Proceedings of The Institute of Acoustics

# NOISE CONTROL BY ENERGY FLOW RESTRICTION FOR IMPACT MACHINES

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## Introduction

The noise radiated from a machine structure can be related to the structure parameters and the operation of the machine, such as structure response, radiation efficiency, structural damping, and bulkiness of the machine structure and the shape of the force pulse. Such a relation will give the effectiveness of modifications in the machine with regards to noise control.

## Energy Balance

The energy that escapes into the structure can be dissipated either as heat due to the structural damping or as radiated noise. Typical machine structures have a high loss factor, seldom lower than 0.01. Therefore an expression for the radiated A-weighted noise energy in frequency bands is given by [1].

$$E_{rad}(A,\ f_o,\ \Delta f) = E_{escape}(f_o,\ \Delta f)\left(\frac{A\sigma_{rad}}{f_o}\right).\ \frac{1}{\eta_s}\ .\ \left(\frac{\rho_o c}{\rho_m 2\pi d}\right) \qquad (1)$$

where $\rho_o c$ is the air impedance per unit area, and $\sigma_{rad}$, $\rho_m$ d, $\eta_s$, the radiation efficiency, material density, bulkiness factor and structural loss factor, respectively, of the machine structure; $E_{escape}(f_o,\ \Delta f)$ is the energy that escapes into the structure in frequency band $\Delta f$ with central frequency $f_o$.

$E_{escape}$ can be expressed in terms of the structure response of the machine at the point of impact and the force pulse spectrum as in power flow methods [2]. Defining a structure response as $V(f)/\dot{F}(f)$, where $V(f)$ is the surface normal velocity at the point of impact in the frequency domain and $\dot{F}(f)$ is the spectrum of the first derivative of the force function. The energy escape is

$$E_{escape}(f_o,\ \Delta f) = \mathrm{Im}[H(f_o)]\ \ |\dot{F}(f_o)|^2\left(\frac{\Delta f}{\pi f_o}\right). \qquad (2)$$

The force derivative spectrum is used since with a static force (zero frequency) no noise is radiated.

## Noise Control Methods

From expressions (1) and (2) it can be deduced that the noise radiated from the machine structure is related to different parameters which can be changed individually and the contribution to noise control due to the modification of each single parameter can be estimated. From these expressions the possible methods of noise control can be listed as follows:

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(i) Increasing the loss factor. This is not always a possible task in a machine structure. Due to the number of parts that are bolted together, the loss factor is already high, and it would be very difficult, using available damping methods, to increase by more than two or three times which will only result in 3 to 4 dB reduction in noise.

(ii) Modifying the radiation efficiency curve. To change this parameter the basic design of the structure must be altered. This is very difficult on already established machines, where the design has been developed over many years just through experience.

(iii) Increasing the bulkiness of the structure. This can only be achieved with a weight penalty which is not desirable.

(iv) Reducing the escape energy into the radiating structure. This is the area where noise control can be achieved to very high levels. The escape energy can be reduced by reducing the structural response, or altering the shape of the force pulse.

## Restricting Energy Flow

If the force pulse duration is increased, energy is transferred from impactor to structure at a lower frequency. If the impactor has a constant velocity before impact, the magnitude of the force pulse for the longer duration is lower, resulting in less energy escaping into the structure, giving a lower radiated noise level. This method of noise control, i.e., by increasing the duration of the force pulse may not always be possible, because this will mean that the process of the machine is being slowed down, resulting in lost production.

A better method for restricting the flow of energy is to reduce the structure response. If a mobility mis-match is introduced in the energy flow path, most of the escape energy is reflected back leaving the structure at a lower level of vibration. Changes in the structure to produce mobility changes can be either due to different materials or to structural configurations, such as resilient layers, bends, changes in cross-section and mass loading. At moderate and high frequencies an attached mass to the machine structure (referred to later as blocking mass) at the point of impact will block most of the vibrational energy without changing the shape of the force pulse.

## Experimental Methods

The measured results for noise radiated from a flat plate are compared to the theoretical estimates using expressions (1) and (2). Changes in the structure response were investigated by putting a blocking mass at the point of impact, both directly fixed and fixed via a resilient pad to the plate. Since the noise energy radiated is estimated in one-third octave bands and the modal density of the plate is high in the frequency range 100 to 10 kHz, a mean value for the response of the plate is taken. This is equivalent to the response of a similar plate but of infinite extent. The point response used is [3]

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$$I_m|H(f_o)| = \frac{1}{16\pi f_o \sqrt{B\rho_s}} \quad \text{where} \quad B = \frac{EI}{1 - \nu^2} \tag{3}$$

where $\rho_s$ is the surface density and $I$ the second moment of area per unit width about an axis in the neutral plane of the plate, $\nu$ is Poisson's ratio and $E$ the Young's modulus. Figure 1 shows the estimated and measured radiated noise energy for two different plate dampings. The agreement between the estimated and measured is within $\pm$ 3 dB which shows that the energy balance expression gives good estimates of the radiated noise from structures.

Other measurements were done when the plate was impacted via the blocking mass, to investigate the resulting reduction in the radiated noise. The point response used in this case is obtained from a mass, spring, substructure (in this case a plate) model. The spring having a large stiffness for the case when the blocking mass is directly fixed to the plate. In this latter case, the mean response is given by |3|

$$Im|H(f_o)| = \frac{1}{8\pi^3 f_o^3 m^2} \cdot \frac{Q}{Q^2 + (Q + \frac{1}{2\pi f_o m})^2} \tag{4}$$

and for the case with the blocking mass on the resilient pad of stiffness $K$ and loss factor $\eta$

$$Im|H(f_o)| = \frac{1}{8\pi^3 f_o^3 m^2} \cdot \frac{Q}{Q^2 + (\eta + \frac{2\pi f_o}{K})^2 + (\frac{2\pi f_o}{K} - \frac{1}{2\pi f_o m} - Q^2 + \eta^2 \frac{\pi f_o}{K} 2Q} \tag{5}$$

where $Q$ is the mean mobility of the plate, $Q = (8\sqrt{B\rho_s})^{-1}$ and $m$ is the blocking mass.

Comparing these responses with the response of the plate, these are much lower especially at high frequencies (figure 2). This reduction in the structure response is also obtained in the radiated noise from the plate (figure 3). The measured noise energy in the case when the plate is excited via the blocking mass on the resilient pad is not as low as estimated due to the reduction in the structure response level. The reason is that since the plate was excited by impact from a steel sphere of 100 mm diameter, the acceleration noise |5| is dominant at the high frequencies over the ringing of the plate.

## Concluding Remarks

Figures 2 and 3 show that a large attenuation is possible in structure-radiated and structure-borne sound by altering the structure response to reflect most of the incident energy. By restricting the energy flow, more energy remains in the impactor and this results in rebound. This was observed in the experimental methods. If the impactor rebounds, secondary impacts with other structure parts may occur and the same amount of energy will be transferred. Also due to rebound subsequent impact precision may be lost. Because of such problems it is not possible to offer a general solution to all industrial machines, but each machine has to be studied individually. Structure modifications should be applied to obtain optimum noise control without affecting the machine operation.

References

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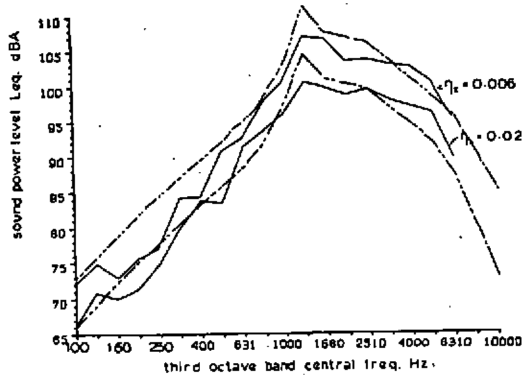


Fig. 1. Noise energy radiated from plate for different loss factors
estimated
measured

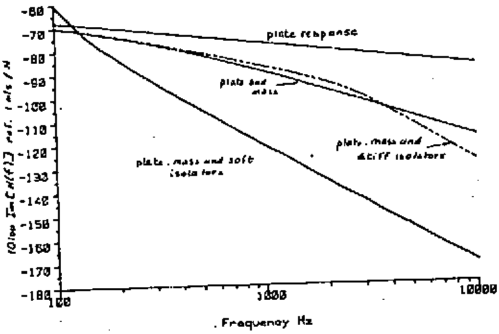


Fig. 2. Structure response with addition of the blocking mass and soft and hard resilient pad.

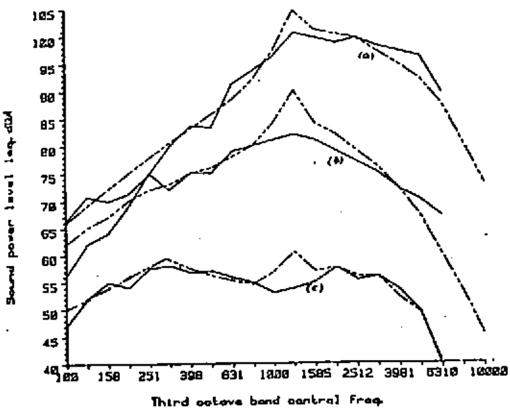


Fig. 3. Radiated noise energy for equal momentum change for different excitation of plate. Showing the high attenuation due to change in structure reponse.
measured
estimated
(a) plate directly excited
(b) excitation through blocking mass
(c) excitation through blocking mass coupled to plate via isolators