SPECIFICATION AND DESIGN OF SILENCERS FOR CONTROL OF LOW FREQUENCY NOISE FROM AN ENVIRONMENTAL TEST FACILITY

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1 INTRODUCTION

Environmental test chambers are used in the automotive industry to verify the resilience of vehicles to varying atmospheric conditions. In just a few hours it is possible to take a car from mid-winter in the Artic, via a high mountain range, to mid-summer in a desert.

Variable speed Roots blowers used to change the temperature, pressure and humidity of the air in the chamber, and these are a major source of tonal low frequency noise which can cause annoyance and disturbance at neighbouring properties if there are gaps in silencer performance.

In the case described here there had been complaints from workers in a nearby building over a significant period of time. Noise from the test chambers had been measured on a number of occasions, but the highly variable nature of the source, the causes of which are described in section 3, meant that it was difficult to determine a clear specification for the required improvement in silencer performance. The variability also complicated the issue of identifying the root cause of the problem since it was difficult to correlate complaints with specific operations of the chambers.

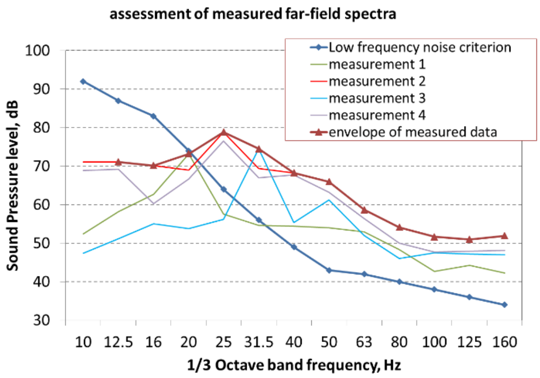
This paper complements a previous conference paper[1], with some wider general discussion of the problem but less detailed technical information.

2 ASSESSMENT OF FAR FIELD LEVELS AND SPECIFICATION OF A TARGET ATTENUATION

An assessment method for low frequency noise is provided by Moorhouse etal[2], which defines a criterion for determining whether a certain level of low frequency noise is likely to lead to complaints. The criterion is normally applied to levels measured inside buildings, whereas only external data at the site boundary were available here. In some circumstances, for example where structural resonances of lightweight buildings are excited, or there are acoustic standing waves in large open plan offices, it is possible that noise levels inside a building could be higher than levels outside. There may also be other adverse effects for occupants of buildings, such as rattling windows and bouncing floors. These factors were not directly considered, but the possibility that they may be contributing to the problem was taken into account by including a margin relative to the standard criterion.

The variability of noise levels at the boundary of the neighbouring property is illustrated in Figure 1a), which shows spectra recorded on four separate occasions compared with the baseline curve for the low frequency noise criterion. The levels exceed the criterion by 10-20 dB at various frequencies in the 20-100 Hz range.

Since the aim of the investigation was to completely avoid future complaints, the cautious approach of considering the envelope of all measured data was taken. The difference between the criterion and the envelope was taken as the minimum attenuation required, shown in Figure 1b), but an additional margin of 5dB was added to account for the uncertainty of the building response. Thus the target for the minimum insertion loss of new silencers was 28dB at 50 Hz and 20dB at 20 Hz, a substantial level of attenuation at these low frequencies.



a)

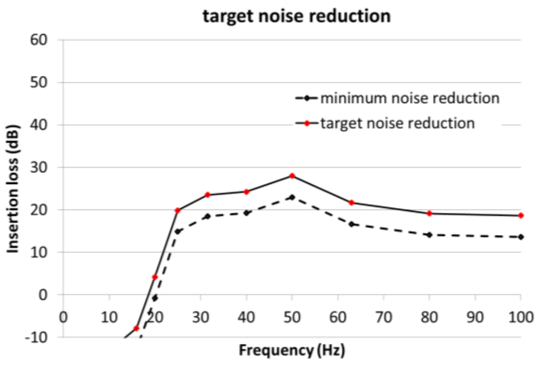


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Figure 1 a) Far-field noise spectra compared with the low frequency noise criterion. b) Target noise reduction for additional silencers.

3 ASSESSMENT OF THE NOISE SOURCE

The chamber ventilation system is shown schematically in Figure 2. There are two independently operated Roots blowers, each of which has two ducts leading to roof level, the discharge duct and the jet intake. The blowers produce tonal noise at harmonics of the lobe frequency, but there were a number of circumstances where they run at close speeds which result in low frequency beating at 1-10 Hz, resulting in complex far-field radiation patterns as the relative phase of the sources changes.

Spatial interference effects could also occur between noise radiation from the jet intake and discharge ducts on the same blower; although these are correlated sources the relative phase is dependent on the speed of sound in the two ducts. Far-field noise levels from the overall system could thus vary widely depending on the relative speeds of the two blowers and the relative noise levels and temperatures of the various ducts.

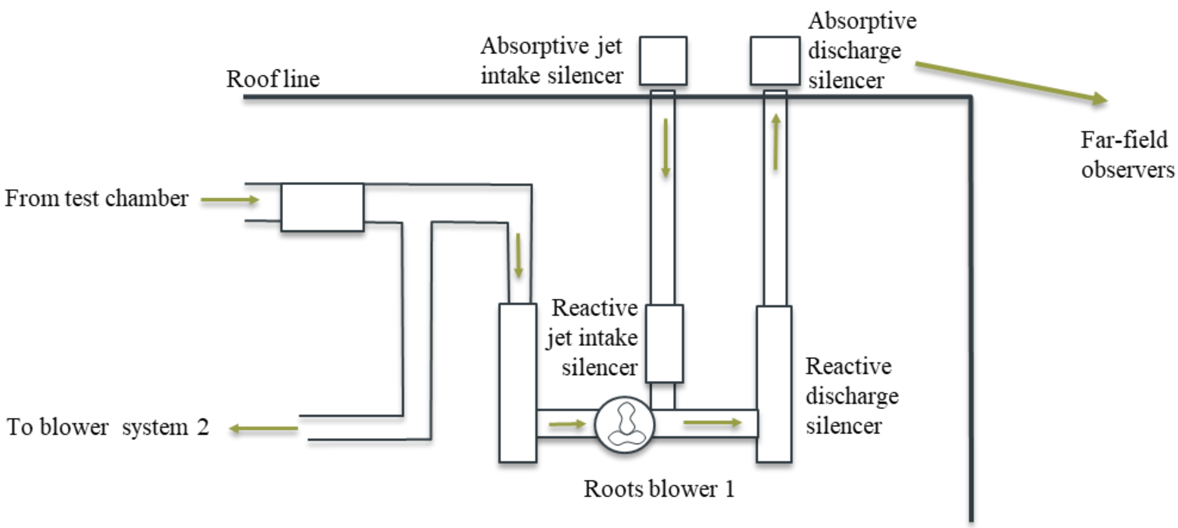


Figure 2: Schematic layout for the ductwork and silencers of each blower system.

One important question in assessing the noise source was the relative contributions of the jet intake and discharge ducts. Roof-top measurements were difficult because the ducts were closely spaced and correlated. In the absence of information from the blower manufacturer, this was determined by lowering a microphone a short distance into each duct and going through a transient run-up and run-down operating cycle for the blower. Taking the max-hold envelope of the instantaneous 1/3 octave band spectra for the two in-duct measurements during the transient gave the result presented in

Figure 3. This data indicates that although the discharge duct has somewhat higher levels at many frequencies, the jet intake had the higher levels below 31.5 Hz. This is likely to be a function of the performance of the different reactive silencers installed in the two ducts.

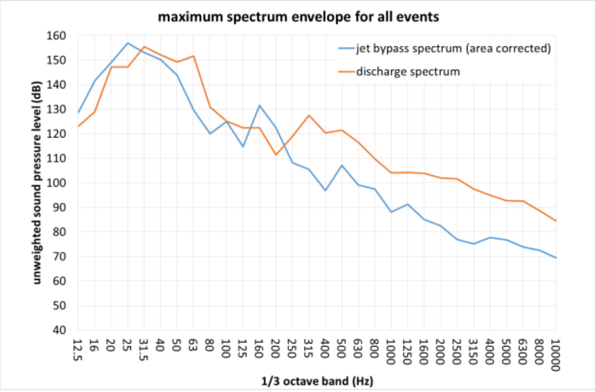
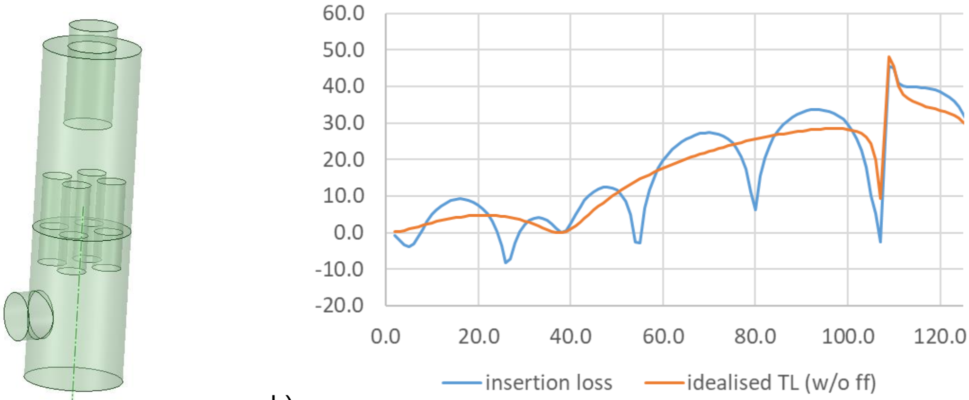


Figure 3: In-duct noise data for the jet bypass and discharge ducts, showing the maximum envelope of all recorded 1/3 octave band spectra.

4 EVALUATION OF CURRENT DUCT AND SILENCER SYSTEM USING FE MODELLING

The schematic layout of the system presented in Figure 2 shows how there was a long tailpipe on each of the reactive silencers, and resonances of these tailpipe were expected to significantly reduce the effectiveness of silencers. Resonances occur when the length of the tailpipe is nominally a multiple of a half wavelength, i.e. the resonant frequencies of an open-open pipe. The design of the existing discharge duct silencer is presented in figure 4a) which comprised two expansion chambers with internal ‘side-branch’ tubes to improve performance at higher frequencies.

The performance of this silencer, including the tailpipe, was predicted using Finite Element method (FE) using the ACTRAN modelling code[3]. Depending on the specified boundary conditions the model can predict both the transmission loss (TL) and the insertion loss (IL) of a silencer. The TL is the theoretical attenuation of the silencer in an infinite duct system, with an incident plane wave at the inlet and no downstream discontinuities. The IL is the performance provided when the silencer is placed into a finite duct system, taking into account the impedance of any upstream components and reflections from downstream discontinuities such as the duct termination.



a) b)

Figure 4: a) CAD schematic of the existing silencer; b) predicted transmission loss and insertion loss of the existing silencer system.

The predicted TL and IL for the existing discharge silencer are shown in
Figure 4b). These results suggest two reasons why there was a noise problem: the TL indicates that the silencer itself provides very little attenuation below 50 Hz, and the IL confirms that tailpipe resonances create significant gaps in attenuation. For the nominal speed of sound of 340 m/s assumed here the resonances occurred at 25, 55 and 80Hz, but because the air temperature on the discharge line could vary from -20°C to + 40°C the speed of sound may vary in the range 318 m/s – 355 m/s, which would significantly alter the frequencies.

Besides modelling the induct silencer performance, ACTRAN can also be used to predict the transfer function from induct sound pressure levels at the duct termination to sound pressure level in the far-field. This model, which includes the use of infinite elements, provides a prediction of the directivity of the sound radiation. In this case the observer was at approximately 200 m lateral distance, at an angle of about 100° relative to the duct axis. The scattering effect of the edge of the building was neglected.

Applying this transfer function to the measured in-duct levels gave the predicted far-field levels presented in Figure 5, showing remarkably good agreement in the key frequency range of 25-63 Hz, especially considering that the induct and far-field levels are both based on spectrum envelopes. The trend at lower and higher frequencies is probably indicative of background noise from other sources.

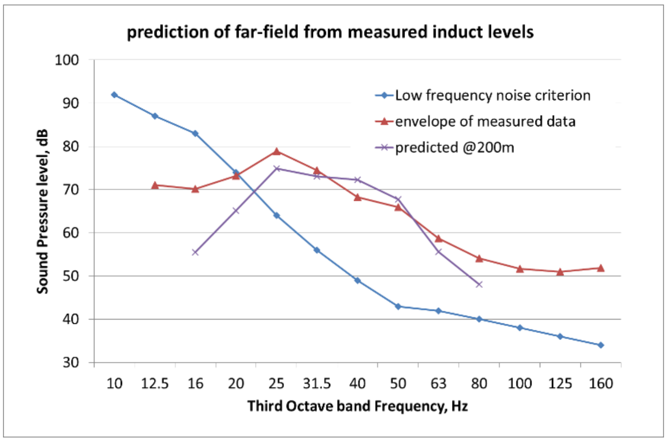


Figure 5: Far-field levels predicted from induct levels using the ACTRAN radiation model.

5 SELECTION OF ADDITIONAL SILENCERS

5.1 Choice of silencer type

There are three main types of silencer for duct borne noise, absorptive, active and reactive, with the following pros and cons:

- Absorptive silencers are not normally considered for low frequencies, but can provide good attenuation in long ducts, particularly where there is space for sufficient depth of absorptive material. An FE model of an idealized design did confirm the potential, but there were commercial disadvantages to such a bespoke design, and durability was a concern.
- Active noise control was considered to have significant potential for this problem as the temperature range of the exhaust gases was relatively low, and the long pipe runs meant that a feed-forward system should work well. However the very high levels of induct noise were considered to be a problem, and although external systems would benefit from the high reflection coefficient of the duct termination, there was concern that there could be reliability and maintenance issues.

- Reactive silencers are the default components used to control low frequency duct borne noise, and in this case could use standard components from the existing silencer supplier. The main issues were that the number of additional large silencers required created a space and weight problem, and it was important to ensure there were no major gaps in performance within the working frequency and temperature ranges.

5.2 Layout of reactive silencers

The target attenuation spectrum in Figure 1b) indicates a requirement for reductions of 20-30 dB in the 25-50 Hz range. This substantial level of attenuation requires either a single silencer with a large expansion ratio or two smaller silencers. In either case, and considering that both jet intake and discharge ducts for the two blowers needed to be treated, it was apparent that there may be a significant weight problem for the building. Space was also an issue, although this could be resolved by installing some silencers on the roof. Potentially also the discharge ducts for the two blowers could be combined to reduce the number of silencers, but this would then need to be increased in size to provide the required flow duct area and expansion ratio. Some weight saving was also possible by removing the existing silencers, but the attenuation from these at higher frequencies was potentially useful.

In the final analysis four options for the discharge ducts were considered, as listed below, with options 2 and 3 being presented in Figure 6. These two designs are quite similar acoustically, but with very different duct lengths and weight implications for the building.

All designs were based on use of one or two additional components: silencer A was suitable for a single blower line, and silencer B was a larger component to be used where the ductwork of blowers 1 and 2 were combined into a single duct. FE predictions for each case were carried out, a process that was facilitated by use of the Transfer Matrix Method[2,3], in which components A and B only needed to be modelled once and the various lengths of ducting were simulated analytically.

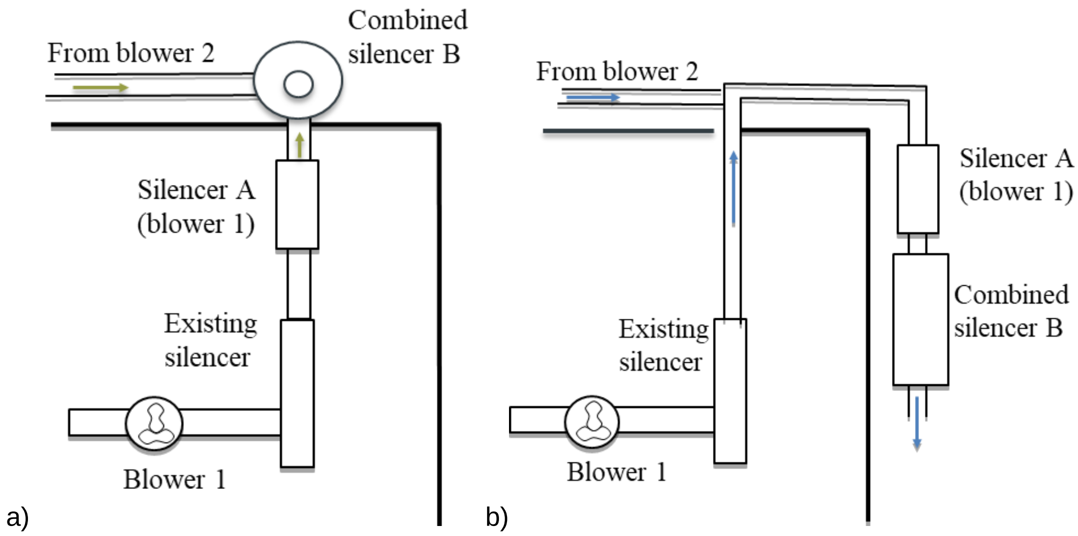


Figure 6: Two of the design options for the discharge duct: a) option 2, silencers A inside the plantroom and B on the roof; b) option 3, silencers A and B outside the plantroom, supported from the ground.

The predicted performance of each option is presented in Figure 7. It is important to bear in mind that a key driver for the design was the level of any dips in predicted attenuation, whereas peaks had no importance. The specification and evaluation of each option for the discharge ducts were as follows:

1. Silencer A fitted inside the plant room on each blower, leaving the existing silencer in situ. This solution could be acceptable, but it was marginal in the 31.5 Hz band.
2. Silencer A fitted in the plant room for each blower and combined silencer B fitted on the roof (Figure 6a). This would definitely reduce the discharge noise to below the criterion, and it may be possible to dispense with the existing silencers to save weight if necessary.
3. Both silencer A and silencer B fitted outside the building and supported from the ground (Figure 6b). This was also predicted to resolve the discharge noise problem. The predicted dips in performance due to the long pipe run between the existing and new silencers were not an issue, and removing the existing silencers to simplify the pipe run was a possibility, but was not necessary for weight reasons. The outlet duct could point down as the exhaust gas was only air close to normal temperatures. Reflections from the facade was considered as an issue since that might add up to 6 dB to far-field levels, but the high level of attenuation meant that this was not a problem.
4. A single combined discharge silencer B fitted on the roof to attenuate both blowers. This relatively low cost option was not a satisfactory solution because the duct length between the existing silencer and the new roof level silencer caused significant dips in performance.

A similar range of design options for the jet intake were also analyzed. Bearing in mind the weight and space constraints applied to the complete system, the final designs selected were:

- For the jet intake, silencer A installed inside the building, as in option 1
- For the discharge duct option 3 was selected, presented schematically in Figure 6b and with the final installation as shown in figure 8c.

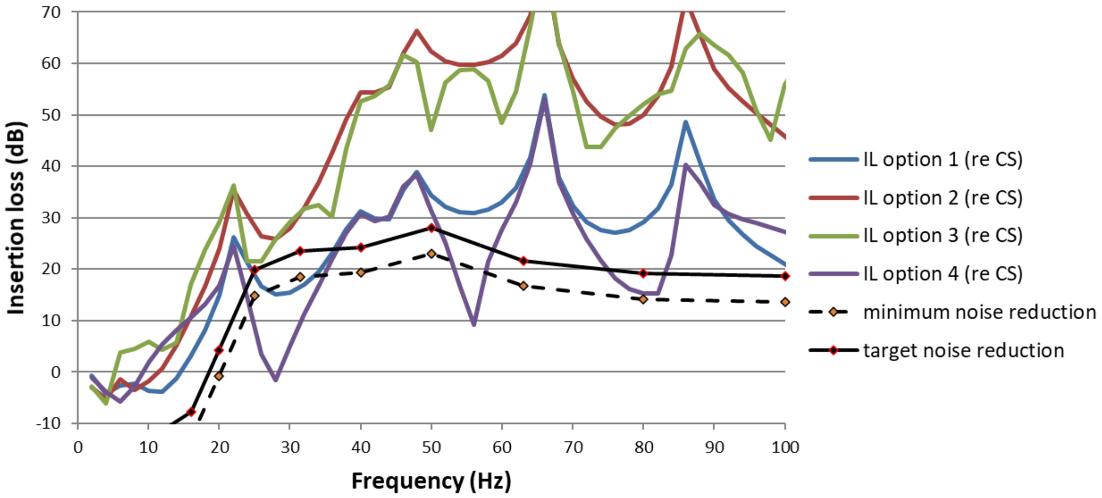


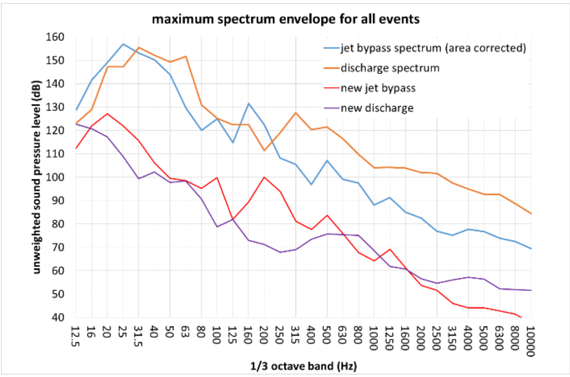
Figure 7: Predicted insertion losses for the proposed discharge silencer options assuming the existing silencer remains in-situ.

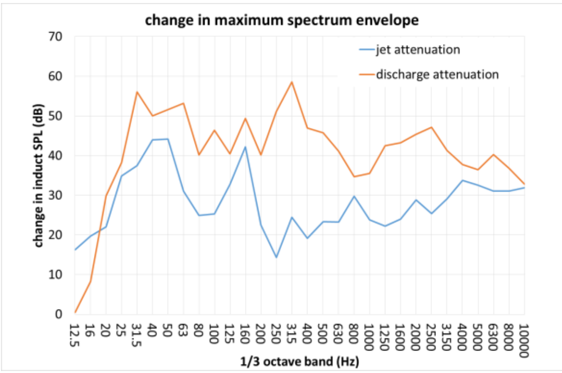
6 FINAL TESTING

A site visit to assess the performance of the installed silencers was made. Far-field levels were too low to be measured against daytime background noise. Instead, induct measurements for the discharge and jet intake ducts were carried out using the spectrum envelope method over a complete transient cycle as before. An area correction needed to be applied to account for the larger outlet duct diameter of silencer B, and the resulting levels were then as shown in Figure 8a). The insertion losses for the single silencer in the jet intake and the two silencers in the discharge duct are presented in Figure 8b).

Since the spectrum envelope method captures the maximum at each frequency, the inferred insertion loss shown is actually the minimum attenuation that was achieved at each frequency, bearing in mind the effect of changing speed of sound during the transient. The predicted minimum attenuation for option 3 shown in Figure 7 was 40-50 dB, which is in reasonable agreement with the measured insertion loss for the discharge duct shown in Figure 8b).

Although the design options for the jet intake have not been presented in detail, the predicted attenuation for the selected single silencer A was similar to that of the discharge duct option 1 plotted in Figure 7. The minima shown are again in reasonable agreement with the measured minimum insertion loss of 20-30dB shown Figure 8b).

a) 

b) 

c) 

Figure 8: In duct sound pressure levels for jet intake and discharge ducts before and after treatment: a) absolute levels, b) area corrected insertion loss. c) Photograph of the installed discharge duct silencers.

7 CONCLUSIONS

This case study highlights the difficulty of determining silencer performance requirements when the sound source is highly variable. The spectrum enveloping method provided a surprisingly easy way of assessing both the level of far-field noise, and hence the target attenuation, and the relative source levels for the jet intake and discharge ducts. The fact that there was good agreement between predicted and measured far-field levels helped to confirm that these were the dominant sources, and that contributions from other paths, such as sound transmission through the building facade, could be neglected in the first instance.

The low frequency noise criterion is normally applied to levels recorded indoors, whereas that information was not available here. Making an allowance for the fact that in some circumstances levels indoors could be higher than outdoors helped boost confidence that the proposed design solution would resolve the complaints.

Most important in the methodology however, was the use of FE to accurately assess the minimum insertion loss that would be achieved by each design option. The good correlation between predicted and measured minimum insertion losses for the final design confirms the efficacy of the approach used, which was the key to the completely successful outcome of the project.

8 REFERENCES

1. Quaranta E. and Smith M.G. Control of low frequency noise from an environmental test facility. *Internoise 2021, paper 2421*.
2. Moorhouse, A. T. & Waddington, D. C. & Adams, M. D. A procedure for the assessment of low frequency noise complaints. *Journal of the Acoustical Society of America,* **126**, 1131 (2009).
3. *Actran User's Guide*, Free Field Technology, Belgium.