SILENCING HIGH VELOCITY HVAC DUCTS IN OCEAN GOING FAST FERRIES

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Abstract

Preliminary acoustic testing of conical diffusers has been completed for use in a proposed high velocity HVAC duct system for large ocean going fast ferries. The flow-induced noise from such a system was identified as a critical limit on the maximum air distribution velocity. The design of the diffuser outlet was found to have a pronounced effect on the noise spectrum radiating from the duct outlet. Upstream system noise can be treated with inline silencers and controlling the expansion of the exiting air can effectively treat the flow-induced noise created. The noise level from a straight jet of air exiting a 50 mm diameter pipe was measured and compared with the same pipe fitted with a 7° diffuser with an outlet diameter of 150 mm. The length of pipe fitted to the end of the diffuser was varied to identify the optimum outlet design. The addition of a 90° elbow and a 135° Y outlet duct configuration were also investigated. Noise levels have been recorded based on a mean duct velocity ranging from 15 to 60 ms^{-1}. Corresponding numerical (CFD) models have also been used to predict noise levels for new diffuser designs.

Introduction

Ocean going fast ferries are designed to operate across most marine climates around the world, which place severe demands on the vessel air conditioning system. The available space for air conditioning ducts in these vessels is also very limited, often leading to substantial installation costs and delays. The use of a compact duct network with air velocities increased from the standard 5-7 ms^{-1} to 20-30 ms^{-1} enables the physical size and weight of the ducting used to be significantly reduced.

The increased power requirements to move the conditioned air through the high-speed duct network places an added demand on the electrical generators used to provide auxiliary power on board the vessel. However, given the current 10-15% power loading there is scope to increase this power requirement by balancing the increased operational cost against the potential weight savings from lower deck to deck heights and reduced weight in the superstructure required to reinforce large holes around standard HVAC ducts. Significant installation cost savings can also be realized with the installation of smaller ducts.

Current noise standards for ocean going fast ferries require a passenger cabin noise level of 65 dB (A) or less. The HVAC systems are typically designed based on ASHRAE design standards [1] using normal building services grade components. Given that these standards are written to meet a noise rating of RC25-RC45 [2] the systems used onboard the fast ferries are well over designed from a noise criteria perspective. With this in mind the actual velocity of the HVAC air entering the cabin space must still be limited to meet minimum comfort levels and prevent localized cold/hot spots.

A hybrid HVAC system using high velocity (smaller diameter) air distribution ducts and standard outlet diffusers is proposed to satisfy the above criteria. The primary challenge of implementing such a system rests with limiting the propagation of flow generated noise into the passenger cabin and the ability to slow down the high velocity flow prior to entering the passenger cabin. Experimental tests were conducted to assess the success of controlling the flow generated noise at the duct outlet. A simple model using a straight circular jet fitted with a 7° conical diffuser was used to both validate this hypothesis and provide a benchmark to validate a numerical model to be used as a future design tool.

Experimental Investigation

Actual sound pressure level measurements have been recorded across a range of jet velocities for a simple circular jet 50 mm in diameter fitted with a 7° outlet diffuser. The measurements were taken in the dual room reverberation suite at the School of Mechanical & Manufacturing Engineering at UNSW. The layout of the two rooms is shown in figure 1. A large centrifugal high flow oil-free (300 l/s) air compressor was used to supply a regulated flow of high pressure air to 3 storage tanks with a combined volume of 6 m^3 at 800 kPa. An electronic motor controller was fitted to the compressor to maintain a constant pressure in the storage tanks and maintain a steady flow into the system.

The airflow into the reverberation suite was adjusted by way of a main throttling valve and a secondary flow control valve. The supply air passed through a two-stage muffler (A) and small plenum box (B) before entering the first (source) reverberation room that had a total volume of 187 m^3. All pipe work was lagged with high density lead impregnated material to minimize the radiation of noise into the reverberation suite. The flow control valves were also located externally to the reverberation room to maximize the upstream distance from the reverberation room. The source room was sealed such that it served as a secondary pressurized plenum chamber. The only outlet from the source room was through a bell mouth fitted to the main air jet test section located in the adjoining reverberation (receiver) room.

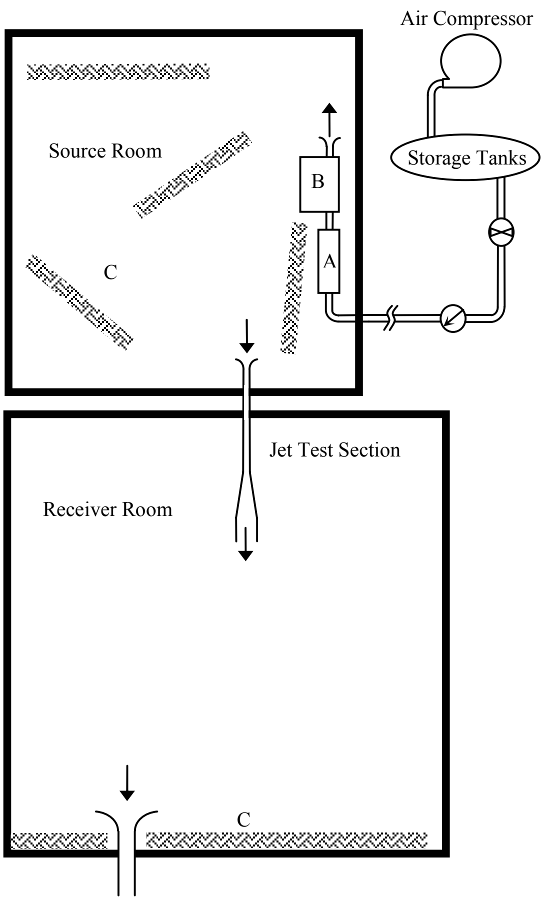


Figure 1. Reverberation suite configuration floor plan

The ambient noise level in the source room was lowered further with the use of acoustic lagging material (C) placed irregularly to maximize their effectiveness. The same material was used to improve the low frequency response [3] of the receiver room used for the flow induced noise measurements. An inline composite reactive/dissipative silencer was fitted upstream of the outlet test section to minimize the level of transmitted noise from the source room. The flanking noise levels were also monitored and maintained to ensure accurate noise level measurements were possible.

Experimental Procedure

All noise readings were taken using Bruel & Kjaer type 4189 (1/2 inch) Falcon microphones connected to Bruel & Kjaer type 2699 microphone preamplifiers. The raw acoustic signals were analyzed with a Bruel & Kjaer Type 2133 Dual Channel Digital Frequency Analyzer. Spatially averaged sound pressure levels were measured with the use of Bruel & Kjaer type 3923 Rotating Microphone Booms. To improve the accuracy of the measurements the boom was moved to three asymmetric locations around the outlet.

Noise readings were taken across a velocity range of 25-60 ms^{-1} based on the time averaged centerline jet velocity for the 50 mm jet with and without the 7° diffuser. Additional tests were also conducted with a variety of outlet duct configurations fitted down stream from the diffuser. After careful calibration of the microphones and data acquisition equipment ambient and flanking noise levels were recorded for future reference. All noise measurements taken were at least 20 dB (re $2e^{-5}$ Pa) higher than the ambient and flanking noise measurements taken.

50 mm Round Jet (no diffuser)

A straight round jet was used as the benchmark for comparison against each outlet configuration. The sound pressure level in 1/3 octave bands was recorded for a 50 mm diameter jet velocity of 35 – 60 ms^{-1} and is shown (as octave bands) in figure 2.

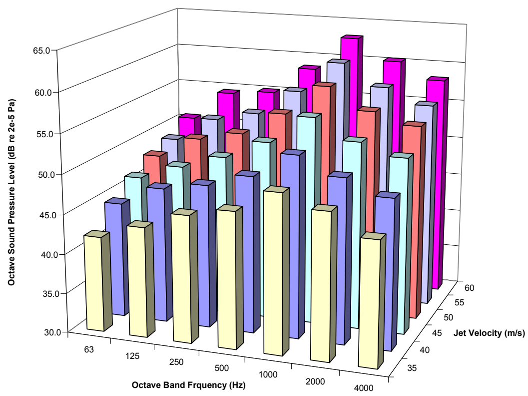
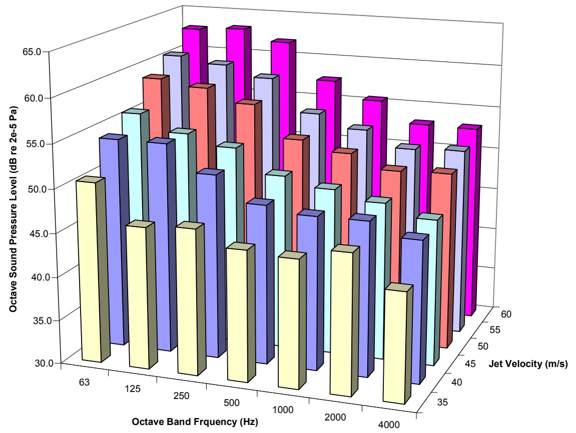


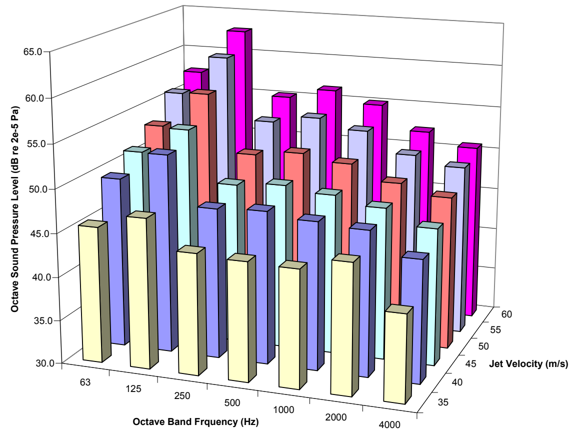
Figure 2. Sound pressure levels for a 50 mm round jet

DIFFUSER OUTLET COMPARISON

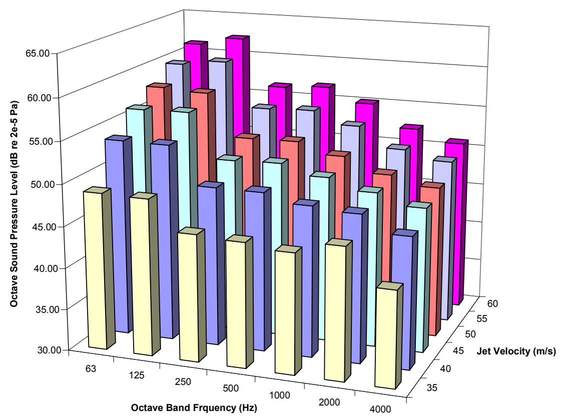
The corresponding sound pressure levels in octave bands for each of the 6 diffuse outlet configurations investigated are shown in figure 3. For the purpose of matching noise criteria curves the individual 1/3 octave band sound pressure levels that were recorded have been translated into the equivalent octave band measurements. The 7˚ conical diffuser (a) had an inlet diameter of 50 mm and an outlet diameter of 150 mm. The straight outlet pipes fitted to the diffuser (b & c) were 300 and 600 mm in length respectively. The 90˚ elbow (d) was fitted to the long outlet pipe from (c) and the outlet was further extended using the short outlet pipe (e) and a 45˚ Y branch (f).



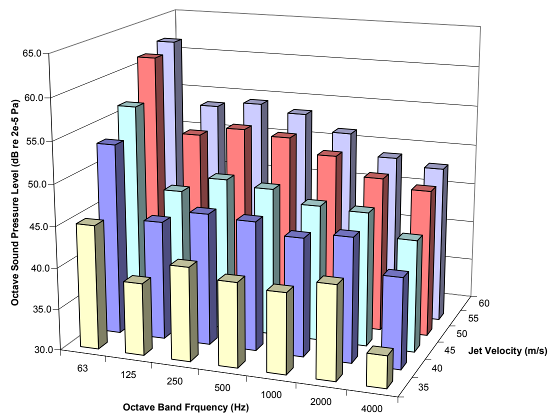
(a) 7˚ Conical Diffuser



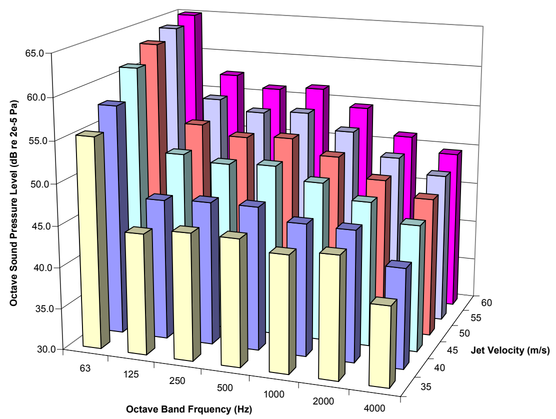
(b) Diffuser with 300 mm Outlet Pipe



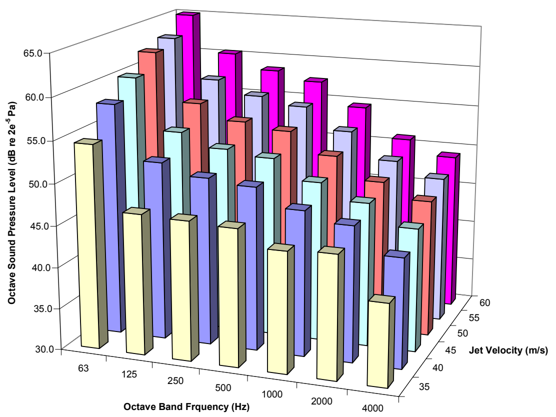
(c) Diffuser with 600 mm Outlet Pipe



(d) 90˚ Elbow added to (c)



(e) 300mm Pipe added to (d)



(f) 45˚ Y branch added to (e)

Figure 3: Octave band sound pressure levels in the receiver room for a jet velocity of 25-60 ms^{-1} for each outlet configuration

The 600 mm long outlet pipe fitted with the 90˚ elbow provided the best attenuation of the expanding jet. The addition of the diffuser had a pronounced effect on the shape of the flow generated noise spectrum, yet the magnitude of the peak octave band sound pressure level was essentially the same. The acoustic energy from the 1, 2 and 4 kHz octave bands was redistributed across the lower frequency bands. Without any outlet pipe the diffuser radiated a very broad band of low frequency noise, while the extended outlet pipe configurations tended to have produced a more concentrated low frequency peak sound pressure level.

In practice a fully operational fast ferry HVAC system would contain multiple outlets in the passenger cabin area causing a significant increase in the total acoustic power. Likewise the total acoustic absorption of the cabin area will be substantially higher than the conditions inside the much smaller reverberation suite. By distributing the sound power across multiple cabin outlets the higher absorption of the cabin space is expected to produce a lower overall sound pressure level. A complete sound power analysis is currently underway to confirm this hypothesis.

Closer inspection of the octave band sound pressure levels for the 35 ms^{-1} jet indicate a RC45 noise rating would be satisfied with little additional attenuation – based on the reverberation room installation. A more realistic jet velocity of 20 ms^{-1} would certainly be well within the RC40 noise rating provided the high frequency SPL's are less than those recorded for the 35 ms^{-1} case. Measurements taken at 25 and 30 ms^{-1} using the alternate outlet configurations suggest this is a valid assumption, as shown in figures 4-6.

A direct comparison between figures 4 and 6 indicates the effectiveness of the conical 7˚ diffuser and outlet duct. Based on this interim success a full HVAC outlet assembly will be tested to validate the application of high velocity HVAC ducts to ocean going fast ferries. Such a system will also have the potential to meet additional niche HVAC applications.

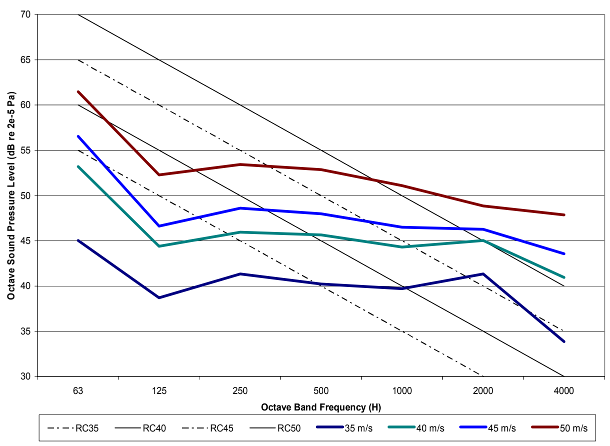


Figure 4: RC noise criteria curve fit for case (d).

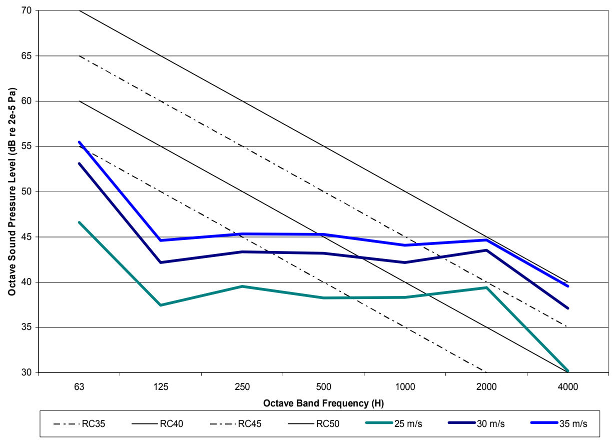


Figure 5: RC noise criteria curve fit for case (e).

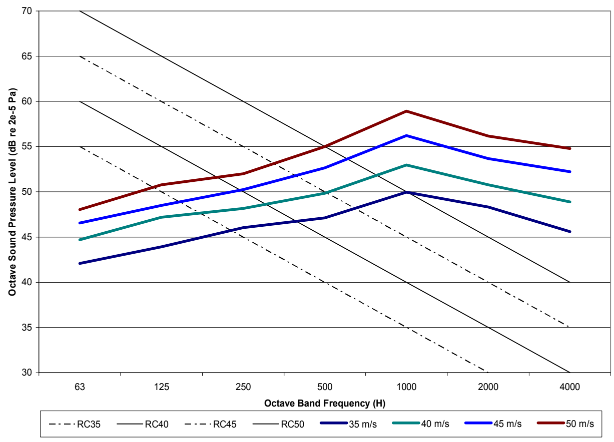


Figure 6: RC noise criteria curve fit for straight 50 mm jet.

Numerical Model

A CFD based numerical model was also developed to predict the flow generated noise (sound power) for a given jet velocity and outlet configuration. The ultimate purpose of the numerical model is to provide a design tool for predicting the noise generated by proposed high velocity HVAC duct systems.

Model formulation

Figure 7 shows the cross section of the three-dimensional mesh model of the 50 mm diameter jet, 7˚ conical diffuser and the surrounding outlet flow domain. The conical diffuser is connected to a 150 mm diameter outlet pipe which in turn is surrounded by an external flow domain with a radial extension of 12 jet diameters. The upstream domain extends 3 jet diameters while the downstream domain extends 36 jet diameters. A more detailed model would incorporate an outlet domain of far greater length and girth with a larger upstream extension. Computational limitations have led to this compromise due to the extensive solution time required to numerically simulate the flow generated noise.

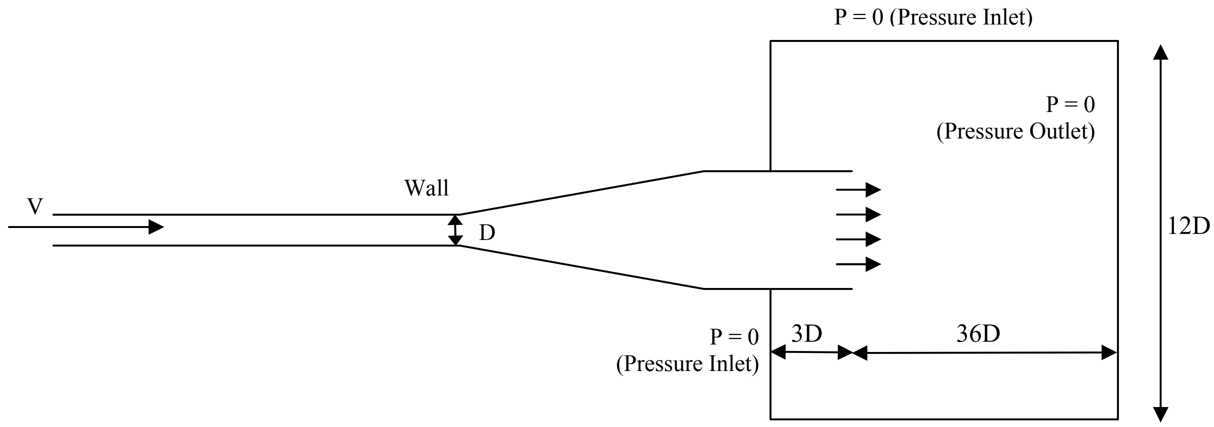


Figure 7: Sectional view of the 7° conical diffuser and the surrounding flow domain

The above model was created and solved using the commercially available CFD package Fluent 6.1. The segregated unsteady solver with a LES viscous model was used to simulate the time varying velocity and pressure within the model. The Smagorinsky-Lilly sub grid-scale model was used with a Smagorinsky constant, Cs of 0.08.

The governing equations for the LES model are derived by filtering the time-dependent Navier-Stokes equations, which effectively filters all eddies on a scale smaller than the filter width (mesh spacing) defined in the model. More detailed information on the derivation of these equations can be found in Ferziger [5].

Boundary conditions

The circular jet of air was modelled as fully developed turbulent pipe flow with a prescribed average mass flux. A fully developed velocity profile was used to define the inlet boundary layer. The external flow domain surrounding the outlet of the diffuser contains a combination of pressure inlets and pressure outlets to fully define the model.

Solution Method

The initial guess for the unsteady LES model was calculated using a steady k-ε turbulence model, using both compressible and incompressible density models. Second order discretization schemes were used for all variables and the PISO scheme was used for the pressure-velocity coupling. The corresponding LES models were then solved using the segregated 2nd order implicit unsteady solver in Fluent 6.1. Central differencing discretization schemes were employed to solve both the momentum and energy equations (compressible model), while the 2nd order upwind scheme was used for calculating density (compressible model). As with the steady k-ε model PISO pressure-velocity coupling was used with second order discretization of pressure.

A time step of 10-30 μs was used for the compressible model and 50-100 μs for the incompressible models. A structured 3-D mesh of 500,000 to 1.3 million cells was used to find the optimum mesh size to adequately resolve the boundary layer and external flow domain while also minimising the total solution time. The compressible models typically required 100,000+ time steps to reach a statistically steady state while the incompressible models only required 10,000 – 20,000 time steps to reach the same level where a suitable noise simulation could be completed. Ideally the model should be run for a greater period of time to improve the quality of the statistically averaged flow variables upon which the acoustic calculation is made.

Numerical results

Time averaging the static pressure and air velocity across the flow domain enables the fluctuating (rms) components to be isolated. Contour plots for fluctuating pressure and main velocity components are shown in figures 8-10. The effectiveness of the outlet diffuser is demonstrated by the reduced pressure fluctuations at the outlet of the duct and the corresponding drop off in the magnitude of the fluctuating velocity components.

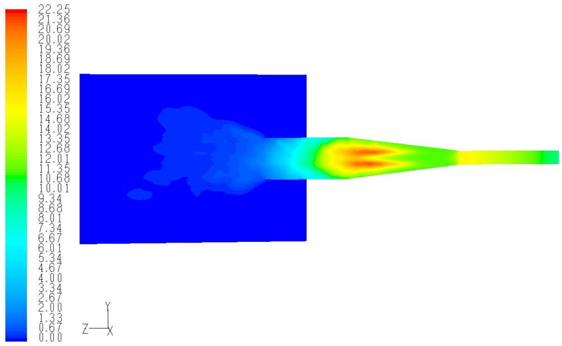


Figure 8: RMS static pressure for 20 ms^{-1} jet

Careful inspection will also show that the fluctuating velocity components are several orders of magnitude greater than the expected acoustic velocities that match the fluctuating pressure. The fluctuating turbulent flow field is masking the acoustic particle velocity – typical of an aero-acoustic problem such as this.

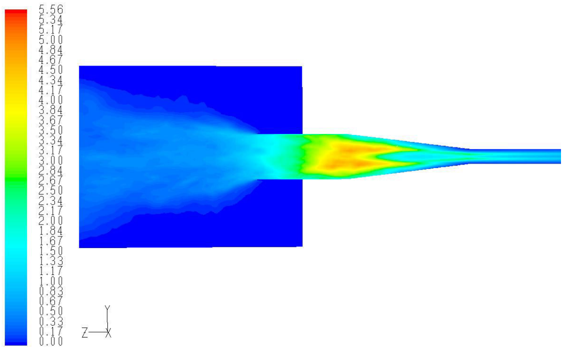


Figure 9: RMS axial velocity for 20 ms^{-1} jet

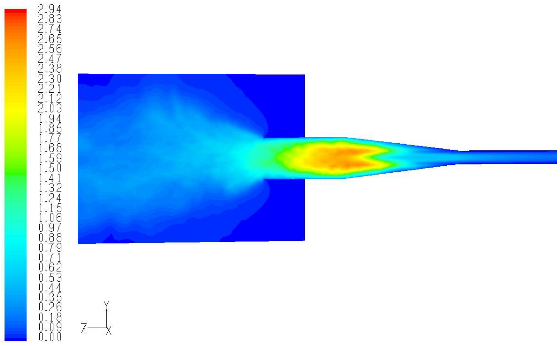


Figure 10: RMS radial velocity for 20 ms^{-1} jet

Acoustic Simulations

An interim noise simulation was carried out using local data at discrete monitoring points within the outlet flow domain and a number of jet centerline velocities. The resultant sound pressure levels (SPL) for a jet centerline velocity of 20 ms^{-1} are shown in figure 11. The SPL was calculated at the diffuser inlet (Jet), diffuser outlet and down stream from the outlet at a distance of both 150 and 500 mm.

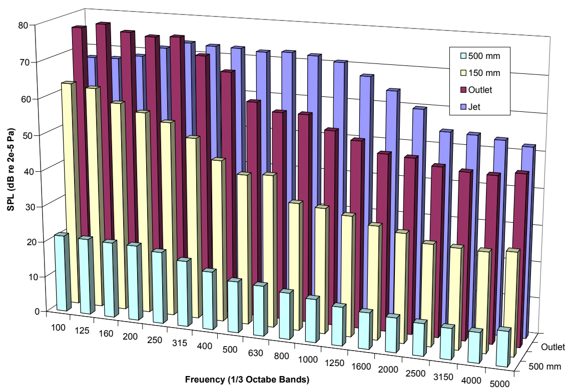


Figure 11: Sound pressure level for 20 ms^{-1} jet

There is a significant drop in the sound pressure with an increase in distance from the duct outlet. A complete sound power analysis is required to match the experimental and numerical models accurately. However the above results suggests that a CFD based numerical model can be used to predict the noise level in such an application. The results illustrated were calculated using a sampling frequency of only 10 kHz and as such will provide limited accuracy in the upper frequency bands presented. Numerical models using an improved mesh and high sampling frequency are currently in development. These methods will also predict the total sound power radiating from the jet by way of a far field acoustic analysis of the CFD flow model.

CONCLUSION

Noise treatment strategies have been identified and tested that enable the use of high velocity HVAC air distribution ducts on ocean going fast ferries. Importantly such a system can be installed with out any significant increase in passenger noise levels above what would otherwise be expected. Preliminary results suggest fitting a 7° conical diffuser and an extended outlet pipe with selected end fittings will enable an air distribution velocity of 20-25 ms^{-1} to be used. The high frequency flow generated noise from the upstream jet was effectively treated with in-line silencers, suggesting that such concerns could be equally rectified in a practical installation.

A numerical model using a LES based CFD model of the outlet diffuser has also been developed to evaluate the effectiveness of different diffuser and outlet configurations. Interim numerical results have shown that prediction of sound pressure and sound power levels at the outlet of such a system is possible. Further development of this modeling process will lead to a suitable design tool capable of evaluating potential high velocity HVAC systems.

ACKNOWLEDGEMENTS

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