correcting the inflow distortion with the existing fan. The combination of the new fan design and the correction on the inflow distortion has the potential to achieve the 20 dB noise reduction (or better) predicted in this project's study. Noise increases caused by the ductwork flow distortion are out of the scope of this Phase I project and will be investigated more closely in Phase II.

Translation of Findings

The findings of this project can be used to prevent workplace injuries such as Noise Induced Hearing Loss (NIHL) by the implementation/installation of the newly design scrubber fan and stator. The new scrubber fan design is not ready for implementation yet since it is only a design and a set of design guidelines. The Phase II project will redesign other scrubber fan system components that contribute to the noise produced by the fan, such as upstream disturbances interacting with the spinning fan, further refine the scrubber fan design, and build and test a prototype scrubber fan system to confirm predictions. AVEC has been closely working with Joy Mining Machinery, a leading manufacturer of mining equipment in the world, in this Phase I effort. In fact, AVEC has already partnered with Joy Mining Machinery and agreed to a licensing condition for the implementation of any future scrubber fan systems. This cooperation between AVEC and Joy has the main advantage of providing a direct path for the translation of technology and the potential for improvement in the wellbeing of miners in shortest amount of time.

The current advanced fan design is up to 20 dB quieter than the baseline fan. If all 20 dB of noise reduction is observed, the noise exposure of the CM operator would likely fall to below the MSHA PEL. This would eliminate the increased risk of NIHL and allow for a much safer work environment. Once the scrubber fan system design is completed in the Phase II, rapid implementation into the workplace can be achieved through new machines and retrofits.

Outcomes/Impact

Multiple beneficial potential outcomes are a result of this project. This project leads to improvements in occupational safety and health by directly reducing Noise Induced Hearing Loss for miners exposed to the continuous miner machine. The new scrubber fan design is projected to have up to a 20 dB noise reduction relative to the current scrubber fans in continuous mining operations. Considering the MSHA Permissible Exposure Limit, a 20 dB noise reduction would decrease the miner's noise exposure to below the MSHA PEL reducing the risk of NIHL. With a decrease in continuous miner noise, the surrounding workplace environment would be quieter. This quieter workplace environment would allow more efficient communication which is vital for worker safety in such a hazardous work environment. Moreover, mine owners/corporations will benefit from the lowered noise by reduced health costs related to NIHL and a more comfortable working environment. Furthermore, the findings in this study can be used to serve as a fundamental baseline for acoustically efficient ducted fan design, particularly for outdated heavy machinery. The trade studies and resulting correlations between fan parameters and overall noise performed within this Phase I project can serve as a first step to designing an acoustically efficient ducted fan.

Section 2

1. Introduction

This section (2) summarizes the technical work and tasks performed for the project "Quieting of Continuous Miner Scrubber Fan Noise", under NIH contract #1R43OH010282-01. This section is organized into chapters. The first introduction chapter discusses the nature of the problem/background, and the approach used to address the problem. The second chapter discusses the currently used scrubber fan (baseline), the aerodynamic and acoustic properties of the fan, and the deficiencies of the fan. The third chapter discusses the trade studies and investigations performed to reduce the noise. The fourth chapter evaluates a collection of new scrubber fan designs and their aerodynamic and acoustic performance. The fifth chapter discusses additional noise control options, such as acoustic liners and serrations. The sixth chapter discusses the impact of the noise reduction on the miner. The final chapters include a summary of the findings, recommendations for future work (Phase II), references, publications, and appendices.

1.1. Description of Problem and Background

In the mining industry, Noise Induced Hearing Loss (NIHL) affects 90% of miners by age 50 [1]. This is a consequence of exposure to noise above the Mine Safety and Health Administration (MSHA) Permissible Exposure Level (PEL). In the particular case of Continuous Miners (CMs), MSHA and NIOSH [2,3] estimate that up to 86% of the CM operators are exposed to levels above the PEL. Additionally, CMs were the number one machine among all mining equipment whose operators exceeded 100% noise dosage [4]. Also, excessive noise levels affect communication inside mines and create a hazard when operators cannot hear alarms. Although most mining related injuries and illnesses have been reduced in recent years, hearing loss is an exception. It estimated that at least 30 million workers are exposed to excessive noise levels and studies show that 70 to 90 percent of all miners have NIHL large enough to be classified as a hearing disability [5].These facts suggest that reducing the noise from the CMs would significantly improve the working environment and also reduce the potential of NIHL.

In order to reduce the noise generated by CMs, it is first necessary to determine the major noise sources. A typical approach is to rank the noise sources according to their loudness and to approach the noise reduction efforts starting with the loudest one. Several techniques can be used to determine (and thus rank) the loudest noise sources. One of such techniques is known as Microphone Phased Arrays (MPAs) [6]. This advanced approach implements a large number of sensors in a particular pattern and post-processing techniques to obtain an acoustic map. These results allow visualizing the major noise sources on a machine (overlaid with a picture or CAD drawing) as "red spots" in a contour plot. AVEC has collaborated in the past with NIOSH Pittsburg Research Laboratory (PRL) to use this technique on a CM and a Vibrating Screen [7]. In the case of the CM, the measurements were used to characterize the machine as well as to quantify the noise reduction obtained with a urethane-coated tail roller. Pictures of the CM and the test setup in PRL's hemi-anechoic chamber are shown in Figure 1 and Figure 2. Although the main goal was to quantify the noise reduction obtained with the newly designed tail roller, a few configurations were run with the scrubber fan ON. These results showed that the scrubber fan system is louder than the conveyor tail roller, even without the urethane coated tail roller (see Figure 3).

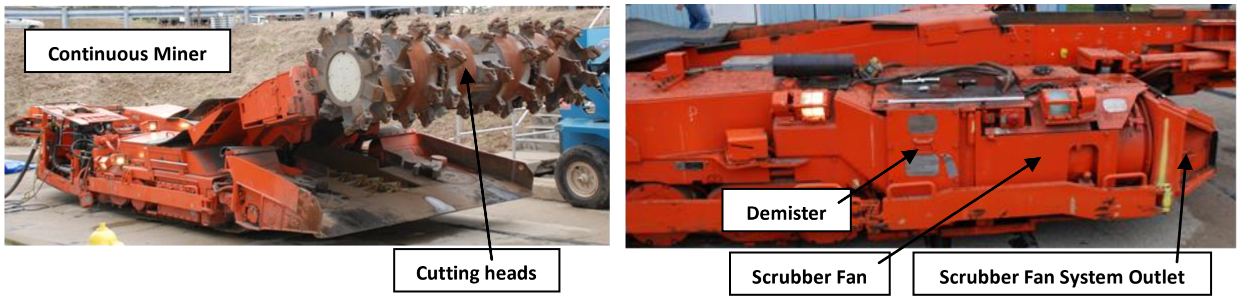


Figure 1: Pictures of the 14CM machine tested by AVEC using phased array measurements at NIOSH Pittsburg Research Laboratory.

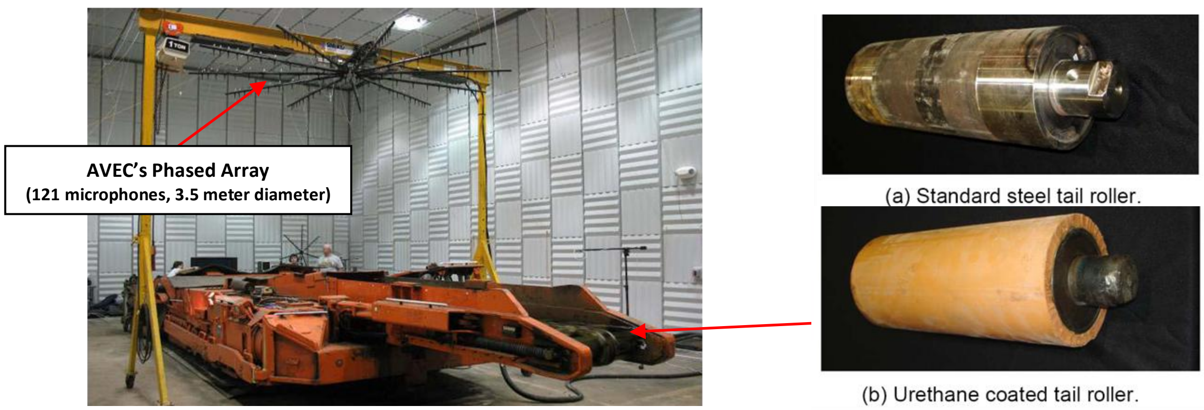


Figure 2: Test setup for phased array measurements of a continuous miner conducted by AVEC at NIOSH Pittsburg Research Laboratory. Detail of tested tail rollers.

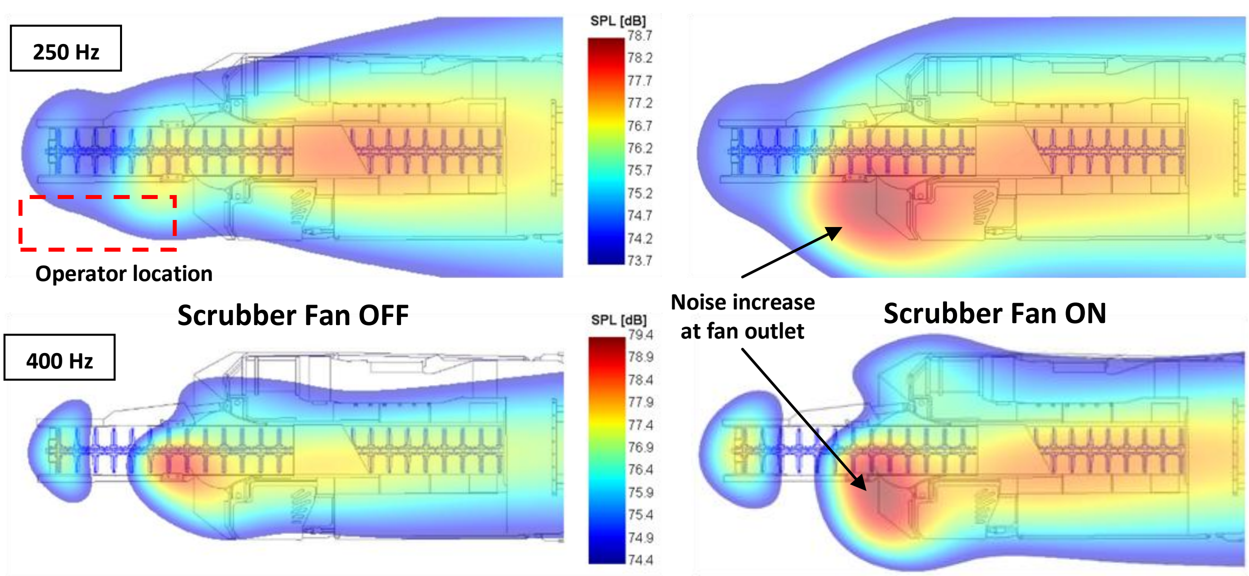


Figure 3: Sample acoustic maps for standard tail roller and scrubber fan OFF (left) and ON (right) showing noise increase at operator location due to scrubber system noise.

As a consequence of this and other noise control devices implemented in recent years, the noise from the scrubber fan system has become a dominant noise source in a CM. A picture of the currently used scrubber fan (rotor, stator, motor, and housing section) is shown in Figure 4. The scrubber system of a CM consists of a duct extending from the cutting heads to the back of the machine. The duct contains a fan and filter screens (see Figure 5 and Figure 6). Previous noise measurements of the scrubber system and a preliminary study of the acoustic sources conducted by AVEC suggest several opportunities for noise reduction.

Figure 4: Scrubber fan currently used on Joy CM.

Figure 5: Scrubber system of a Joy Mining Machinery CM (Susey, 2008).

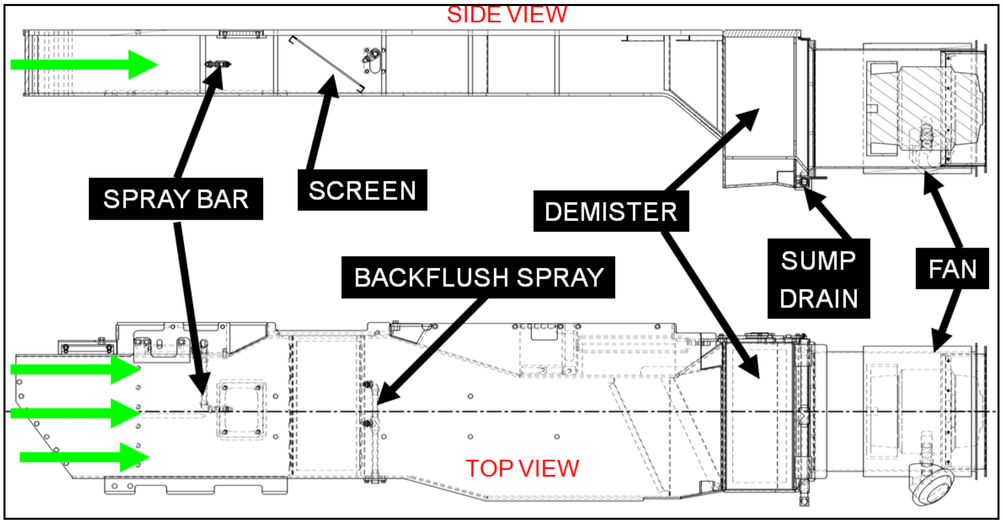


Figure 6: Diagram of scrubber system of a Joy Mining Machinery CM.

Although several noise control devices have been proposed in previous work, it is believed that in order to obtain a significant noise reduction the problem needs to be attacked at its root, the fan itself. This fan noise represents a minimum noise level that can be attained even if all other noise sources were reduced or removed.

For several years, the industry's design objective for this type of fan was high efficiency and reliability. However, the fan was not designed with low-noise as a priority. Although acoustic performance and efficiency are often in opposition, technologies developed in the last 20 years and the availability of modern computational tools allow for reduction of noise with minimal impact on efficiency. The authors of this report have developed a similar approach for ducted fans and open rotors. As a result, previous projects have proven that significant noise reduction can be achieved while maintaining aerodynamic efficiency (2-3% reduction in efficiency). As in many applications, the trade-off between noise and efficiency must be set as an objective. This type of objective would be determined in collaboration with the industry to ensure that the final product can be implemented beyond the research stage. Joy Mining has made the commitment to support and participate in the Phase II project (see letter of support, Appendix B). This early collaboration with industry suggests that a successful scrubber fan design would rapidly transition to the mining field and therefore quickly benefit workers in the mining industry.

As a testimony to the high noise levels in underground mines, AVEC was recently contacted by an Industrial Hygiene specialist requesting assistance to reduce scrubber fan noise on their underground CM. The actual email received from this Industrial Hygiene specialist is shown in Appendix A. Discussions with this industry specialist confirmed that the scrubber fan noise was indeed the highest level noise source in their mines. This specialist discovered this Phase I research abstract when researching the high noise levels of the scrubber fan system online.

1.2. Objective/Approach

The **objective** of this Phase I work is to conduct a proof-of-concept design for a particular CM using state-of-the-art computational tools and noise control technologies. The newly designed fan properties (aerodynamic and acoustic) are compared to those of a current CM fan to determine the potential of the proposed approach.

The **approach** to achieve noise reduction in the CM is to target one of the major noise sources, the fan (rotor-stator) of the scrubber duct system. Although several attempts have been made over the years to reduce the noise from CMs by acoustic treatment and other small modifications, the fan noise itself has not been targeted in previous studies, e.g. control at the source. Furthermore, most of these fans were designed over 20 or 30 years ago. The main problem with the current fan is that it was designed with efficiency, pressure rise and reliability in mind, ***while not considering noise as a major design variable***.

It is the hypothesis that completely redesigning the fan (rotor and stators) using modern computational tools and technologies would have a very significant impact on the noise. In this process, low noise will be considered the major target. This is an approach that AVEC has successfully implemented in different military and commercial projects where noise is a major concern, e.g. unmanned air vehicles. Even in these highly-demanding applications it was found that aerodynamic efficiency can be kept high while significantly reducing noise.

The first task in this project is to analyze the current scrubber fan design to define a baseline. The analysis of the baseline fan includes both aerodynamic and acoustic performance. Full analysis of the baseline design allows for a metric to compare new designs against. The second task is to perform trade studies to determine the acoustic effect of varying fan properties. Multiple parameters of the scrubber fan are considered available for change, such as: rotor and stator blade count, rotor-stator spacing, hub to tip ratio, chord distributions, and others. Some of the parameters are assumed to be held constant, such as: aerodynamic performance (flow rate and pressure rise), rotor rotational speed (3600 RPM), and rotor tip diameter. The next task is to refine the studies and make a set of actual designs. Of these preliminary designs, the quietest and most efficient design will be selected and full computational fluid dynamic (CFD) analysis will be run. This CFD solution will be compared to the full analysis performed on the baseline fan to estimate the overall noise reduction. In addition to redesigning the scrubber fan, other noise sources are explored. Preliminary studies on acoustic liners and Howden Buffalo experimental data detailing the flow characteristics of the scrubber system are taken into account.

Note that the advanced scrubber fan design from this Phase I effort should not be considered fully optimized. True optimization will be performed in the Phase II accompanying a complete design of the scrubber fan system, i.e. filter screen, demister, rotor/stators, exhaust, and acoustic liners.

1.3. Review of Previous Work

For completeness, the previous work performed on the topic of noise reduction has been reviewed and summarized in this section. The first major study was performed for and summarized by the master's thesis of Benjamin Carter at Pennsylvania State University (PSU) in 2003 [8]. The second major study was performed by Howden Buffalo in 2008 [9].

The thesis by Carter investigated various methods to reduce both broadband and tonal noise. A 3600 RPM, 7000 CFM, 30 HP vaneaxial fan was used in this analysis. This vaneaxial fan consisted of a 12 blade, 3.68" chord rotor and a 15 blade, 10.5" chord stator. The combination of 12 rotor blades and 15 stator vanes will be represented in this document as 12/15 (shorthand). The currently used scrubber fan provided by Howden Buffalo for the current project consisted of a 12 blade, 5.62" chord rotor, a 15 blade, 9.8" chord stator, and featured a perforated housing downstream of the rotor trailing edge. This partially perforated housing is shown in Figure 7. Various fan housings were also analyzed in Carter's thesis. It is important to note that all experiments performed for Carter's thesis did not consider the aerodynamic performance of the fan. Primary methods to reduce broadband noise were through sound absorption materials and the reduction of turbulence. Absorption of sound was performed via absorptive foam (acoustic blanket) integrated into the fan housing system. Reduction of turbulence was performed via installing an inlet shroud and a discharge extension at the fan outlet. The inlet shroud was a converging, circular piece that was bolted to the inlet of the vaneaxial fan in order to smooth the air flow. The discharge extension was bolted to the outlet of the vaneaxial fan and removable deflecting vanes which direct the discharge airflow. However, these reduction techniques (inlet shroud and discharge extension) would not be available for implementation in the continuous mining machine due to space restrictions; these techniques were investigated to highlight the importance of flow management. The methods to reduce tonal noise were: increasing the spacing between the rotor and the stator, varying the rotor blade circumferential spacing (modulation), and modifying the stator vane leading edge geometry.

Figure 7: Baseline fan casing for PSU tests featuring partially perforated housing and acoustic blanket/foam.

Experimental testing of the above mentioned noise reduction techniques led to the following conclusions. Absorptive foam integrated into the fan system was found to reduce broadband noise levels. Specifically, foam wrapped around the fan housing and foam lining the discharge extension produced the largest noise decrease. The installation of the inlet shroud and discharge extension sufficiently smoothed the airflow to reduce turbulence ingested by the fan. This reduction in turbulence slightly decreased the noise but more importantly highlighted the importance of flow management. It was found that smooth airflow (i.e. minimizing ingested turbulence into the fan from the scrubber/demister) reduced noise. It was found that increasing the distance between the rotor trailing edge and stator leading edge reduced noise as well. Unevenly spaced rotor blades (i.e. modulated blades) produced strong tones at harmonics of the shaft rotational frequency as well as the blade passage frequency. Evenly spaced rotor blades produced tones primarily at harmonics of the blade passage frequency. Due to this, it was suggested to abandon the modulated blade spacing in favor of an evenly spaced rotor. The stator vane leading edge geometry affected the amount of radiated noise at harmonics of the blade passage frequency. Modifying the stator vane leading edge from a straight edge perpendicular to the airflow to a diagonal profile (i.e. vane sweep) reduced the tonal noise. Additional literature is available on noise reduction through modifying vane geometry and fan sweep [10, 11, 12].

It should be noted that the absorptive foam (acoustic blanket) used as an acoustic liner in this study is not a traditional acoustic liner design, i.e. there is no solid backing to the liner to prevent sound leakage, and the wall perforations and wall thickness are extremely large (1/4”) and thick (1/4”) for acoustic liners. More discussion of the deficiencies of the PSU acoustic liner (and currently implemented acoustic liner) is in section 5.1.

The results and conclusions from the Carter thesis were used to create a new fan housing. The new housing was manufactured by Howden Buffalo and featured increased rotor-stator spacing, stator leading edge sweep, and a fully perforated housing (with perforations and absorptive foam extending over the rotor plane). Original tests using this housing were performed for the thesis. Tests were also performed using this housing at a Howden Buffalo facility in 2008 [9]. A review of the measurements and results are shown here. A picture of one of the tests with the previously described PSU fan housing manufactured by Howden Buffalo is shown in Figure 8. The primary objective of the testing was to determine the effect of scrubber ductwork and fan interaction on aerodynamic performance and outlet noise levels. Another objective of the test was to determine the aerodynamic performance and acoustic impact of modifying the existing ductwork or fan in order to reduce fan noise levels or improve aerodynamic performance. Various duct-filter screens were tested with and without an acoustically lined discharge insert and discharge silencer supplied by Joy. A spool piece (straight pipe extension) between the scrubber discharge and the fan inlet was also investigated.

Figure 8: PSU fan casing featuring a fully perforated housing.

The overall noise results from the tested configurations are shown in Table 1. The most pertinent combinations (serial numbers are 8022A, 8048, 8049, 8049A, and 8050A) are shown in Figure 9. Results show that noise is reduced by 7 dB when the discharge insert and silencer are added. It was found that the addition of the spool piece (additional spacing) decreased noise by about 3 dB (8048B to 8049B). The results also showed that the addition of the scrubber and filter increase noise by 3 dB (8022A vs 8048B). The cause of the increased noise is very likely the flow distortion created by the demister's 4" blockage along the bottom of the duct and 1" blockage across the top of the duct. The duct cross-section and resulting duct blockage of the demister outlet are clearly seen in Figure 10. Implementation of the PSU fan casing resulted in at least another 3 dB sound level decrease at the cost of fan performance [17]. Recall that the PSU fan casing not only had additional absorptive foam, but it also had a different rotor-stator blade count and additional rotor-stator spacing.

Table 1: Howden Buffalo scrubber fan test matrix and results.

TSN	Scrubber & filter	Spool piece	Fan	Insert	silencer	Flow CFM	Overall dB
8022A	N	N	Y	N	N	7913	113
8048	-212	N	Y	N	N	7914	116
8048A	-112	N	Y	N	N	8156	116
8048B	-512	N	Y	N	N	8097	116
8049	-512	N	Y	Y	Y	7803	109
8049A	-512	Y	Y	Y	Y	8059	110
8049B	-512	Y	Y	Y	N	8067	113
8050	-512	Y	Y*	Y	N	8017	110
8050A	-512	Y	Y*	Y	Y	8008	106
8050B	-512	Y	Y*	N	Y	8008	106
* Fan is PSU fan casing with 505551-1516 rotor							

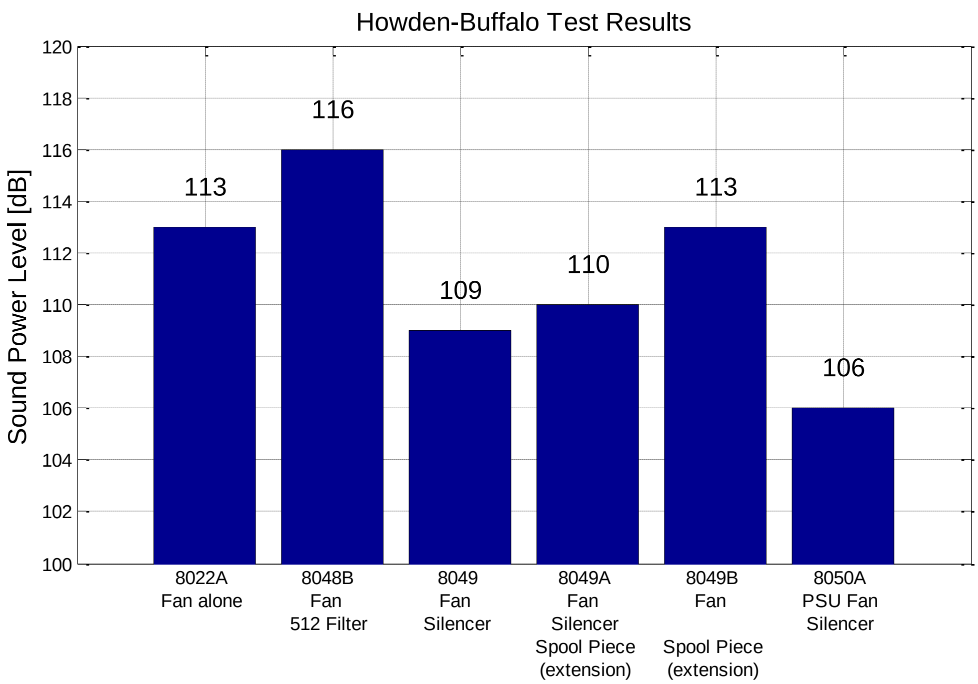


Figure 9: Howden Buffalo noise test results.

Figure 10: Spool piece inlet cross-section and resulting duct blockage (photo by Howden Buffalo Inc.).

Test configuration # TSN8022A tested just the fan with the outlet duct while all of the other tests included fan and scrubber ductwork interaction. Comparing test TSN8022A to test TSN8048 shows that the scrubber to fan interface is a large source of noise (+3 dB) due to poor flow characteristics. The 3 dB increase indicates the flow distortion interacting with the fan is as loud as the fan itself. A Pitot tube traverse of the airflow at the scrubber outlet showed that the current scrubber outlet vanes discharge the air at approximately 20° relative to the centerline of the duct [9]. Howden Buffalo concluded that flow distortion is the major factor in the increase in system sound. **At least a 3 dB noise reduction appears to be possible by simply straightening the flow leaving the scrubber/ demister and eliminating the duct blockage** [17]. Potential options for this are either a redesign of the scrubber itself or inserting a flow straightening device.

Howden Buffalo recommends using turning vanes to straighten the flow leaving the scrubber [18], however, this upstream vanes can also produce distortion and noise. This is the reason that all commercial aircraft engines with upstream vanes to straighten the flow were put out of service. Additionally, it was recommended that the corner blockage on the top and bottom of the demister should be decreased. Also, the orange screen guard visible in Figure 10 can be removed with the addition of flow turning vanes. A square to round transition combined with flow straightening vanes located at the square end could potentially eliminate flow irregularities. ***Note that the approach recommended by AVEC in the Phase II is to redesign the scrubber system (duct, demister, etc.) itself such that the outlet flow is straight rather than to make "corrections" to the existing one.***

2. Analysis of Baseline Scrubber Fan

The physical, aerodynamic, and acoustic properties of the baseline scrubber fan must be determined. This section details these characteristics of the baseline scrubber fan.

2.1. Description of Baseline Fan

The currently used scrubber fan system and 2 additional rotors were received on 3-27-2013. The scrubber fan was equipped with the standard flow rotor while the 2 additional rotors were for high flow and low flow applications. The 2 additional rotors (high and low flow) were not analyzed at this stage. The supplied scrubber fan has 12 rotor blades and 15 stator vanes. The scrubber fan is shown in Figure 11. The standard flow rotor, stator, housing, and other properties of this currently used scrubber fan system are also referred to as the “baseline” fan. Also note that the combination of 12 rotor blade and 15 stator vanes will be represented in this document as 12/15 (rotor blades/stator blades: used as shorthand).

Figure 11: Scrubber fan currently used in CM.

The fundamental rotor properties were measured with a coordinate measuring machine (CMM) at Techsburg on 3-28-2013. Figure 12 shows the CMM. Measurements of the stator vanes were primarily performed using technical drawings.

Figure 12: Coordinate measurement machine

Table 2 shows various properties measured by the CMM for the baseline normal flow scrubber fan as well as other parameters measured from technical drawings. It is worth noting that the rotor and stator chords tend to vary linearly from the hub to tip. A picture and diagram of the rotor airfoil is shown in Figure 13. This airfoil has a very blunt/rounded trailing edge (best seen in the diagram).

Table 2: Properties of baseline scrubber fan system.

Property	Value
Hub Radius	8.58”
Tip Radius (casing/housing)	10.59”
Rotor chord	5.39” (hub), 5.85” (tip)
Stator chord	9.82” (hub), 9.78” (tip)
Rotor-stator hub spacing	1.7”

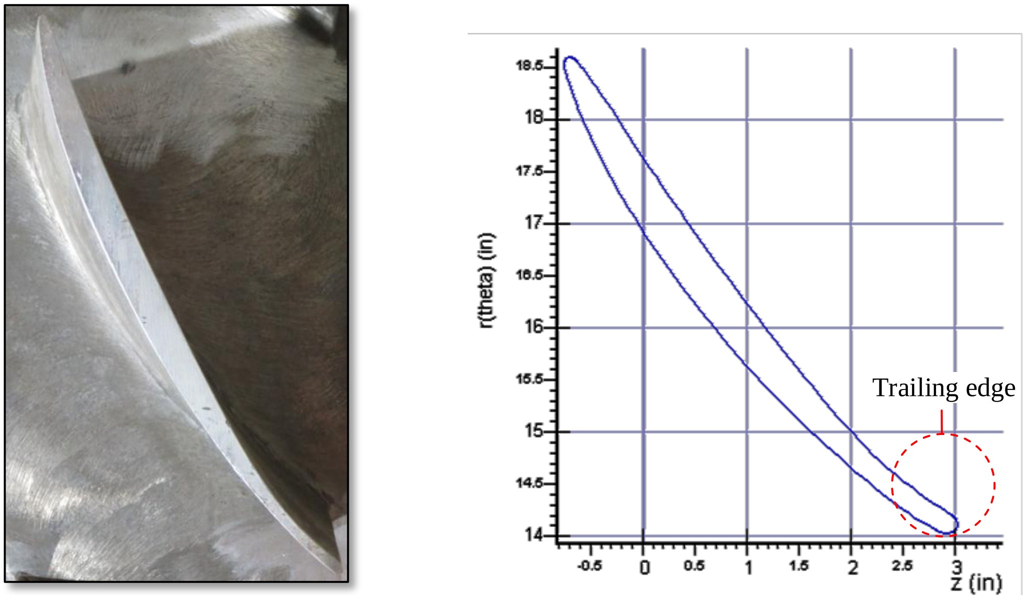


Figure 13: Baseline physical rotor airfoil and measured rotor tip airfoil.

The baseline scrubber fan consists of a 12 blade rotor and 15 vane stator. The axial spacing at the hub between the rotor and stator is 1.7” (or 30% of rotor chord). This standard flow scrubber fan operates between a flowrate of 6000 to 9000 CFM. These flowrates produce a static pressure rise between 15.5 and 5 IWG respectively. Additionally, the total pressure rise across the stage is between 16 and 6 IWG, respectively. Any new/advanced fan design produced from this project must be able to operate at these conditions or better (i.e. higher flowrates and higher pressure rise).

Explanation of fan variable flowrate and pressure rise: Fan flowrate is set to the maximum 9000 CFM at the beginning of a shift. As the demister separates the particles from the flow, the system resistance increases thus increasing the static pressure rise that the fan must operate. There comes a point when the fan cannot produce enough torque to pull the flow through the system. This occurs when the system resistance reaches a maximum. The system stalls at this pressure rise and the demister filter must be emptied and cleaned. For the baseline fan, the stall point is at 5913 CFM at a total pressure of 16.10 IWG.

2.2. Aerodynamic Analysis of Baseline Fan

For the study of the baseline rotor and stator, a CFD solution of the baseline rotor and stator were computed using CFD. The CFD solution was calculated by Techsburg in ADPAC. The CFD solution was computed using the conditions shown in Table 3. The flow being ingested by the rotor is assumed to be clean (well behaved) and straight. The solution includes the effect of the spinner and stators. The CFD solution at 25%, 50%, and 75% span are shown in Figure 14. Figure 15 shows the CFD solution without streamlines at the 50% span along with a second view showing the inflow flow velocity and streamlines. The normalized axial velocity for a rotor blade and stator vane is shown in Figure 16. The pressure and streamlines for a rotor blade and stator vane are shown in

Figure 17. The reference pressure is atmospheric. The rotor wakes were extracted from the CFD solution at two locations, directly at the rotor trailing edge (TE) and directly at the stator leading edge (LE). The solution at these locations are shown in Figure 18A and B. The color scale is the axial velocity normalized by the freestream velocity (141.7ft/s). The wake profile at the rotor trailing edge is more traditionally shaped than the wake at the stator LE. Although the wake can still be seen in the stator trailing edge plot (Figure 18B), the low velocity (blue) section at the tip of the stator trailing edge plot indicates that a tip vortex was formed at the rotor and is propagating downstream.

Table 3: Parameters used in the computation of the CFD solution.

Parameter	Value
Flow	8000 SCFM
P_0 rise	19 in. H_2O
$\eta_{adiabatic}$	91.3%
Torque	37.9 ft-lb
Power	26 HP

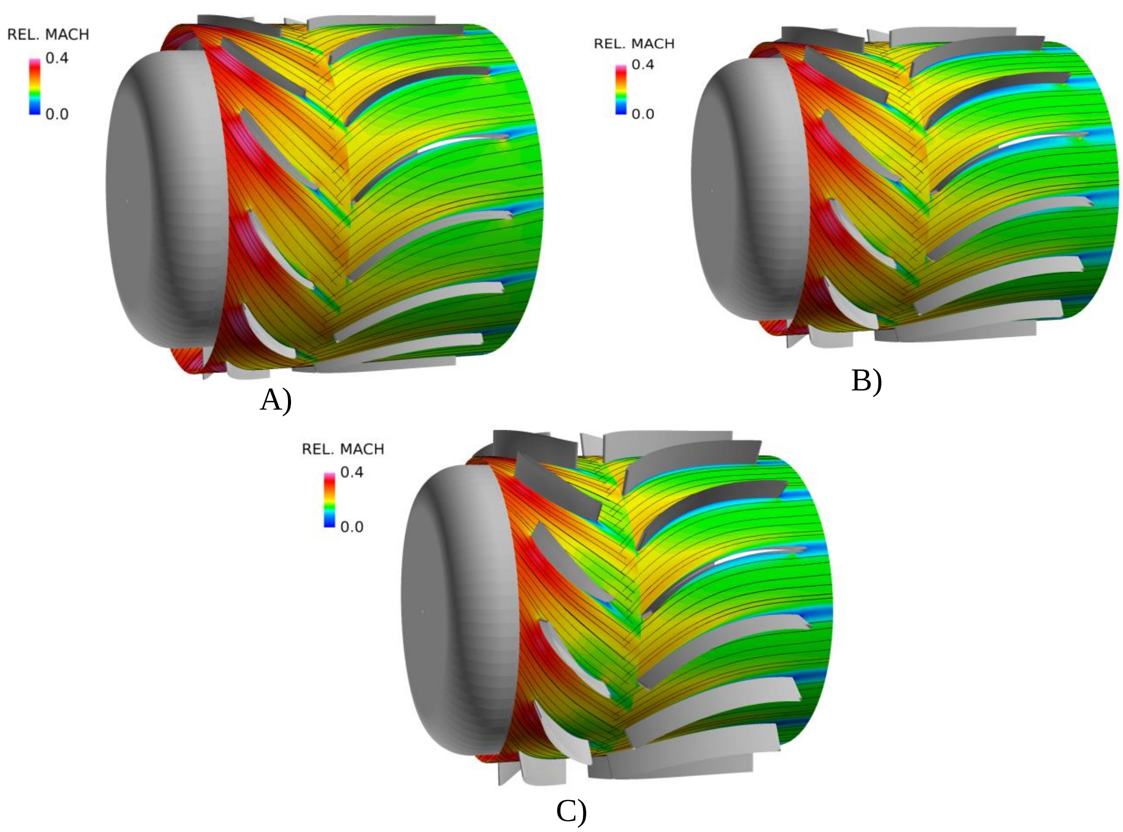


Figure 14: CFD solution of the baseline fan and stators at A) 25% radius, B) 50% radius, C) 75% radius. All shown with flow streamlines.

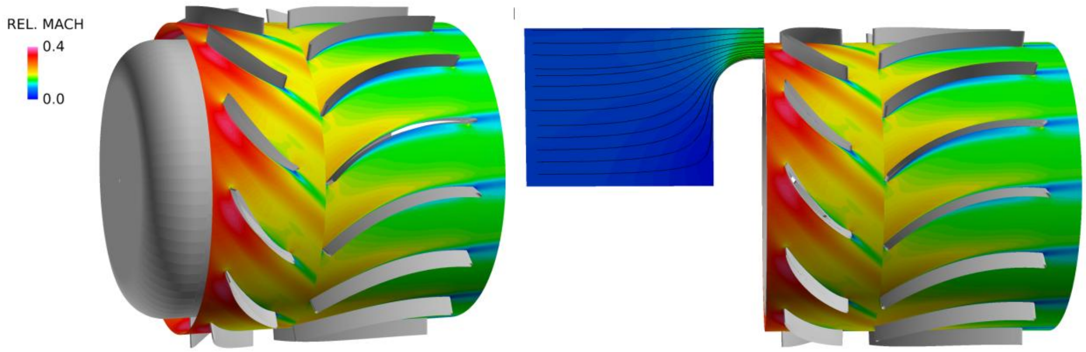


Figure 15: CFD solution of the baseline fan and stators at 50% radius.

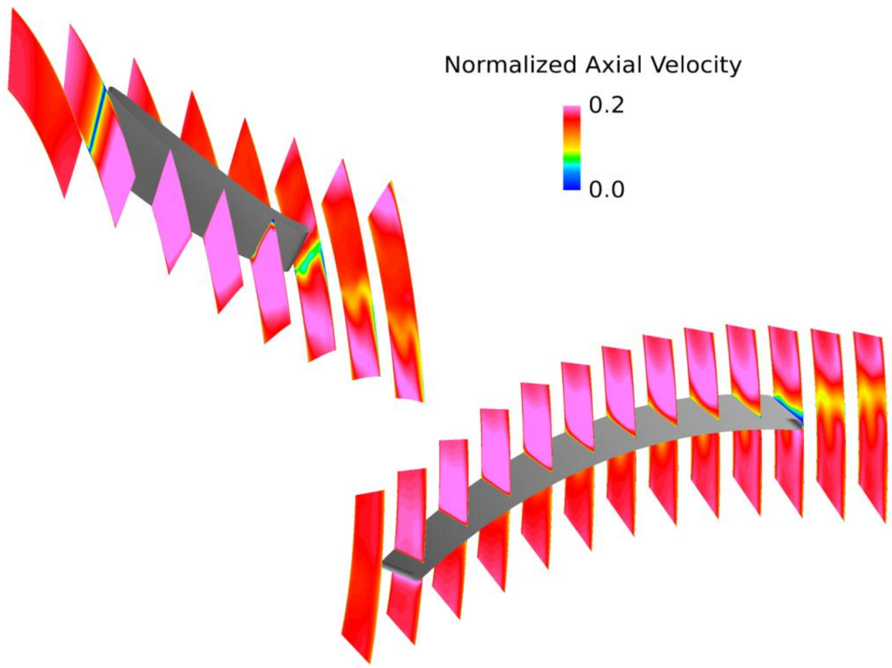


Figure 16: CFD axial velocity profile at multiple axial locations along the rotor and stator. Axial velocity is normalized to 943 ft/s

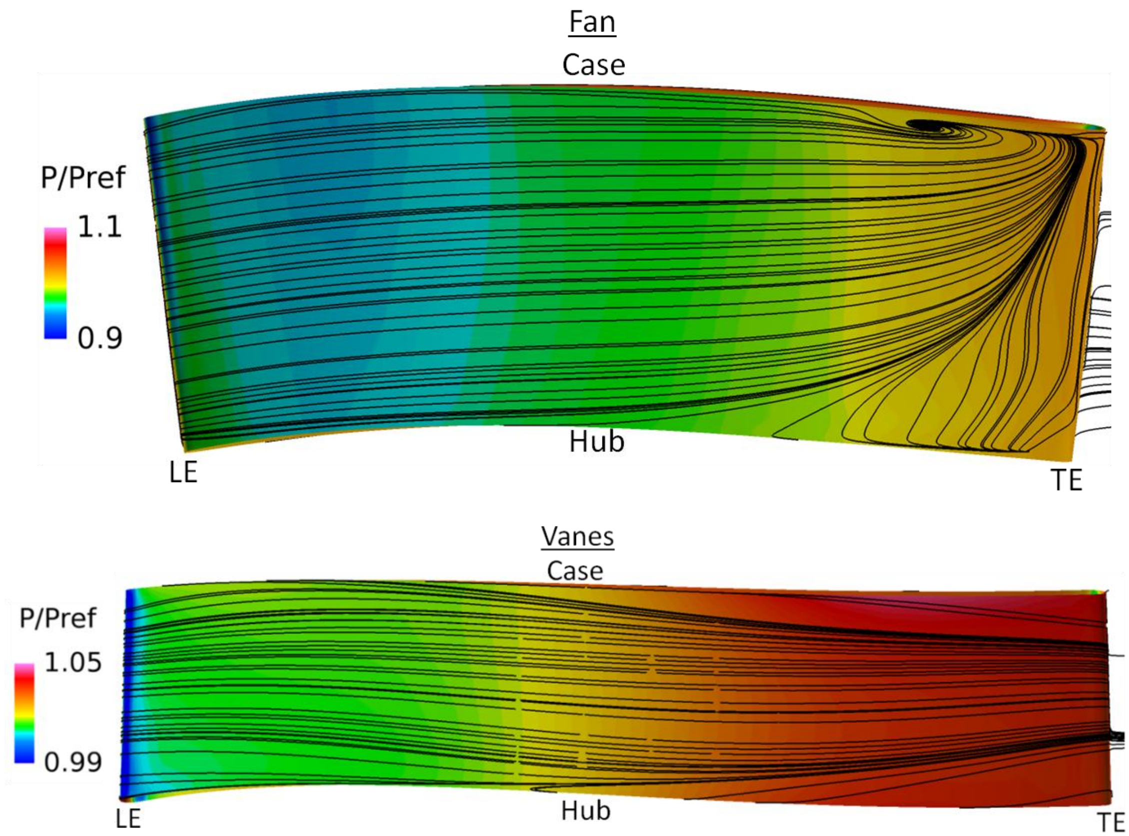
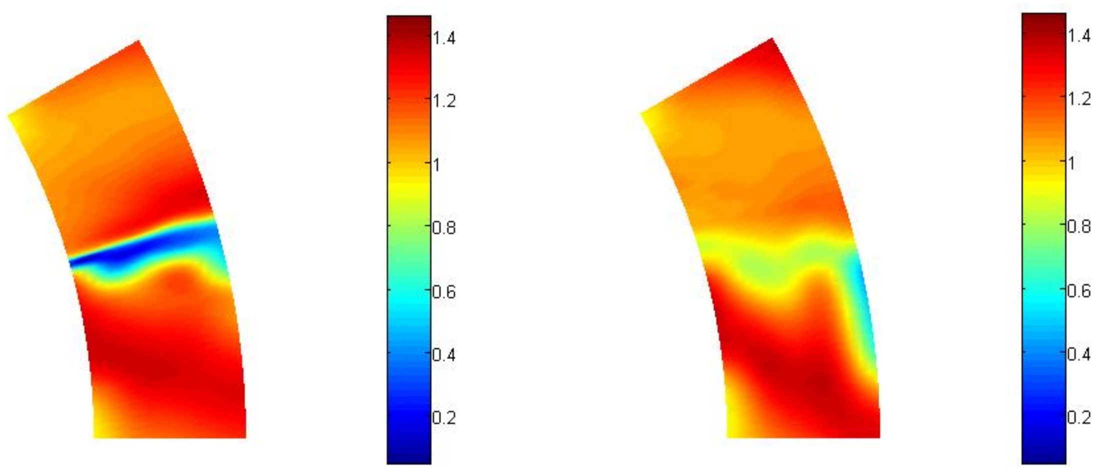


Figure 17: The pressure and streamlines on the suction surface of a rotor blade and stator vane.



A) B)

Figure 18: CFD solution at the A) rotor TE and the B) stator LE. The scale is Vaxial/Vinf.

The baseline fan was found to produce 8000 CFM at 14.0 IWG (total pressure rise). Note that this value is slightly different (higher) than the nominal values of the baseline fan obtained from the specification sheets (Joy). This is due to slightly different definitions of pressure rise and differences between CFD and actual measurements.

2.3. Acoustic Analysis of Baseline Fan

The noise levels of the baseline scrubber fan system were predicted using proprietary AVEC noise prediction tools. Noise was predicted for each source and then combined to represent the total noise from the scrubber fan. The sources predicted were the rotor-alone broadband, rotor-alone tones, and rotor-stator interaction tones. These noise sources are traditionally the most prominent for axial fans [13,14] Although other sources may exist in the system, they are not predicted for one of two reasons: the noise source is not part of this Phase I project (i.e. the grazing flow noise over the perforated housing, or the interaction noise caused by the inflow distortions), or the source is traditionally very quiet. For example, literature indicates that the rotor-stator broadband noise source is traditionally 20 dB below the rotor-alone broadband [22]. The noise levels predicted for the baseline fan are shown in Figure 19 and Figure 20. The rotor-stator tonal noise (yellow) is by far the dominant noise source. The rotor-alone broadband (blue) is the second most important source, while the rotor-alone tonal noise is predicted to be very low. The noise source that contributes the most to the overall A-weighted noise level of the scrubber fan is the rotor-stator tones. Specifically, the 2nd BPF of the rotor-stator interaction noise is very high.

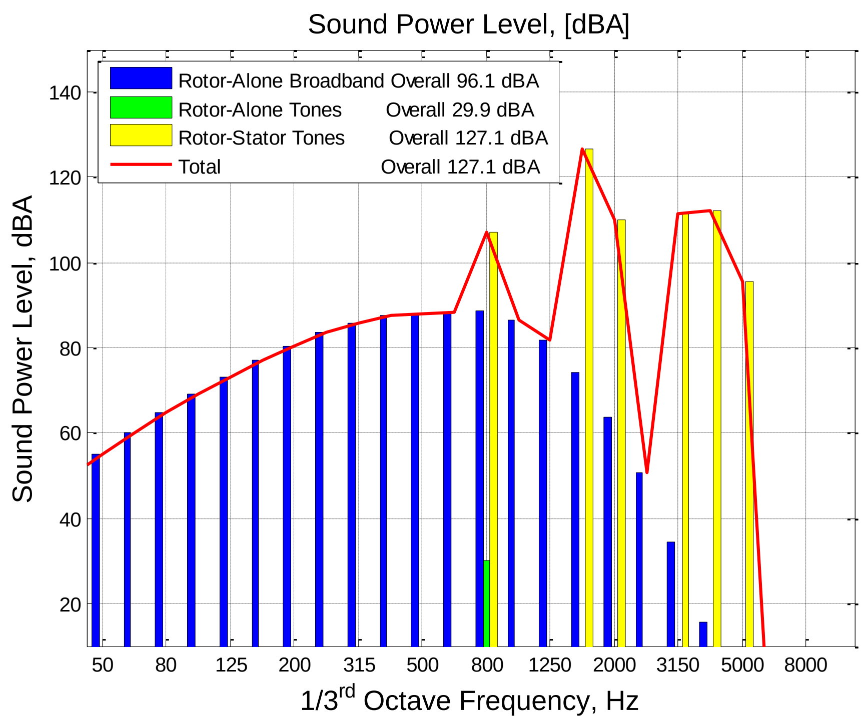


Figure 19: Noise levels for the baseline scrubber fan system shown in third octave bands. Note, noise due to **flow distortion is not modeled** or predicted here.

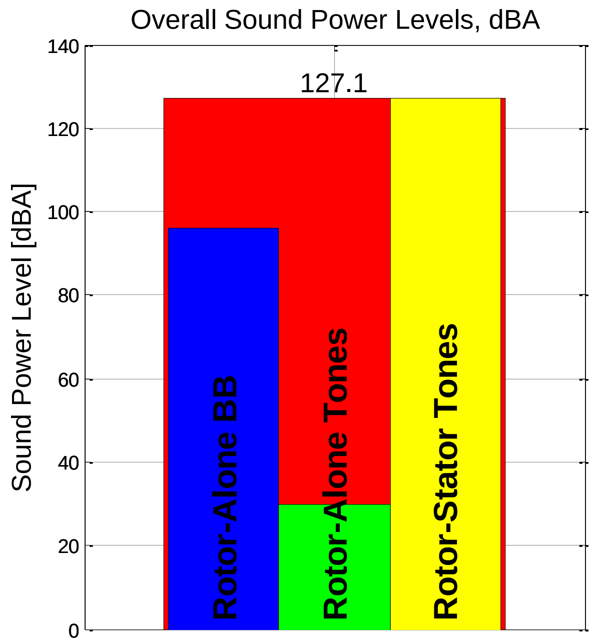


Figure 20: Overall noise levels for the baseline scrubber fan system, broken down by predicted noise source. Note, noise due to **flow distortion is not modeled** or predicted here.

Actual measured data of a continuous miner has been made available to AVEC as part of this Phase I investigation for comparison with the above noise predictions. Two measured data sets have been obtained: one data set measured by Howden Buffalo in 2008 (see previous work section), and one data set measured by NIOSH PRL at their facility. This data is shown here to demonstrate the high noise levels of the scrubber fan. The data is also compared against AVEC noise predictions as a validation of the noise prediction tools. The measured and predicted spectrums of the baseline fan are shown in Figure 21. Although the measured and predicted spectrums do not match perfectly, the key characteristics of the spectrum are captured well. For example, the first few BPFs of the predicted and measured spectrums match, and the low frequency broadband noise matches well. The overall noise levels, shown in Figure 22, are within 3.3 dB for the Howden Buffalo and predicted levels. Also, the predicted noise levels are higher than the measured which is considered conservative.

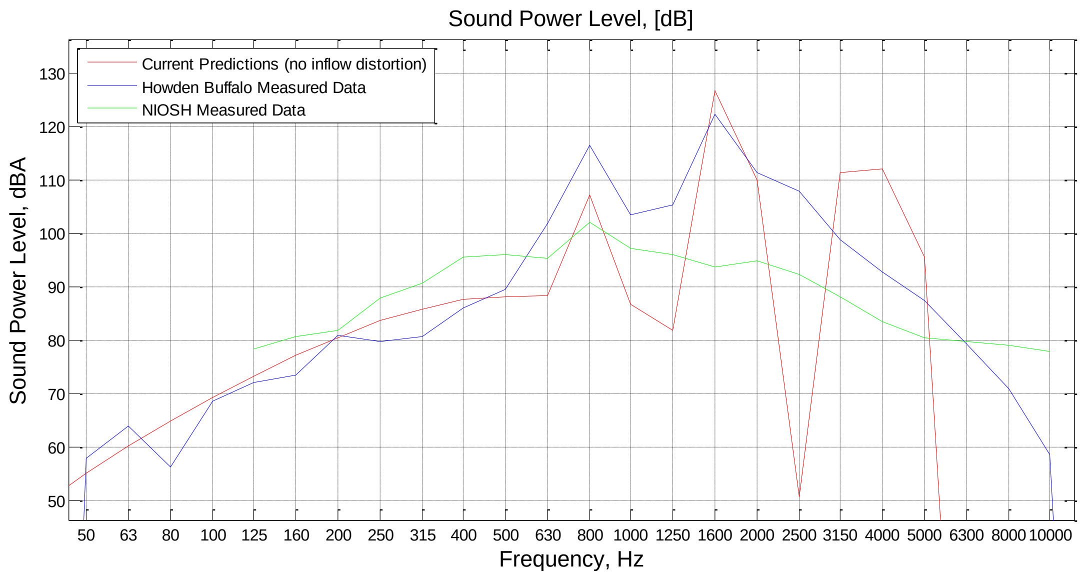


Figure 21: Measured and predicted third octave band sound power levels.

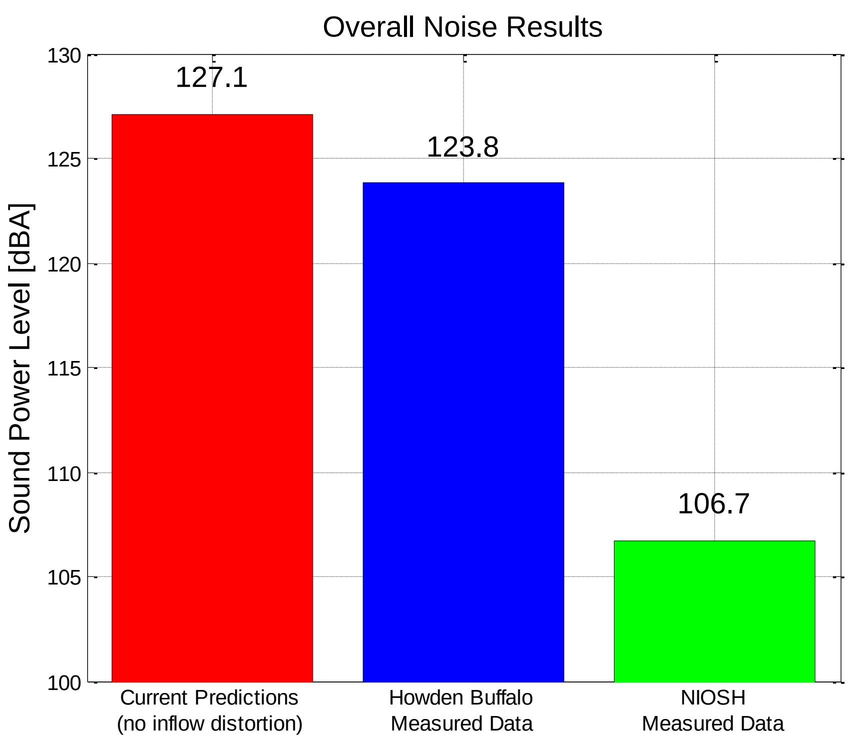


Figure 22: Measured and predicted overall sound power levels.

It is desired to use the predicted noise levels (sound power level) above to compare to the MSHA PEL (a dose of 100% or an 8 hour time weighted average of 90 dBA sound pressure level at the miner's ear). However, sound power levels do not directly correspond to the type of measurements used to estimate PEL dosage. Therefore, it can be assumed that the predicted baseline sound power levels above correspond to any baseline measurements of PEL dosage performed in the literature for CM operators. The measurements performed for a NIOSH report [3] are used for this purpose. Data from the measurements of 14 CM operators and helpers ranged from 44% to 347% MSHA PEL dose, where 100% is the maximum MSHA PEL. As an example, the reference's author included a representative cumulative dose plot with task observations for a single CM operator (8 hour shift). This plot is reproduced in Figure 23. It is assumed that the reference's author included this plot because it was the most representative of the CM operator's dosage history. Therefore, it was assumed that the average 8 hour dosage of a CM operator is approximately 300% of the MSHA PEL dose (5 dB exchange rate). This equates to a miner being exposed to an 8 hour TWA of 100 dBA (sound pressure level), or 10 dB above the MSHA PEL. This increases the miner's risk of NIHL. Therefore, if the noise level of the current/baseline CM machine is reduced by 10 dB, the CM operator would be able to operate the CM machine for the full 8 hour shift without increasing the risk of NIHL. This establishes a clear target for the noise reduction of the scrubber system, e.g. 10 dB or more.

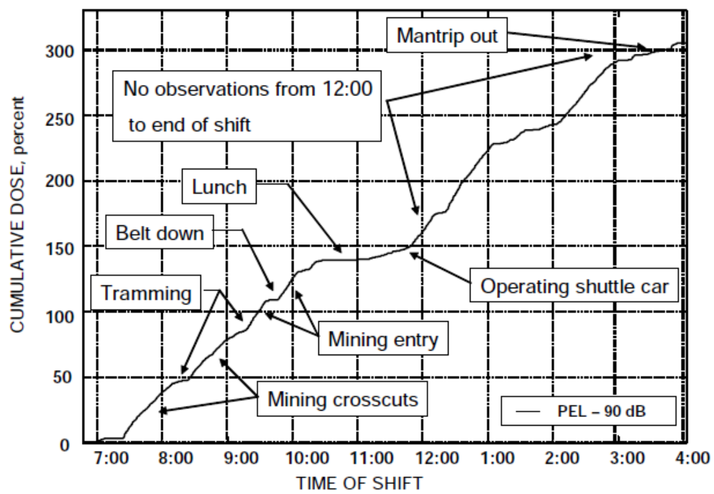


Figure 23: Cumulative dose plot for continuous mining machine operator. 100% dose corresponds to an 8 hour TWA of 90 dBA.

2.4. Baseline Fan Deficiencies

This section discusses the noise sources in the baseline scrubber fan. Based on the previous experience of AVEC engineers and rough calculations during this project, the following characteristics of the scrubber fan were determined to cause increased noise levels. The first three characteristics are considered the most important (bolded for effect).

The **rotor-stator spacing is very short**, i.e. about 1.7" axial spacing from the rotor trailing edge to the stator leading edge. Rotor-stator spacing is usually measured by the number of rotor chord lengths and the generally recommended spacing is more than 2 rotor chord lengths. In this case, 1.7" spacing corresponds to 0.29 rotor chord lengths which is significantly lower than the current "best practices".

The rotor and stator **blade count is not optimal**. Blade count is referred as the number of blades in the rotor and stator. Simple calculations have been performed by AVEC engineers and have determined that the 12/15 combination is not optimal. Simply selecting an appropriate rotor and stator blade count combination has the potential to significantly reduce tonal noise levels at the 1st and even 2nd BPF.

The rotor airfoil has a very **blunt trailing edge**. Literature [15] is widely available which proves that strong turbulent vortex shedding occurs for airfoils with thick trailing edges. This vortex shedding is a source of high frequency rotor-alone broadband noise. Also, the sharp trailing edge blade shape has the potential to reduce the rotor's velocity deficit, thus reducing rotor-stator interaction noise. The rotor airfoil trailing edge thickness can be seen in Figure 13 in section 2.1. The figure shows that the trailing edge thickness is about 50% of the airfoil thickness while "best engineering practices" result in a much sharper edge, typically 1% of chord or less.

The stator chord is unnecessarily long. The baseline scrubber fan features stator vanes with a 9.8" chord. For the given task of removing torque from the flow, this chord length is quite long. Also, reducing the stator chord length would allow for increased rotor stator spacing (see above).

The stator vanes were also found to be basically a cambered flat plate, i.e. not a traditional airfoil shape. Although the type of the airfoil used for the stators (flat plate vs airfoil) does not affect noise significantly, it is generally best practice to use a traditional airfoil shape. This will also reduce the required chord length.

The current acoustic "liner" is not a conventional design. In fact, this configuration is more consistent with an "acoustic blanket". As pictured (Figure 24), the current design likely allows noise to leak out instead of absorbing/reducing the acoustic energy.

The baseline scrubber fan housing has perforations for the acoustic "liner". The perforations are 0.25" diameter holes and approximately 0.25" deep (wall thickness). Perforations this large are likely to cause "grazing flow noise" which is an increase in noise in the high frequency due to flow turbulence as it passes over the perforated wall. These large perforations also increase the system drag, reducing efficiency.

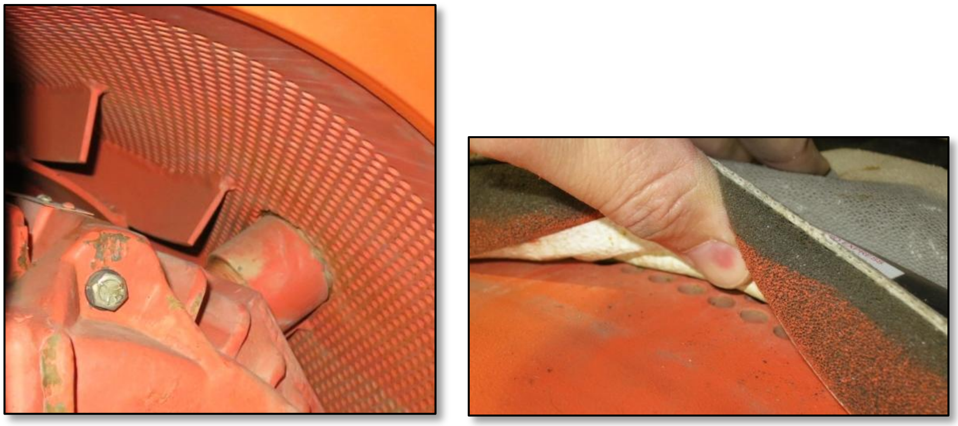


Figure 24: View of perforated housing downstream of the stator vanes (left) and foam material encasing the perforates (right).

The upstream hub spinner is very flat/blunt (Figure 25 left). This upstream flat hub introduces a large surface and relatively sharp turn for the flow. The sharp turn can cause flow non-uniformity and flow separation. Essentially, this hub is not streamlined.

The downstream hub geometry is not streamlined either (Figure 25 right). In fact, the downstream hub geometry is basically the electric motor housing. Although this is likely used for thermal management purposes, this design appears to be very non-streamlined. This is likely to cause separated flow and vorticity.

Figure 25: Upstream (left) and downstream (right) hub geometry.

The current ductwork and demister box causes significant flow distortions. These flow distortions include the demister box, blockages causing non-uniform flow, and the presence of upstream pipes. Testing by Howden Buffalo [17] reported that the flow leaving the demister box is 20° off axis. Additionally, large duct blockages were discovered at the junction between the scrubber fan housing and the demister exit. This junction is shown in Figure 26. At this junction, the demister exit is square shaped while the scrubber fan housing is circular. Further flow blockage and distortion is created by the presence of metal bars across the top and bottom of the duct leading to the demister inlet. These flow distortion causes are not addressed in the Phase I, but will be a key part of the design process in the Phase II project.

Figure 26: Spool piece inlet cross-section and resulting duct blockage (photo by Howden Buffalo Inc.).

In most of the cases listed above, the baseline fan deficiency is unnecessary. The additional noise could be greatly reduced with a proper design that exhibits minimal flow irregularities throughout the ductwork. The interfacing of the ductwork combined with the poorly designed demister exit is primarily responsible for the flow non-uniformity. Non-uniform flow and separation is the root of unfavorable aerodynamic and acoustic performance. Correcting the fan's inlet flow and optimizing the fan and hub geometry while implementing a proper acoustic liner will greatly reduce the overall noise.

3. Overview of Trade Studies Performed in the Project

A number of trade studies were performed to determine the effect of individual fan/system parameters on scrubber fan noise. The trade studies were performed in two main groups. The *preliminary* trade studies were performed to determine the effect of individual fan/system parameters on scrubber fan noise. The *advanced* trade studies were performed using a more advanced aerodynamic prediction tool and utilized the lessons learned from the preliminary studies. The trade studies that were performed were:

- Rotor and Stator blade count study
- Rotor-Stator spacing study
- Radial chord length distribution study
- Rotor tip speed study
- Hub radius study
- Converging/diverging duct
- Converging rotor area study
- Rotor-stator spacing study (refined)

These parameters were selected because they are fan properties that can be easily designed and implemented in currently operating continuous mining machines. Fan properties such as the outer diameter, total length, and RPM were not investigated as changes in these properties would require redesigning a portion of the CM. The current dimensions that the baseline fan occupies in the duct will remain constant such that the new design could be a retrofit (optionally). Keeping a constant RPM allows the currently installed motor to be used. These parameters will be analyzed in the Phase II project to determine their effect.

The conclusions and lessons learned from these trade studies are discussed briefly in this section. The complete review and discussion of the trade study results are not shown in this report for brevity.

3.1. Preliminary Trade Studies

This section discusses the preliminary trade studies performed to determine the effect of individual fan/system parameters on scrubber fan noise.

Rotor-Stator Blade Count Study

A preliminary analysis of the rotor and stator blade count was performed using a set of conventional tools. This preliminary study suggested that the optimal blade count pairs would be 11/17, 12/17, and 13/18 (rotor/stator blade counts), i.e. *not* the baseline 12/15 pair. These blade count pairs could potentially eliminate the noise from the 1st and also 2nd BPFs (approximately 720 Hz and 1440 Hz) which are both major contributors to overall noise for the baseline configuration. The 12/15 baseline configuration was found to produce noise at the 1st and 2nd BPFs (also observed in the measured spectrums) and was also found to potentially have higher noise at some of the higher harmonics as well.

Rotor-Stator Spacing

The rotor-stator spacing was analyzed using the fundamental analytical/numerical fan design tool (same tool used in rotor-stator blade count study). Recall that the rotor-stator spacing for the baseline is 1.7” or 0.29 rotor chords. This is considered very close when compared to the suggested best practices of 2 to 5 chord lengths. Fans with increased rotor-stator spacing were designed and analyzed. As expected, maximizing the rotor-stator spacing resulted in a decrease in rotor-stator interaction tones. For example, increasing the spacing from 1.7” (0.29 rotor chords) to 10” (1.77 rotor chords) decreased the noise level by over 11 dB. Spacing was not increased beyond 10” due to space limitations of the scrubber fan system. Since rotor-stator interaction tones are the overwhelmingly dominant noise source, the overall noise decreases significantly when this parameter is optimized.

Rotor and Stator Chord Length

The rotor and stator chords were varied to determine their effect on overall noise. Again, the fundamental analytical/numerical fan design tool was used. It was found that decreasing the rotor and the stator chords decreased the overall noise. Decreasing the stator chord significantly decreased the overall noise while decreasing the rotor chord had a smaller effect on the noise.

Rotor Tip Speed (RPM)

It was decided that due to constraints, such as the size and type of electric motor, the rotor tip speed would not be varied in this Phase I project. However, the effect of the rotor tip speed was briefly investigated for completeness. This preliminary tip speed study was not as conclusive as the previous trade studies. However, the general trend of overall noise decreasing with decreasing rotor tip speed was observed.

Lessons Learned from Preliminary Trade Studies

The most important lesson learned from these trade studies was that the baseline scrubber fan system does not have an appropriate blade count. Additionally, increasing the blade count enables a shorter chord to be used. The rotor and stator chord should be minimized to reduce noise without compromising the aerodynamic performance of the fan. The second most important lesson was that the rotor-stator spacing was extremely small and can be increased significantly within the geometric bounds of the current fan. It was found that the rotor-stator spacing was the most dominant parameter due to the fact that increasing the spacing produced the largest decrease in predicted overall noise.

3.2. Advanced Trade Studies

Based on these lessons learned from the preliminary trade studies, a collection of more advanced scrubber fan designs were produced and analyzed in a few additional precise trade studies. A new, more advanced tool was used to assist with the aerodynamic design of the scrubber fans. The advanced trade studies included varying the hub radius (constant/converging/diverging), and a more precise analysis of rotor-stator spacing. The results of these advanced trade studies are discussed here.

Hub Radius Study (constant axially)

Fan designs were generated for a hub radius 8.5” diameter (baseline) down to 7” diameter. Recall that the outer fan radius was held constant due to space constraints. This study found that decreasing the hub radius (increasing rotor annulus area) generally increased noise.

Converging/Diverging Duct

Fan designs were generated for several hub configurations that featured converging or diverging hub radii between the trailing edge of the rotor and the leading edge of the stator. The converging duct case, example shown in Figure 27 left, was hypothesized that less aggressive blades could be used to compress/accelerate the flow because of the converging duct. The diverging duct case, shown in Figure 27 right, was hypothesized that the additional flow diffusion would help with aerodynamic efficiency. The results of this study showed that neither the converging nor the diverging ducts reduced noise. In fact, the study showed that the noise was increased over the baseline for both of these configurations.

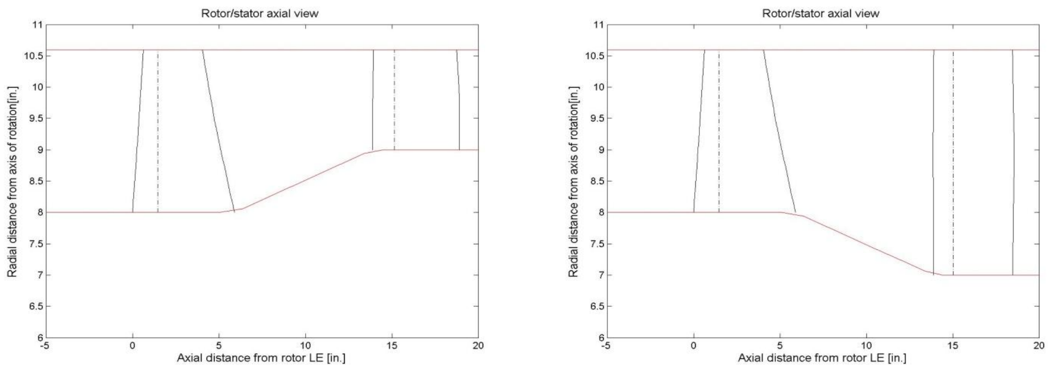


Figure 27: Meridional View of the 1” Converging and Diverging Ducts, respectively.

Converging Rotor Area Study

The rotor leading edge hub radius was decreased while keeping the rotor trailing edge hub radius constant. This created a converging annuls area across the rotor, see Figure 28. It was hypothesized that the reduced annulus area through the rotor would allow for increased aerodynamic efficiency and assist with compressing or accelerating the flow. The results showed that the converging rotor area did not decrease noise, but in some cases actually increased noise.

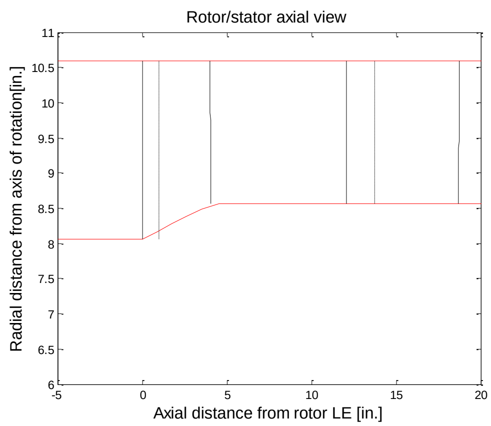


Figure 28: Meridional Views of Rotor LE Hub Radius Variance for 13/18 Fan.

Rotor-Stator Spacing Study

A simple spacing study was performed utilizing the more advanced aerodynamic fan design tool. Spacing was varied from 2” to 8” (trailing edge rotor to leading edge stator). This study was also repeated for multiple rotor-stator blade count pairs. The results of this study were definitive that increased spacing reduces noise. The largest decrease in noise due to the increased spacing was predicted to be nearly 25 dB for the 13/18 configuration. Such a large noise decrease may not be attained in actual circumstances, but if at least half of these reductions are achieved, then the miner would experience significantly lower noise levels and reduced NIHL.

The preceding trade studies effectively provided input and guidance with which to begin the design of the new advanced scrubber fan.

4. Evaluation of New Scrubber Fan Design

This section will present the new advanced scrubber fan that was designed based on lessons learned from the above trade studies. The first subsection discusses the acoustic results of the advanced scrubber fan design. The second section discusses the aerodynamic performance of the advanced scrubber fan. The performance of the advanced scrubber fan will be compared to the baseline.

4.1. Description of Advanced Scrubber Fan Design

Based on the preliminary and advanced trade studies discussed in section 3, the advanced scrubber fan design was created. A visualization of the advanced design is shown in Figure 29. Some of the major characteristics of the advanced scrubber fan are shown in Table 4. The characteristics of the baseline scrubber fan are also shown in this table for comparison. Based on the results of the trade study, the rotor and stator blade count was set to 13/18. The rotor-stator spacing was maximized to 8” such that the distance between the rotor leading edge and the stator trailing edge is approximately within the current geometric bounds of the baseline scrubber fan. The stator chord length was shortened to 7” and a traditional airfoil shape was implemented for the stator vanes. The rotor airfoil shape was changed to feature a sharper trailing edge (see Figure 30). The inlet hub geometry was adjusted to be much smoother. Also, predictions for this advanced scrubber fan assume that the acoustic liner used will have traditionally designed perforations so as to not create additional noise due to grazing flow.

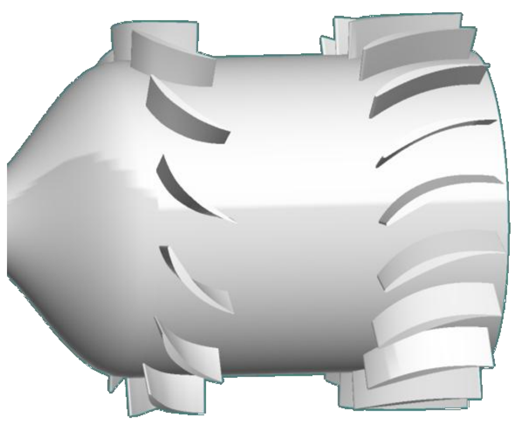
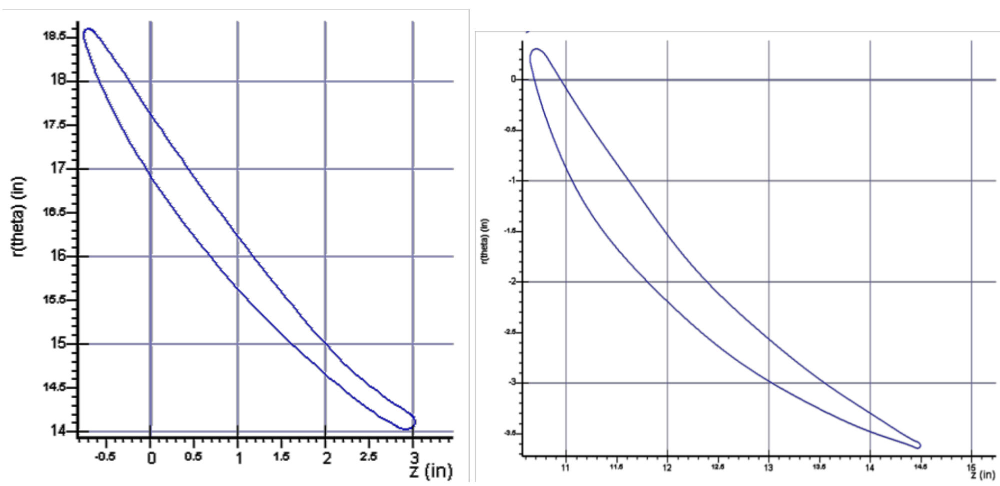


Figure 29: Advanced Scrubber Fan design.

Table 4: Primary characteristics of the advanced scrubber fan compared to the baseline.

	Baseline Design	**Advanced Scrubber Fan**
Blade count	12/15	13/18
Spacing	1.7”	8”
Stator Chord	10.5”	7”
Rotor Airfoil	Blunt TE	Sharper TE
Upstream flow distortions	Significant inflow distortions	Assumes minimal inflow distortions
Acoustic Liner/ Perforations	Unconventional	Tuned to peak frequencies (conventional)
Inlet hub geometry	Blunt/flat	Smoother



A) B)

Figure 30: A) Baseline rotor airfoil, and B) advanced scrubber fan rotor airfoil.

4.2. Acoustic Performance of Advanced Scrubber Fan Design

The acoustic performance of the advanced scrubber fan is discussed in this section. The acoustic analysis/prediction of the advanced scrubber fan follows the exact same procedures and methods that were used for the baseline fan noise analysis.

The overall noise from rotor-alone broadband, rotor-alone tonal, and rotor-stator interaction sources are summed. The overall A-weighted 1/3rd octave band noise spectra computed from the baseline fan CFD solution and the advanced scrubber fan CFD solution are presented in Figure 31 and Figure 32, respectively. The rotor-alone broadband noise clearly has a greater effect on the overall noise for the baseline fan. Additionally, the 1st and 2nd BPFs dominate the overall noise for the baseline fan while the advanced scrubber fan predicts that the first 2 BPFs do not radiate noise. Realistically there will be some amount of sound power at these cut-off BPFs, albeit the levels will likely be quite low. For both fans, the rotor-alone broadband noise dominates the low frequency range. The rotor-stator interaction produces the highest sound power levels for both fans. The baseline noise predictions (Figure 31) show that rotor-alone broadband and tonal noise does not contribute significantly to the overall noise. The baseline fan's rotor-stator interaction noise essentially accounts for 100% of the fan noise. For the advanced fan design (Figure 32), the rotor-alone broadband and tonal noise is insignificant to the overall fan noise.

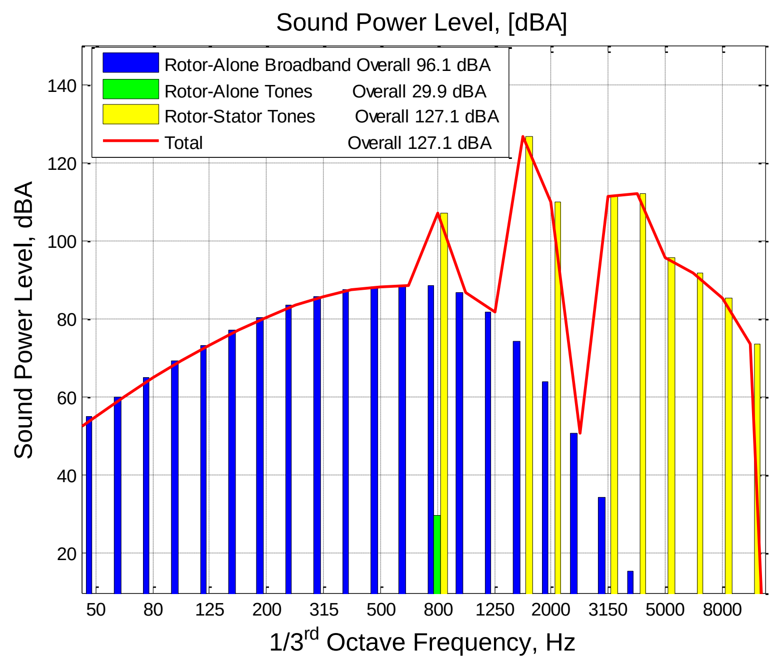


Figure 31: Overall A-weighted 1/3rd octave band noise spectrum for baseline fan.

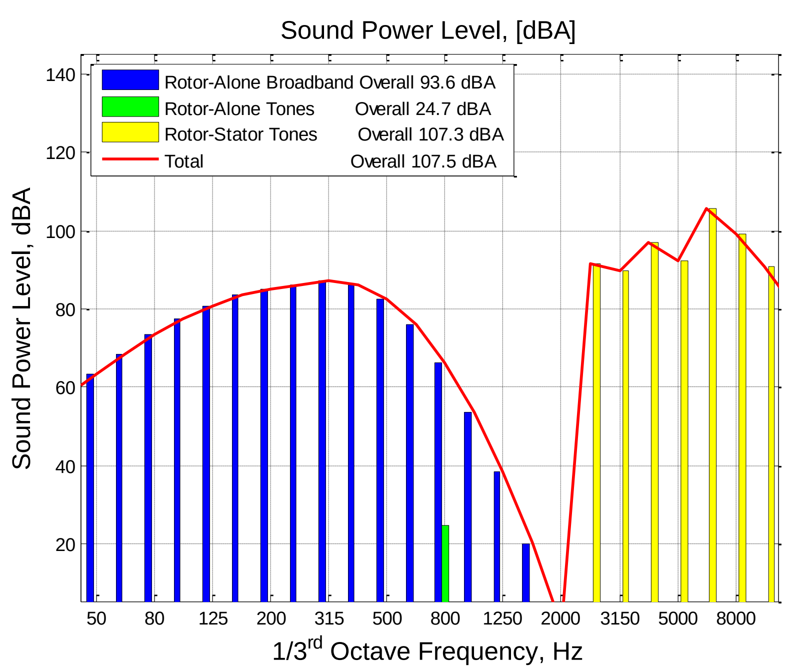


Figure 32: Overall A-weighted 1/3rd octave band noise spectrum for advanced scrubber fan.

The overall A-weighted noise for the baseline fan and the advanced scrubber fan with the relative contributing sound power levels from all considered sources are shown in Figure 33. The advanced scrubber fan is approximately 20 dB quieter than the baseline configuration. The rotor-alone broadband and rotor-alone tonal noise decreased slightly. The rotor-stator interaction noise greatly decreased and is the dominant source of noise.

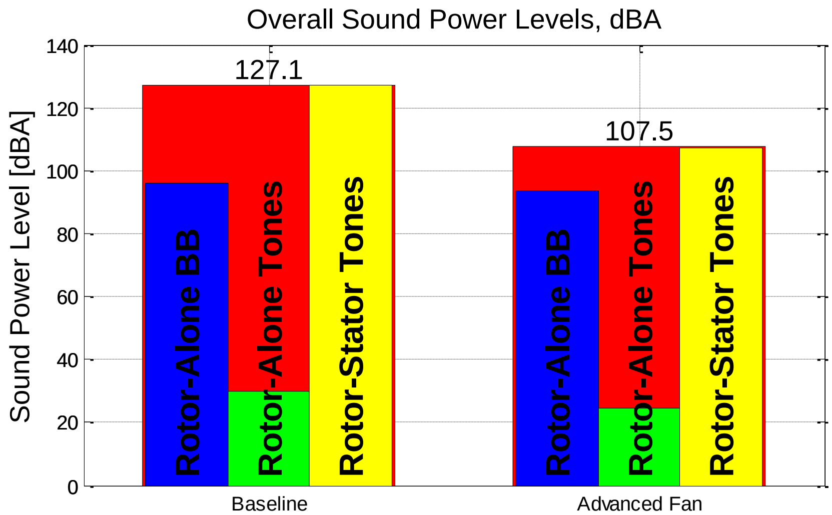


Figure 33: Overall noise for baseline and advanced scrubber fan.

The impact of the predicted 20 dB noise reduction on the miner is not discussed in this section. See section 6 for further discussion of the impact of the noise reduction on the miner.

4.3. Aerodynamic Performance of Advanced Scrubber Fan Design

This section discusses the aerodynamic performance of the advanced scrubber fan.

The aerodynamic performance of the advanced scrubber fan was analyzed using CFD. The CFD solution was calculated by Techsburg in ADPAC, similar to the baseline fan analysis. The flow being ingested by the rotor is assumed to be clean (well behaved) and straight. The solution includes the effect of the spinner and stators. The CFD solution at 25%, 50%, and 75% span are shown in Figure 34. The normalized axial velocity for a rotor blade and stator vane is shown in Figure 35. The pressure and streamlines for a rotor blade are shown in Figure 36. The reference pressure is atmospheric.

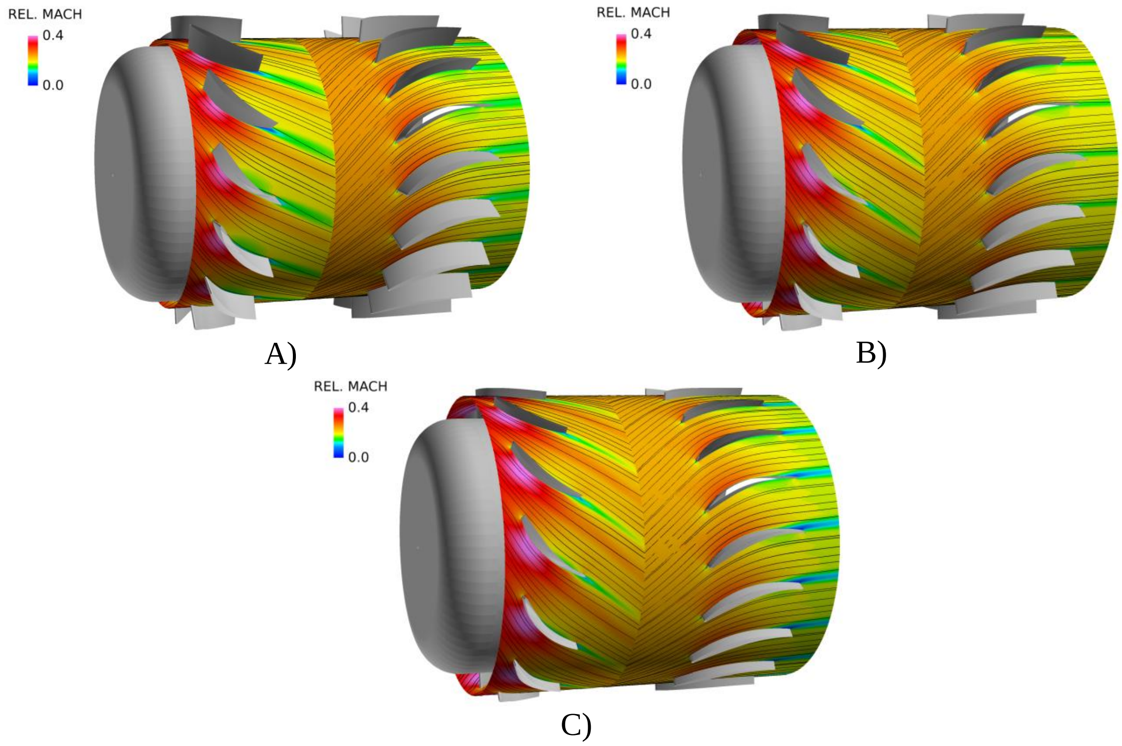


Figure 34: CFD solution of the advanced scrubber fan and stators at A) 25% span, B) 50% span, C) 75% span. All shown with flow streamlines.

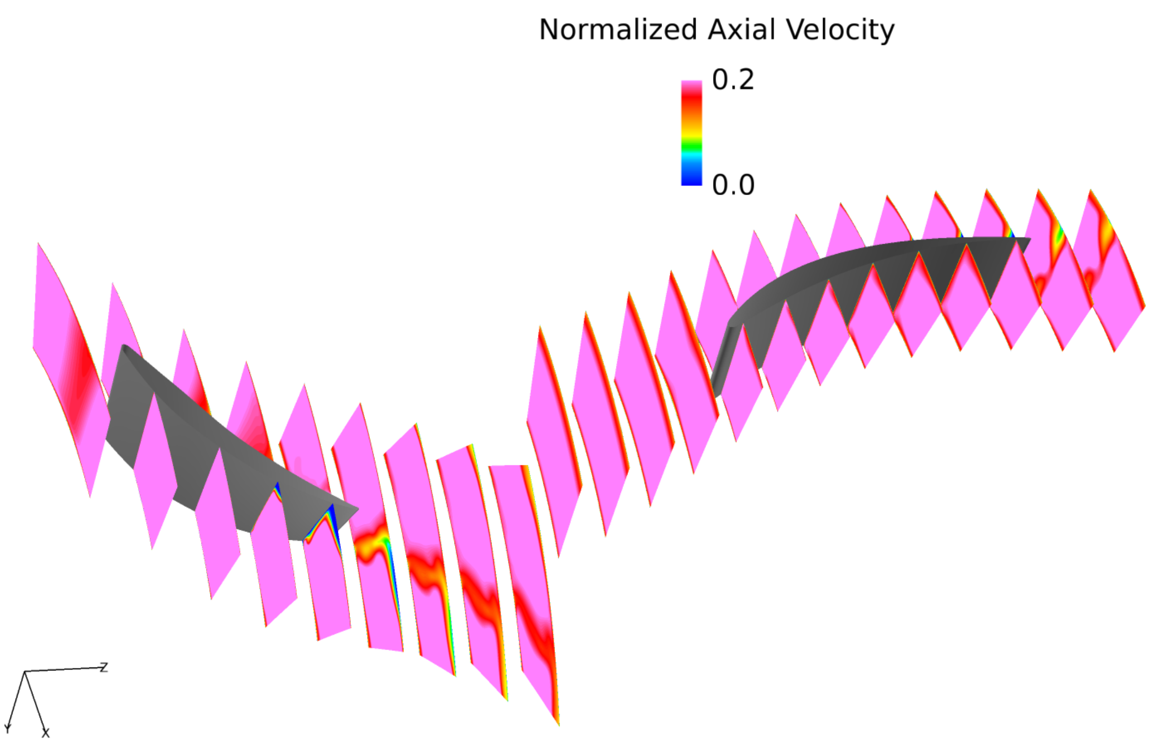


Figure 35: CFD axial velocity profile at multiple axial locations along the rotor and stator. Axial velocity is normalized to 943 ft/s.

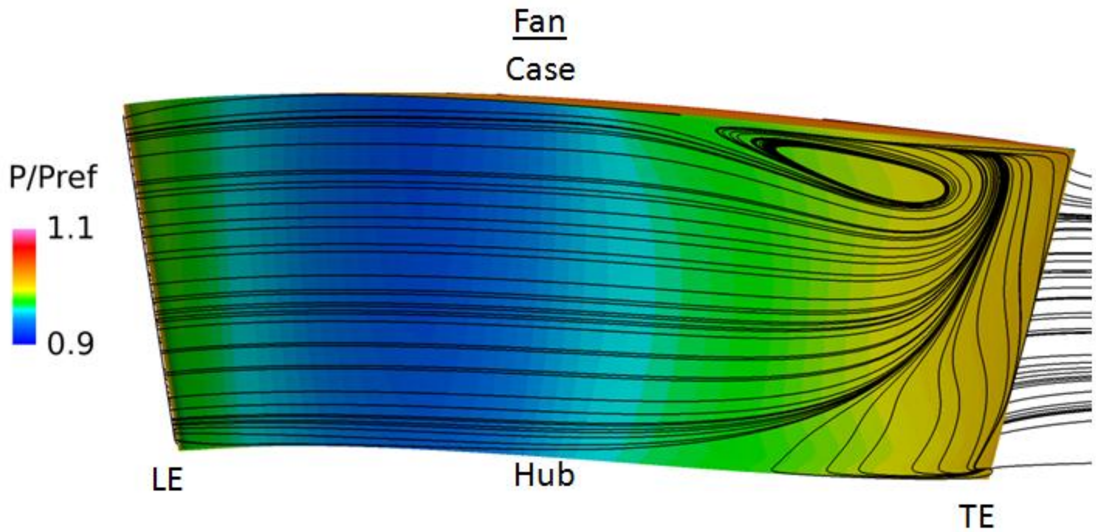
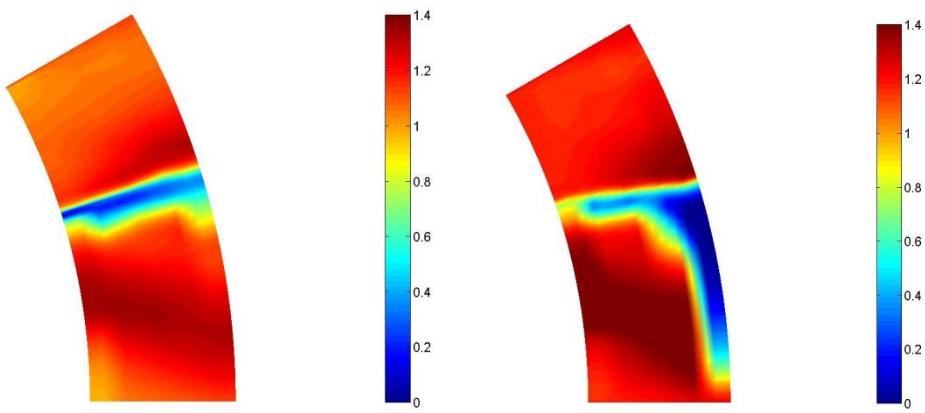


Figure 36: The pressure and streamlines on the suction surface of a rotor blade.

As was done with the baseline, the rotor wakes were extracted from the CFD solution at the rotor trailing edge (TE) for use in the noise prediction software. The rotor wake for the baseline fan and advanced fan are shown in Figure 37. The color scale is the axial velocity normalized by the freestream velocity (141.7 ft/s). Both plots were set to the same color scale. The main rotor wake deficit is observed as the blue region extending from the hub to the tip. The rotor wake deficit for the advanced fan is slightly smaller than the baseline wake deficit in this region, especially near the hub and mid-span. However, the wake for the advanced fan shows a significant wake in the tip region. There is likely separation occurring along the chord of the rotor blade near the tip. The results shown above in Figure 36 suggest this as well. The separation near the tip manifests as a significant wake observed as the long blue color region along the tip in Figure 37.



A) B)

Figure 37: CFD solution at the A) baseline rotor at the TE and the B) advanced rotor at the TE. The scale is Vaxial/Vinf where Vinf is 141 ft/s.

The performance curve containing the total pressure rise for the advanced scrubber fan and the baseline scrubber fan are shown in Figure 38. The new scrubber fan was designed with a higher head (total pressure rise) than the baseline to ensure that at least the same performance is achievable. This slight difference in pressure rise is likely within the margin of error for the aerodynamic performance prediction tools.

The efficiency of the advanced scrubber fan design is predicted to be approximately the same as the baseline fan's efficiency. This fan efficiency for the baseline and the advanced scrubber fan design is shown in Figure 39. Additionally, the power required for the fan (shaft power) is predicted to be higher for the advanced scrubber fan (Figure 40). It is believed that the higher power required is not a problem as long as the power required does not exceed the available power of the currently installed motor. The currently installed motor is rated for 30 Hp, which is sufficient for the maximum predicted power draw of the advanced fan in Figure 40.

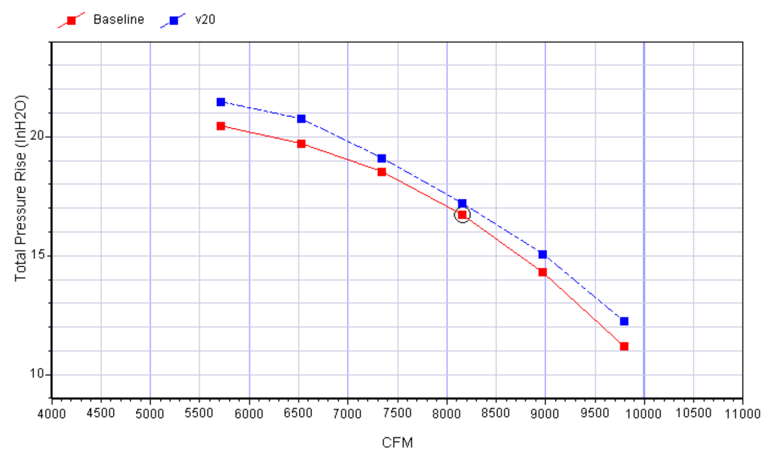


Figure 38: Fan performance curve, pressure rise and flow rate.

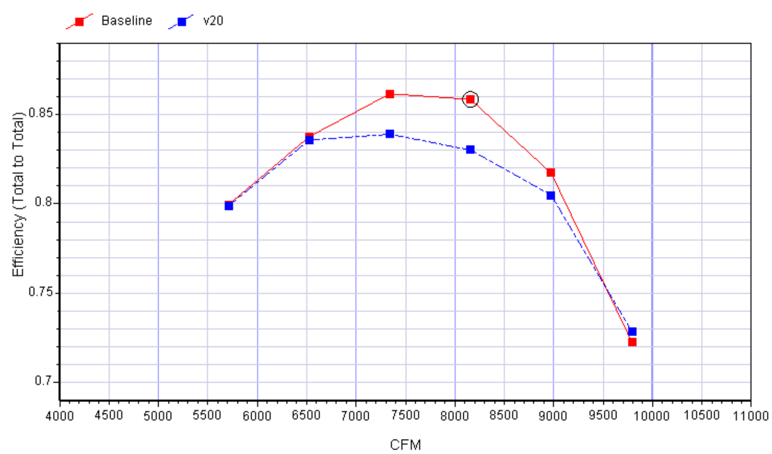


Figure 39: Scrubber fan design efficiency.

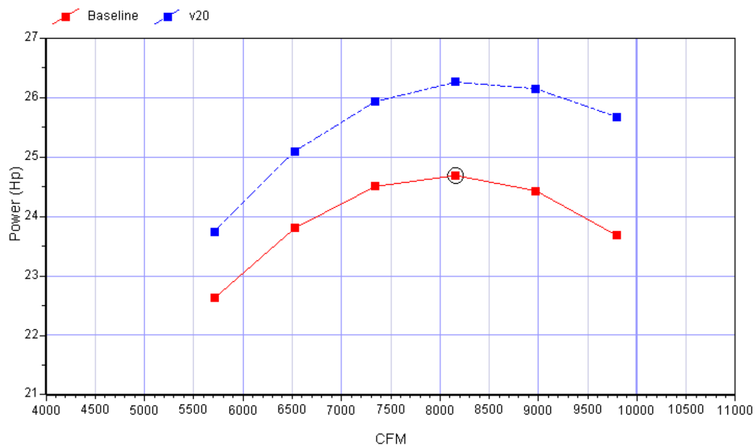


Figure 40: Power required from electric motor.

5. Additional Noise Control Options

This section discusses the research performed for additional noise control options for the scrubber fan, and also additional noise sources not specifically addressed elsewhere in the report. The first subsection discusses acoustic liners, liner types, and their proper implementation. The second subsection discusses the upstream flow distortion and how it affects noise produced from the scrubber fan. The third section discusses alternative noise control methods which could be analyzed and implemented in future designs. These additional noise control options reinforce the notion that the scrubber fan must be investigated and designed as a complete system.

5.1. Acoustic Liners

A deficiency in the current baseline scrubber fan is the absence of a proper acoustic liner. The current fan casing treatment includes perforations that have a diameter and depth of 0.25” with a roll of foam wrapped around the outside of the fan casing. This configuration is not traditionally considered an acoustic liner. One of the faults of this design is that the casing wall perforations allow noise to propagate through the wall and foam to radiate to the surrounding environment instead of propagating only in the fan duct/casing. Additionally, the large diameter of the perforations results in strong vortex shedding and turbulence which produces self noise. This is referred to in this report as “perforated plate” noise or “grazing” noise. Extensive industry research in grazing flow over perforated surfaces indicates that smaller diameter perforations with thinner walls are desired to be used with acoustic foam for effective noise attenuation. It is also a common practice to bond a very thing wire mesh to smooth out the flow over the perforation and minimize the vortex shedding.

When considering traditional/conventional acoustic liners, there are two basic types of liners: locally reacting and bulk reacting. A locally reacting liner features a local acoustic impedance value that is independent of whatever occurs at any other part of the liner.

Sound propagation does not occur in any other direction than normal to the surface. Locally reacting liners are typically comprised of a solidly backed honeycomb structure with a perforated face sheet. The honeycomb structure forms an array of Helmholtz resonators that can be tuned to attenuate sound in certain frequency ranges by simply varying the depth of the honeycomb cavity. The primary difference between a bulk reacting liner and a locally reacting liner is that sound is able to propagate in all directions through the bulk liner. Bulk liners typically consist of a wire mesh face sheet over a cavity filled with an acoustically absorptive material that has a solid backing. They are somewhat better to attenuate broadband noise.

For the implementation of the advanced scrubber fan on the CM, it is suggested to use a bulk reacting liner consisting of a wire mesh face sheet containing an air cavity. It is recommended that the face sheet material be the only absorptive component (for example fiber metal sheet) and leave the liner cavity empty due to the expected dust and water particles in the flow downstream of the demister. If absorptive material is placed in the liner cavity it would likely become clogged and useless over time.

The amount of noise reduction (attenuation) from the recommended bulk liner has been estimated as part of this study using the methods described in reference 16. The acoustic liner would primarily be used to attenuate the broadband noise; this broadband noise has a maximum sound power level around 800 to 1000 Hz. Other frequencies will be analyzed to determine attenuation at a range of other frequencies ranging from the first to second BPF. Attenuation rate calculations were performed for a 1” thick (air cavity) acoustic liner with flow resistivity of 1ρc and 2ρc. These results are presented in Table 5 and Table 6. An estimate of the noise reduction assuming a liner length of 20” is also presented.

Table 5: Estimated attenuation for 1” thick 1ρc liner.

Frequency (Hz)	Attenuation (dB/inch of duct length)	Attenuation for 20” of treatment (dB)
800	0.24	4.8
1000	0.36	7.2
1250	0.4	8.0
1600	0.6	12.0

Table 6: Estimated attenuation for 1” thick 2ρc liner.

Frequency (Hz)	Attenuation (dB/inch of duct length)	Attenuation for 20” of treatment (dB)
800	0.32	6.4
1000	0.48	9.6
1250	0.58	11.6
1600	0.76	15.2

5.2. *Effect of Upstream Flow Distortion*

This section discusses the noise source caused by upstream flow distortion. This section also discusses the noise reduction potential from reducing or eliminating the flow distortions.

For the current scrubber fan system configuration, upstream flow distortions caused by system elements have been found to be a significant noise source. These flow disturbances are caused by required components in the flow, such as the demister, and support struts. The demister is used to remove water droplets and solid particles from the air and the support struts are used to hold up the duct walls. As was discussed in the previous work section 1.3, the Howden Buffalo test in 2008 discovered that the demister created a significant amount of flow non-uniformity [17]. The demister box blocks the flow in the duct at the bottom and top (Figure 10). The demister box also exhausts the flow at a 20° angle from the axial centerline. The shape of the duct also contributes to the flow distortion. The ductwork upstream of the demister box is rectangular while the ductwork downstream of the demister box is circular. There is also an aggressive diverging duct area directly upstream of the demister box [18]. All of these discontinuous duct shapes cause flow distortions.

The flow non-uniformities enter the fan and generate noise. Although the noise is generated at the fan, the actual design of the fan has little effect on the noise produced. As long as the fan is spinning, this “rotor-interaction” noise source will exist. The only way to reduce this noise is to decrease the amplitude of the flow distortion or increase the distance between the flow distortion and the fan to allow for the distortion to dissipate. For the CM scrubber fan system, space limited. Therefore, increased distortion-fan spacing is not a viable solution. Eliminating the sources of the flow distortion would be the better solution by far.

As mentioned previously, this flow distortion source was studied previously by Carter [8] and Howden Buffalo [9]. Both studies concluded that flow distortion is a major factor in the increase of system sound. Howden Buffalo’s measurements found that at least a 3 dB noise reduction appears to be possible by simply straightening the flow leaving the scrubber/ demister and eliminating the duct blockage [18].

It is important to realize that the CM components that cause flow distortion are weaknesses of the baseline system (discussed in section 2.4). In order to eliminate or greatly reduce the flow distortion ingested by the rotor, the scrubber system must be redesigned and analyzed as a complete system and not as single components. This higher order design strategy will be implemented in Phase II.

5.3. *Other Noise Control Options*

Other noise control options can be implemented in future advanced scrubber fan designs. Examples of these potential future noise control options are rotor trailing edge serrations, stator vane lean and sweep, rotor modulation, and potentially others. These noise control options are not recommended to be implemented at this time since their benefit would be small, especially in comparison to the flow distortion noise source and the potential reduction from the acoustic liner.

Serrated trailing edges are a modification to the trailing edge designed to reduce self-noise generation. Significant literature proving the noise reduction capability of serrations are available [19,20,21]. The serrations appear as a sawtooth pattern. The serrated trailing edges for the scrubber fan system would be implemented on the rotor blades trailing edge.

A modulated rotor features variable angular spacing between blades. Note that rotor modulation is currently implemented on the baseline fan. However, this modulation was found by Carter to increase noise levels. Note that rotor blade modulation should reduce noise, so additional studies should be performed to determine if this technology can help reduce noise if properly implemented.

6. Impact on Miner

Recall from the introduction section that the scrubber fan is one of the loudest noise sources on the CM. In fact, in a test performed by NIOSH [3], where CM operators and CM helpers wore a noise dosage meter, it was shown that the MSHA Permissible Exposure Level (PEL) was exceeded by 86% of the monitored CM operators. Restated, 86% of the CM operators/helpers tested exceeded the 100% dose. Recall that the (PEL) is defined as an 8-hour time-weighted average (TWA_8) sound level of 90 dBA, or 100% dose. MSHA uses a 5 dB exchange rate. In mines where CM machines were used, the CM operator/helper was the occupation with the highest noise exposure/dose. Data from the measurement of these CM operators/helpers ranged from 44% to 347% PEL dose. In section 2.3, it was assumed based on NIOSH data [3] that the average CM operator was exposed to approximately 300% dose of the MSHA PEL (see Figure 23) in an 8 hour work shift. To eliminate the increased risk of NIHL, the noise from the CM would have to be reduced by about 10 dB.

It was predicted that the advanced scrubber fan design would reduce the noise from the CM scrubber fan by 20 dB (section 4.2). However, this prediction does not include the effect of inflow distortion. And also does not include noise from other CM components, such as the chain conveyor system. As such, the actual amount of CM noise reduction will be less than 20 dB. In fact, the inflow distortion noise source was measured to be approximately the same level as the baseline fan. Thus, if the inflow distortion noise source is not addressed, the actual expected noise reduction would only be about 3 dB. Although the scrubber fan inflow distortion noise source is related to the scrubber fan, the method of noise control for this noise source is to address the demister box design, i.e. redesign in Phase II (see section 5.2). Additional reduction of the inflow distortion noise source can also be obtained by implementation of an acoustic liner (see section 5.1). If the inflow distortion source is reduced, there is significant potential to meet the MSHA PEL requirement, i.e. additional 10 dB noise reduction.

In summary, the noise caused by the scrubber fan itself has been significantly reduced (i.e. 20 dB). However, other noise sources, such as the scrubber fan inflow distortion noise, have become the dominant noise source. These other sources will be addressed in the Phase II.

7. Summary/Future Work

The current baseline scrubber fan systems on CM machines are not designed with low noise as a primary goal. The high noise levels produced by these machines lead to noise induced hearing loss in mining personnel [1,5]. A redesign of the CM scrubber fan system using low noise as the primary design goal reduces the likelihood of hearing loss in mining personnel and increases workplace safety. Deficiencies of the current baseline scrubber fan design were identified. To summarize, the baseline fan design was found to be very acoustically inefficient. Various trade studies were performed to determine the effect of various fan parameters on fan aerodynamic and acoustic performance. Trade studies included studying the effect of rotor and stator blade count, rotor-stator spacing, hub radius, and others. Based on these trade studies, a 13 blade rotor and 18 vane stator design-pair was created and iterated on to converge on the advanced scrubber fan design. This advanced fan design is predicted to be about 20 dB quieter than the current baseline scrubber fan. However, flow distortion produced by the poorly designed ductwork and demister which is then ingested by the fan was also found to be a significant source of noise in the scrubber fan system. This inflow distortion and other noise sources will become dominant if the scrubber fan noise source is reduced, thus reducing the effectiveness of the new advanced scrubber fan. The other, new and dominant, noise sources, such as the inflow distortion, will be investigated in the Phase II project.

8. Acknowledgements

Special thanks are given to Joy Mining and their cooperation in supplying the baseline scrubber geometry and relevant system parameters, and also providing a physical sample of the baseline for analysis. Special thanks are also given to Howden Buffalo for their cooperation in supplying the baseline scrubber system.

AVEC is grateful to CDC/NIOSH for the opportunity to research this important area of reducing the risk of hearing loss in miners.

9. Publications

No publications were produced as a result of this work during this project period.

10. References

1. Camargo H.E., Ravetta P.A., Burdisso R.A., and Smith A.K., "A Comparison of Beamforming Processing Techniques for Low Frequency Noise Source Identification in Mining Equipment", Proceedings of the International Mechanical Engineering Congress and Exposition, IMECE 2009, paper IMECE2009-12194, Lake Buena Vista, Florida, November 13-19, 2009.
2. MSHA [2008]. MSHA Standardized Information System, Number of Noise Samples @90PELDose(computed)>=100 by Machine Code, 2006. Denver, CO: U.S. Department of Labor, Mine Safety and Health Administration, Office of Injury and Employment Information. Date accessed: March 20, 2008.
3. Bauer, E.R., Babich, D.R., and Vipperman, J.S. "Equipment Noise and Worker Exposure in the Coal Mining Industry," Pittsburgh, PA: U.S. Department of Health and Human Services, Public Health Service, Centers for Disease Control and Prevention, National Institute for Occupational Safety and Health, DHHS (NIOSH) Publication No. 2007-105, IC 9492. 2006.
4. Kovalchik, P.G., Matetic, R..J., Smith, A.K., Bealko, S.B. "Application of Prevention through Design for Hearing Loss in the Mining Industry." National Institute for Occupational Safety and Health. 2008.
5. Babich, DR and ER Bauer. "Summary of longwall and continuous miner section noise studies in underground coal mines." Mining Engineering, v. 58, No. 11, Nov., pp. 41-46. 2006.
6. Camargo H.E., Ravetta P.A., Burdisso R.A., and Smith A.K., "Application of Phased Array Technology for Identification of Low Frequency Noise Sources," Journal of Low Frequency Noise, Vibration and Active Control, Vol. 28, No. 4, pp. 237- 244, 2009.
7. Ravetta P.A., Macko N., and Burdisso R.A., "Phased Array Measurements of a Continuous Miner," Final Report to NIOSH PRL, June 3rd 2008.
8. Carter, B.A. "Noise reduction of a vaneaxial fan for a continuous mining machine," Master of Science, The Pennsylvania State University, 2003.
9. Howden Buffalo Inc., "Reduction of Noise Emissions from a Vane-Axial Fan in a Duct Collector in a Continuous Miner," Test Results for Howden Buffalo, Inc. Project Number 990978, September 2008.
10. Woodward, R.P., Elliot, D.M., Hughes, C.E., and Berton, J.J., "Benefits of Swept-and-Leaned Stators for Fan Noise Reduction," *Journal of Aircraft*, Vol. 38, No. 6, Nov-Dec 2001.
11. Dittmar, J.H., Elliott, D.M., and Fite, E.B., "The Noise of a Forward Swept Fan," NASA/TM-2003-212208, November 2003.
12. Nakahama, T., and Ishibashi, F., "Effects of Fan Blade Forward-Swept and Inclined Amounts in Electric Motors," *IEEE Transactions on Energy Conversion*, Vol. 25 n.2, pp. 457-464.
13. Sharland, I.J., "Sources of Noise in Axial Flow Fans," *J. Sound Vib.*, Vol. I , no. 3, 1964, pp. 302-322.

14. Hanson, D. B. and Parzych, D. J., “Theory for noise of propellers in angular inflow with parametric studies and experimental verification,” *NASA CR-4499*, Mar. 1993.
15. Brooks, T., Pope, D., and Marcolini, M., "Airfoil Self-Noise and Prediction," NASA Reference Publication 1218, National Aeronautics and Space Administration, 1989.
16. Bies, D.A. and Hansen C.H. “Engineering Noise Control: Theory and Practice, 3rd Edition,” Spon Press, 2003.
17. Susey, R., “Report on Miner Fans Tested per TP-00005,” Howden Buffalo Internal Report, September 2008.
18. Susey, R., “Suggested Improvements for Miner Fans Tested per TP-00005,” Howden Buffalo Internal Report, September 2008.
19. Geiger, D.H., “Comparative Analysis of Serrated Trailing Edge Designs on Idealized Aircraft Engine Fan Blades for Noise Reduction,” M.S. Thesis, Submitted to Dept. of Aerospace and Ocean Engineering, Virginia Tech, 2004.
20. Gliebe, P.R., “Serrated Fan Blade”, US Patent 6,733,240, filed 07/18/2001, issued 05/11/2004.
21. Jones, L.E., and Sandberg, R.D., “Numerical Investigation of Airfoil Self-Noise Reduction by Addition of Trailing-Edge Serrations,” AIAA-2010-3703, 16th AIAA/CEAS Aeroacoustics Conference, 7-9 June 2010, Stockholm, Sweden.
22. Gliebe, P. R., “Fan Broadband Self Noise Prediction Model,” AIAA 2002-2490 8th AIAA/CEAS Aeroacoustics Conference & Exhibit, 17-19 June 2002, Breckenridge, Colorado.

11. Appendix A

Email from Industrial Hygiene Specialist at OCI (mining company).

-------- Original Message --------
Subject: Progress on Continuous Miner Fan Engineering
Date: Wed, 21 Aug 2013 14:18:47 +0000
From: Braithwaite, Scott [redacted]
To:[redacted]@avec-engineering.com [redacted]@avec-engineering.com>

Patricio,

I am an industrial hygienist evaluating my location's continuous miners. As is commonly known, the ventilation fans on these machines are a major source of noise. I came across your contact information on the sbir.gov website while researching noise reduction for the fans. Would you be able to give me any guidance on what your research has developed? Are there any major equipment manufacturers that you are working with that could also offer information on improved noise reduction?

Thank you for your time.

Scott Braithwaite
Industrial Hygiene Specialist

OCI

OCI Wyoming, L.P.
[redacted] **office**
[redacted] **fax**
[redacted]

12. Appendix B

Letter of Support from Joy Mining.

JOYGLOBAL

Jim Krellner
Manager, Design Team R&P
Underground Mining

November 20, 2013

Mr. Kyle Schwartz
Project Principal Investigator
AVEC, Inc.
3154 State Street, Suite 2230
Blacksburg, VA 24060

Re: SBIR Proposal "Quieting of the Continuous Miner Scrubber Fan System Noise"

Joy Global Underground Mining LLC f/k/a Joy Technologies LLC d/b/a Joy Mining Machinery is the global leader in the development, manufacture, distribution and service of underground mining machinery. As such, it is continuously engaged in technology development to improve efficiency and reliability, and to minimize the noise exposure to mine workers. Applied research and development is needed to use the latest technologies from other engineering fields to advance these goals. A key piece of equipment is the continuous miner. A dominant noise source in our continuous miner is the scrubber fan system (SFS). Reducing the noise emission from this system will have an immediate beneficial effect on mine workers.

We were very pleased with the results of the Phase I project where AVEC achieved approximately 20 dB of noise reduction while maintaining the aerodynamic performance required for dust collection. The proposed plan for Phase II includes the design and manufacture of a complete prototype SFS using the latest technologies from the commercial and military aeronautic fields where low noise emissions are of critical importance.

There are several aspects of the Phase II proposal that we find particularly appealing:

- Realistic Approach: AVEC and Techsburg have a realistic and solid approach to addressing this challenging problem. The AVEC/Techsburg team will redesign the key components in the SFS such as the demister and ductwork, and will refine the fan-stator design. AVEC will also implement a more traditional and realistic liner/silencer suitable to the environment.
- Approach Includes Installing a Prototype SFS on a Complete CM: AVEC's plan to manufacture a full size prototype SFS for testing is impressive and will demonstrate the technology with high confidence. This significantly reduces the technology's risk outlook. AVEC also proposes an installation of the SFS prototype on a complete continuous miner. We view this as the pinnacle of validation testing, maximizing the chances for a successful outcome (which would be to add a commercial version of the SFS to our new continuous miners).
- Experience and Expertise: The team has the experience and expertise in fan/propeller aerodynamics (Techsburg) and acoustics (AVEC), two critical fields for a successful outcome.

This project is very important to our long term company goals. If funded, Joy will support this work by providing technical drawings/CAD files of the relevant SFS parts (demister, ductwork, motor etc.). Joy will also provide a complete SFS to be used in lab testing as a baseline reference. Joy will also commit to installing the new prototype system on a continuous miner at

120 Liberty Street, Franklin, PA, 16323
+1 814 432 1487 Phone | jim.krellner@joyglobal.com

November 20, 2013
Page | 2

a Joy location and to performing check-out/validation testing of relevant parameters, e.g., flow rate and pressure rise. AVEC and Joy already have an agreement in place giving Joy a first option to license the technology from AVEC. If Phase II is successful, Joy also expects that it will continue to pursue with AVEC, additional scrubber fan system designs for similar continuous miners. We strongly support the work proposed going forward.

If there are questions or I can be of further assistance, please do not hesitate to contact me.

Best Regards,

JOY GLOBAL UNDERGROUND MINING LLC

By: ____________________
Name Printed: JIM KRELLNER
Title: ENGINEERING MANAGER