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LOW FREQUENCY NOISE RADIATION FROM A COAL WASHERY SCREEN

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ABSTRACT
Vibration of components in buildings in the area of a coal washery was attributed to a large de-watering screen. Measurements were made of the environmental noise levels and a mathematical model of the acoustic source was constructed. The model indicated that the screen sump vent was augmenting the radiation from the screen itself. A strategy to de-tune the system was implemented and follow up measurements showed a clear improvement.

1 - INTRODUCTION
Vibration of components in buildings near a coal washery were observed. At 16.3 Hz, the sound did not constitute a noise annoyance, but drove sympathetic vibration of the structure of nearby buildings which in turn rattled windows, light fittings etc... Preliminary measurements indicated that the noise was generated by a vibrating coal de-watering screen. CSIRO Thermal and Fluids Engineering was invited to investigate the problem and to recommend a reduction strategy. This paper describes environmental noise measurements, mathematical modelling of the acoustic source and development of an improvement strategy.

2 - ENVIRONMENTAL MEASUREMENTS
From noise measurements made at various locations, there was clear evidence of a strong tone at 16.3 Hz, which was the operating frequency of the washery screen. A 15.5 Hz tone, shown in the typical spectrum in Figure 1, was not associated with mine operations.

3 - ACOUSTIC MODEL OF THE NOISE SOURCE
The de-watering screen comprised two nominally flat, rigid, porous screens, fixed in a box-frame which oscillated at 16.3 Hz and 10 mm peak-to-peak displacement at 45° to the screen surface. The screen was situated above a metal sump with almost no clearance except at one end, where the sump extended past the screen box, providing a vent. The water level in the sump varied from about 1.3 m to 3.5 m below the lip of the sump.
The top of the screen radiated sound directly into the environment while pressure fluctuations in the sump drove an oscillating air column in the vent, which radiated sound also. The relative phase depended on where the frequency lay relative to the system resonance.

3.1 - Model of the sump and vent
The effective volume displacement of the screen is represented by a complex variable $\boldsymbol{Qoe}$ (Re[$\boldsymbol{Qoe}\ e^{j\omega t}$], where ω is the radian frequency, t is time and complex variables are indicated by bold characters). The volume displacement of the vent is given by the equation for vibrations of a spring-mass-damper system driven by base displacement [2]

$$Qoe/Qv = 1 - \xi^2 + \mathrm{j}\ 2\ \ \zeta\ \xi \qquad (1)$$

where $\boldsymbol{Qoe}$ is the effective volume displacement of the screen (+ve inwards), $\boldsymbol{Qv}$ is the volume displacement of the vent (+ve outwards), ξ is the frequency ratio ω/ω_0, ω_0 is the natural radian frequency of

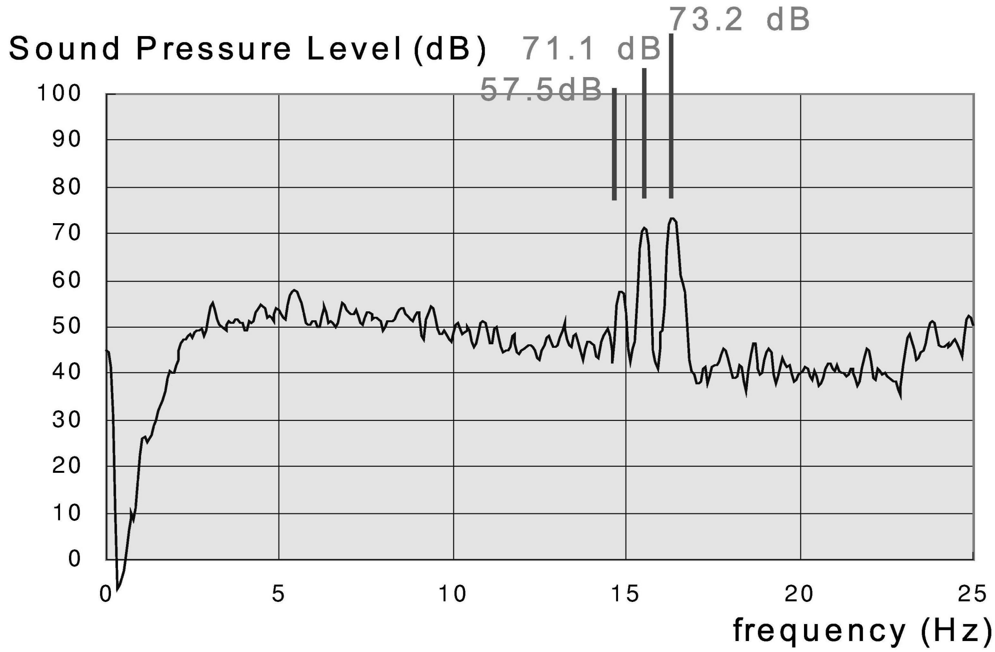

Figure 1: Spectrum of sound 1.5 km from washery, mid sump level, before modification.

the resonator ($\sqrt{k/m}$), m is the effective mass of the air column in the vent, k is the spring stiffness of the sump volume, ζ is the damping ratio of the resonator d/dc, d is the mechanical damping coefficient and dc is the critical damping coefficient $2\sqrt{mk}$.
The net source strength $\boldsymbol{Q}$ is obtained by adding the contributions of the screen and vent.

$$Q/Qoe = 1/\left(1 - \xi^2 + \mathrm{j}\ \ 2\ \zeta\ \xi\right) - 1 \tag{2}$$

3.2 - Influence of screen porosity
Since the screen was porous, the effective source strength $\boldsymbol{Qoe}$ of the screen was equal to the nominal displacement volume $\boldsymbol{Qo}$ discounted by the volume of air which slips through the screen. The slip velocity $\boldsymbol{U}$ was assumed proportional to the pressure difference across the screen and inversely proportional to the flow resistance R of the screen. That led to the relation

$$\begin{aligned} Qo/Qoe = \rho\ c^2\ So/\left(\mathrm{j}\omega RV\right)\ \left(1 - Qv/Qoe\right) \\ +\rho\ c/R\ \left[R_1(2ka\ ) + \mathrm{j}\ X_{\ 1}(2ka)\ \right] + 1 \end{aligned} \tag{4}$$

where c is the speed of sound, So is the area of the screen, R_1 is the piston acoustic resistance function and X_1 is the piston acoustic reactance function [1].

3.3 - Amplification caused by the Sump and Vent
The analytical model described above was programmed on an XL spreadsheet. The influence of screen, sump and vent geometries were incorporated to allow estimates for various configurations. The following data was used for the base configuration: sump volume V = 28.7 m ^{3}, So = 21 m^2, vent area S = 1.87 m^2, vent column length l' = 0.93 m, f = 16.3 Hz, ζ = 0.01, R = 1500 MKS Rayls. The source strength of the base case relative to a rigid impervious piston of the same nominal volume displacement is shown in Figure 3 left as a function of driving frequency. The strength at the driving frequency exceeds that of a rigid impervious piston by nearly 3 dB.

4 - STRATEGIES FOR NOISE REDUCTION

Since the operating frequency is close to the resonance peak, it was possible to reduce the sound level by increasing the resonant frequency of the vent-sump system. The model predicted a reduction of 11.4 dB when the vent area was increased from 1.87 m^2 to 8 m^2 as shown in Figure 3 left to centre.
The resonant frequency could be increased and the peak height reduced by reducing the volume of air V above the water in the sump e.g. by allowing the water level to sit higher. Figure 3 right shows the effect of reducing the sump volume V to 15 m^3. The result was a decrease of 6.2 dB.

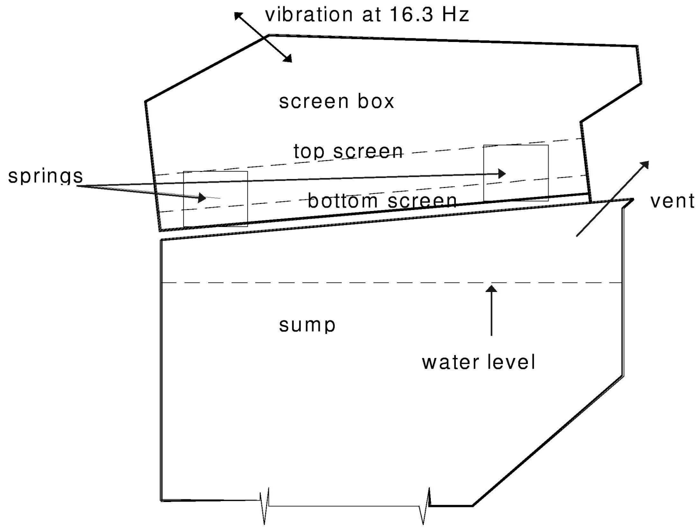

Figure 2: Diagram of screen and sump.

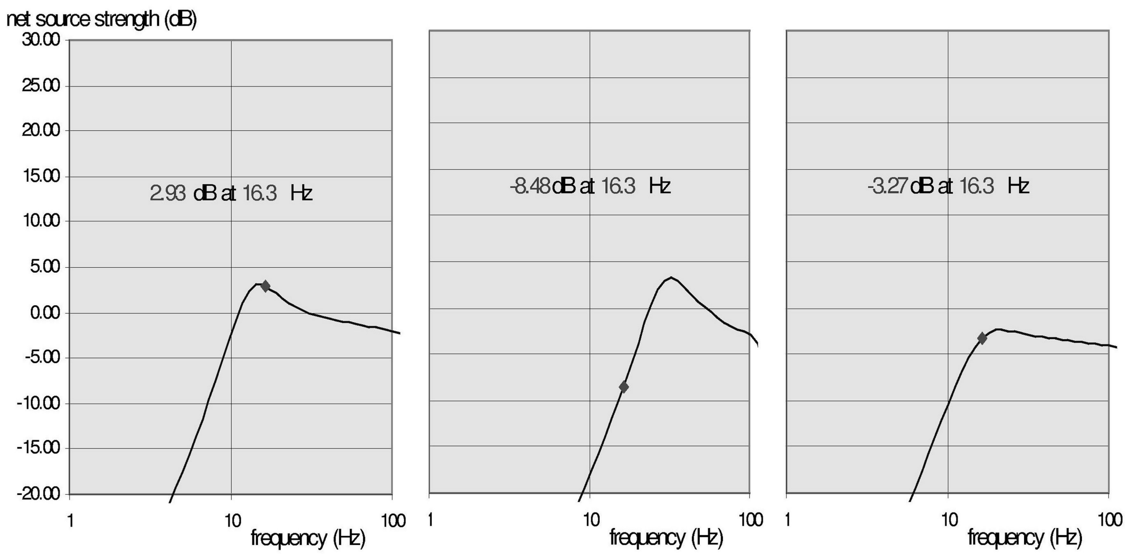

Figure 3: Source strength relative to rigid piston with screen displacement at 16.3 Hz; left: base case, vent area $S = 1.87$ m^2, sump vol. $V = 28.7$ m^3; centre: $S = 8$ m^2; right: V = 15 m^3.

5 - BEFORE AND AFTER COMPARISON

The effective vent area was increased from 1.87 m^2 to 3.93 m^2 by cutting holes in the sump walls. The mathematical model predicted a reduction in sound of 3 dB for this case. In comparing results of measurements before and after the sump modifications, allowances must be made for the variations in water level because control of the level was limited due to operational requirements.

Figures 1 and 4 show spectra of measurements made at one location before and after the sump modifications with low and high sump levels.

Using the model to account for variations in sump level, the following observations on the before and after measurements were made:

- The model predicted that an increase in the sump openings from 1.87 m^2 to 3.93 m^2 would decrease the sound generated by 3 dB for all other conditions constant. The measurements indicated a reduction of 4 dB.
- The model indicated a decrease of about 10 dB in sound as the sump level increased from by 1.5 m. Measurements indicated a corresponding decrease of about 6 dB.
- Both model and measurements indicated that the sound was sensitive to screen porosity when the

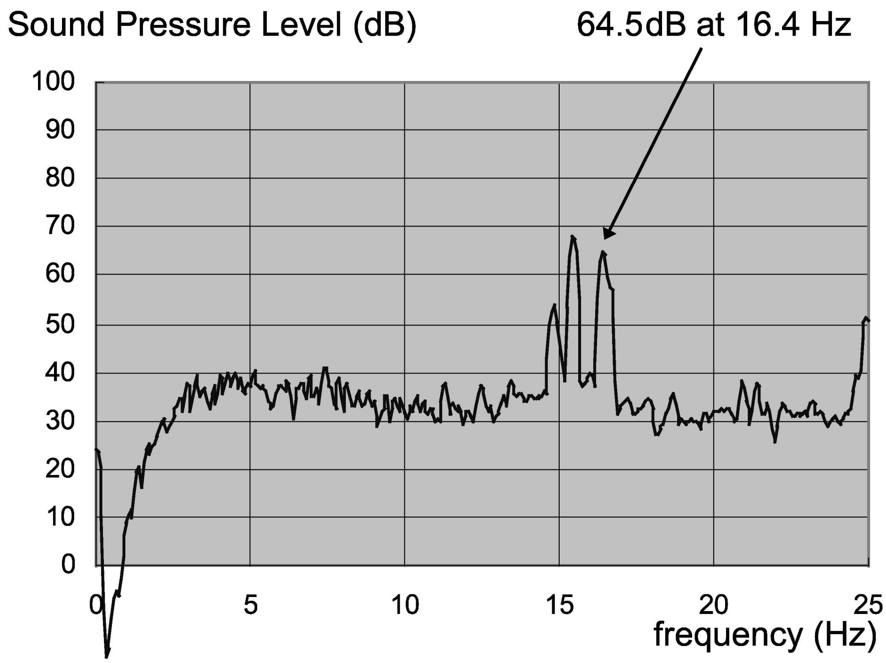

Figure 4: Sound spectrum 1.5 km from washery, high sump level, after modification.

screen-sump-vent system was operating near the resonant frequency but much less away from the resonance.

6 - CONCLUSIONS

Low frequency noise was generated by a vibrating coal washery screen. A mathematical model of the acoustic source indicated that the sump and screen were acting as a vented enclosure.
By increasing the effective vent area from 1.87 m^2 to 3.93 m^2 the mathematical model predicted a reduction in sound of 3 dB at 16.3 Hz, while measurements indicated a reduction of about 4 dB.
The sound level was also sensitive to water level in sump. A reduction of about 8 dB was predicted for a water level increase of 1.5 m. Environmental measurements gave results that were consistent with this.

REFERENCES

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2. **L L. Beranek and I. L. Ver**, *Noise and Vibration Control Engineering*, Wiley and Sons, 1992