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VIBRATION NEUTRALISERS FOR CONTROLLING PUMP NOISE

M J Newman & K Nordmark

IGP AS, Brattorgt 5, Post Box 2564, N-7001 Trondheim, Norway

1. INTRODUCTION

Low frequency ground borne noise from a water supply pumping station in Trondheim had caused complaints from neighbours after a new set of pumps were installed. Many methods had been tried to reduce the noise in the neighbouring dwellings.

In this paper the results from the implementation of vibration neutralisers to the water pump units are presented. The design of the neutralisers is novel in that they attenuate vibration in 2 translational and 2 rotational directions.

The Jakobsli pumping station was extended in 1991 and three pumps installed. The pumps can deliver 225L/s of water with a head of 53m, the flywheels weigh 300 kg and the electric drive motors are rated at, 175kW, 1485 RPM, 50Hz.

2. THEORY/BACKGROUND ON VIBRATION NEUTRALISERS

The addition of auxiliary systems to attenuate vibration (and thereby noise) is a well tried and tested control measure. The accepted classical reference is Den Hartog [1]. The technique is similar to some methods used in active vibration control, where a virtual earth is produced that effectively eliminates the vibration; as this is a passive technique it only works at one frequency/auxiliary system.

The auxiliary system consists of, in its simplest form, a mass and a spring. To analyse systems with more than several degrees of freedom, mobility methods provide useful insight into the effects of adding an auxiliary system.

Mobility is defined as velocity/force and is the inverse of mechanical impedance. For lumped parameter components, point mobility is defined where they are attached to another system e.g. a spring $M_{spring} = i\omega/k$ where ω is the radian frequency and k is the spring constant. For distributed parameter systems (all real components) transfer mobilities are also required to describe the effect caused by a force at one point (B) on the velocity at a second point (A), M_{AB}. These are used to obtain the effect that the auxiliary system has on the original system away from the attachment point. Figure 1 shows a block diagram of the original vibrating system and the auxiliary system.

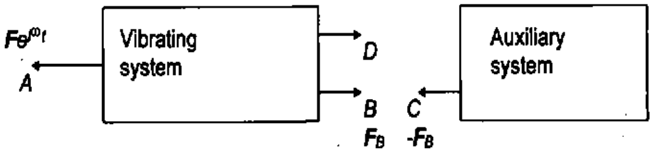

Figure 1. Vibrating and auxiliary systems

The mobility of the auxiliary system is: $M_{CC} = M_{kA} + M_{mA} = \dfrac{i\omega}{k_A} - \dfrac{i}{\omega m_A}$

and $M_{CC} = \dfrac{-i\omega}{k_A}\left[\dfrac{1-\Omega_A^2}{\Omega_A^2}\right]$ where Ω_A is ω/ω_A, when $\Omega_A = 1$ then $M_{CC} = 0$ and point C cannot move. This ignores the effect of damping, it also predicts large amplitude displacements of the auxiliary mass.

The velocity at D (the receiver position) due to the force at A after the auxiliary system has been added at B is: $\dot{x}_B = F_A M_{BA} + F_B M_{BB}$ the velocity at C is $\dot{x}_C = -F_B M_{CC}$, when these are coupled together ($\dot{x}_C = \dot{x}_B$) and $F_B = -F_A M_{BA}/(M_{BB} + M_{CC})$.

At the receiving point D the velocity is modified (attenuated) by the auxiliary system:

$$\dot{x}_D = F_A M_{DA} + F_B M_{DB} = F_A\left[M_{DA} - \frac{M_{DB} M_{BA}}{M_{BB} + M_{CC}}\right].$$

The effectiveness of the auxiliary system in attenuating the vibration is dependent on both the force F_B that it generates and the transfer mobilities M_{DB} and M_{BA}.

Two extremes in the effectiveness of the auxiliary system give an indication of the importance of positioning of the attachment point. If it is placed at the excitation position A, B is coincident with A and $M_{BA} = M_{AA}$, $M_{BB} = M_{AA}$, $M_{DB} = M_{DA}$ hence: $\dot{x}_D = F_A \left[\dfrac{M_{DA} M_{CC}}{M_{AA} + M_{CC}}\right]$. At the auxiliary system resonance global attenuation is achieved, theoretically $\dot{x}_D = 0$. The vibrating system is stationary at that resonance frequency and the auxiliary system vibrates with high amplitude, neglecting the effects of damping. If the auxiliary system is placed at a node of vibration, for its resonance frequency, on the vibrating system, $M_{BB} = 0$, $M_{DB} = 0$ and $\dot{x}_D = F_A M_{DA}$, which is unaffected by the addition of the auxiliary system. Within active vibration control these effects are calculated as the controllability of a system [2].

3. PRACTICAL INSTALLATION

Vibration isolation of the pump units during the design stage would have been the easiest solution to ground borne noise. There was resistance in the design team to this solution. Hence, for a fixed speed machine, vibration neutralisers were a natural choice of retrofit measure.

Figure 2 shows a sketch of the pump foundation with the vibration neutralisers. The neutralisers are constructed with steel rods and weights that give sharp resonance's. The neutralisers were symmetrical hence they can attenuate vibration in two axes and in two rotational directions. The lowest set of neutralisers on the columns were made with 50mmØ steel rods and 10mm steel plate 300x300mm 9off. The remaining neutralisers were made with 25mmØ spring steel rods and 10mm steel plate 200x200mm 19off for 25Hz and 10mm steel plate 6off for 74Hz. In total the extra weights weighed approx. 1000kg/pump.

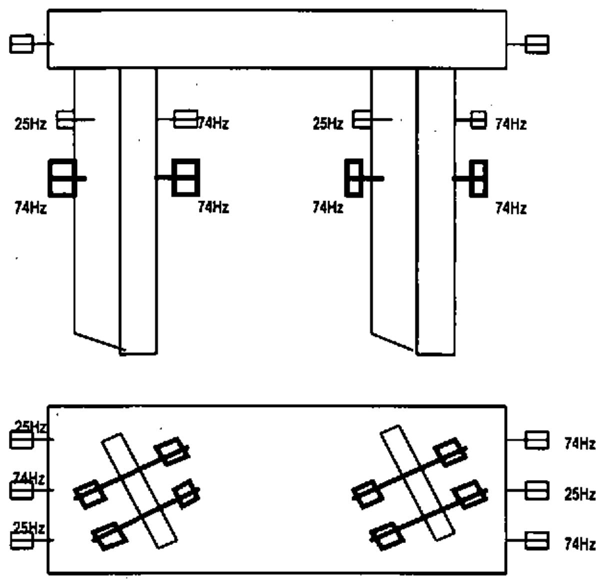

Figure 2. Neutralisers placement on the pump foundations

4. RESULTS

The final measure of the effectiveness of the neutralisers is given by the reduction in the noise levels in the neighbouring dwellings. Figure 3 shows the 1/3 octave spectra before and after implementation. In the bands (25 and 80 Hz) where the neutralisers were tuned in, the noise is attenuated by 12dB and 13dB. The pump units were the main sources of noise from 25 to 160 Hz, above this frequency (6 blades 158Hz) other noise sources dominated. In figure 3 the A-weighted noise level from 25 to 160 Hz is shown for the before and after situation. The noise levels have been reduced by 8dB.

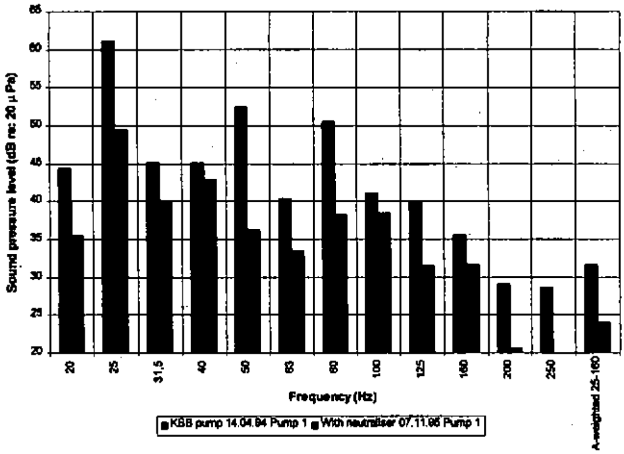

Figure 3. Sound pressure levels measured in a neighbouring dwelling

5. CONCLUSIONS

The results show that vibration neutralisers are well adapted to attenuating noise from constant speed machines. The theory indicates that correct positioning of neutralisers is essential to achieve the desired results.

Comparison of noise levels show that the installation of vibration neutralisers on the pump foundations, has reduced the noise levels with two pumps running from 32 dB(A) to 28 dB(A). With one pump running the noise levels have been reduced from 32dB(A) to 24 dB(A).

6. ACKNOWLEGEMENTS

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7. REFERENCES

[1] J.D. Den Hartog, Mechanical vibrations (McGraw-Hill, 1956)
[2] L. Meirovitch, Dynamics and control of structures (McGraw-Hill, 1989)