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THE DYNAMIC ABSORBER AS A GUIDE MOMENT COMPENSATOR

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INTRODUCTION

This paper describes a method of reducing the noise radiated into the water from a ship hull. Underwater noise is recognized as a problem in certain marine activities such as seismic profiling, operation of navy ships, etc. On the ship in question it was found, by using narrow band frequency analysis, that the primary noise source was the rigidly mounted main engines. The underwater noise spectrum was dominated by pure tone components with frequencies of 25 Hz, 31.25 Hz and 62.50 Hz, the 31.25 Hz component being the dominating component. With a specially developed technique, the intensity vector field of the radiated underwater noise was determined. This investigation showed that the main part of the radiated sound power originated from the hull structure just below the main engines and gears, the hull being driven in a bending mode.

NOISE SOURCE

The main engine is a 5 cylinder 4-stroke diesel engine operated at a fixed speed of 750 rpm, delivering 750 HP. This means that the dominating frequency component, 31.25 Hz, originates from the 2 1/2 order guide moment. The guide moment, which makes the engine rock about a longitudinal axis, can be interpreted as the reaction of the engine to the torque variations during an engine cycle.

METHODS OF SOURCE LEVEL REDUCTION

Several methods can be implied to reduce the movement of the bottom of the ship due to the rocking motion of the engines. One of the traditional, well proven, solutions is to mount the engine resiliently. This is, however, a rather expensive solution. As this engine runs at a constant speed and only one frequency component dominates the noise, the application of a dynamic absorber was investigated.

CALCULATION BASIS

The dynamic absorber is basicly an auxiliary spring mass-system with the exciting force at the spring, and tuned to the exciting frequency of the primary system (engine/hull structure). In the frequency range of interest, calculations show a great number of natural frequencies of the hull structure and therefore it is not possible, with a reasonable effort, to calculate the impedance of the primary system with analytic methods alone. Measuring the magnitude of, and the phase relations between v_x and v_y on both sides of the engine, makes it possible to determine the position of the longitudinal axis of rolling. By combining this with a simplified model, representing the system at 31.25 Hz, and assuming that the engine acts as a rigid body, it is then possible to estimate the impedance of the engine foundation. The guide moment M_{yy} drives the engine, which has a moment of inertia I_{ra} about the rolling axis. The following equation exists, representing the foundation forces F_x, F_y as a couple of magnitude $F \cdot 2r$, where r is the distance from the longitudinal axis to the engine bedplate.

$$|Z| = \frac{M_{yy} - I_{ra}\frac{d\omega}{dt}}{2r^2\omega} \tag{1}$$

The magnitude of the impedance was calculated to be $9.9 \cdot 10^6$ Ns/m as a total for one side of the foundation.

The impedance of a base excited spring-mass system is

$$Z = \frac{\frac{ck^2}{\omega^2} - jkm\left[\omega m - \frac{k}{\omega} + \frac{c^2 k}{\omega m}\right]}{c^2 + \omega m - \frac{k}{\omega}} \tag{2}$$

Which, at the resonance frequency, when c is small and neglecting small value terms, reduces to:

$$|Z| = \frac{mk}{c} \tag{3}$$

or in a more convenient way

$$|Z| = \frac{2\pi f m}{\eta} \tag{4}$$

m: mass
c: damping coefficent
k: stiffnes
η: loss factor

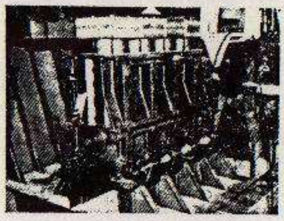
In order to obtain a reduction of the vibration level of 6 dB, the impedance of the absorber should be equal to the impedance of the bedplate. This means that the loss factor has to be very small in order to minimize the necessary mass and volume of the absorber. The lowest obtainable loss factor is app. 0.0006, corresponding to the loss factor of steel. There will be unavoidable losses in clamping of the spring, so consequently a conservative value of η= 0.01 was used in the calculations. This means that the mass of the absorber has to be 500 kg with one absorber on each side of the engine.

DESIGN OF ABSORBER

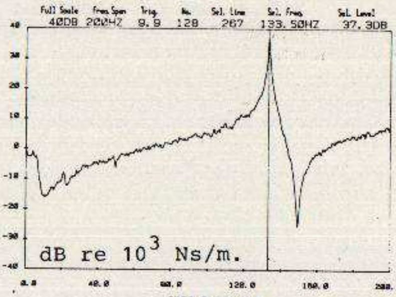
As the force from the absorber has to be distributed along the engine bedplate, the necessary mass was divided into 5 smaller absorbers of 100 kg each. As the absorbers change the input impedance of the engine bedplate, the resulting direction of movement is unknown. The designed absorber should therefore be able to adjust itself to the actual direction of the movement. A rotary symmetric design was therefore applied. This design is favourable, as it is able to adjust itself to the direction of the movement automatically as the resonance frequency is independent of the direction of the movement in the vertical plane.

SCALE MODEL EXPERIMENT

To verify the calculations a scale model experiment was set up. Four dampers were installed, two on each side of a model engine, which was mounted on a foundation type similar to the actual one, on a 1:5 scale model of a ship bottom structure. The engine was excited with a transverse force, simulating the guide moment, using bandlimited white noise to drive the exciter. The model ship bottom structure was kindly provided by The Acoustic Laboratory, at the Technical University of Denmark.



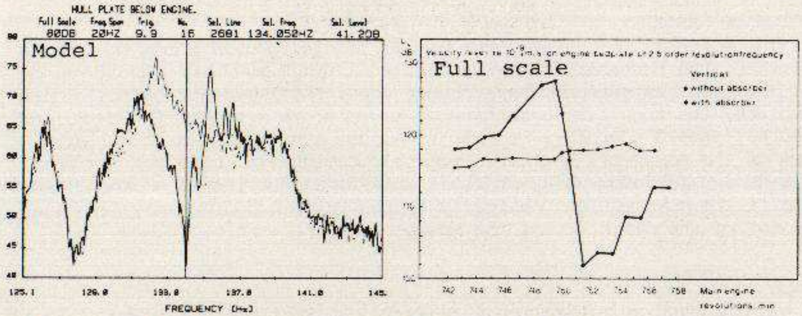
The model experiment set-up.



Measured impedance of absorber model.

The absorber model was designed to work at approximately 132 Hz. The total moving mass was 1.5 kg which, with a loss factor of 0.01, gives an impedance of app. $125 \cdot 10^3$ Ns/m. The measured impedance was $73 \cdot 10^3$ Ns/m. This means that the losses at the clamping of the beam were higher than expected and equal to a loss factor of 0.017. The insertion loss of the absorbers was measured at several positions on the plating and bottom of the ship model. The mean measured insertion loss was 26 dB, at the optimal frequency.

The simplified model for determining the impedance of the ship foundation was applied to the model and the calculated values were within 1-3 dB of the measured values.



Velocity level with absorbers (thick curve) and without (thin curve).

FULL SCALE PROTOTYPE DYNAMIC ABSORBER

A full scale dynamic absorber was designed and manufactured with special care considering low losses in the clamping of the spring. The impedance was measured to be $5.6 \cdot 10^6$ Ns/m corresponding to a total loss factor of 0.0035. The insertion loss of the absorber, mounted on the engine bedplate, was measured on board the ship as a function of engine speed. On the basis of the calculated impedance of the bedplate and the measured impedance of the absorber, the mean insertion loss was predicted to be 11.6 dB. The maximum insertion loss was measured to be 15.5 dB in the vertical direction and 7.5 dB in the transverse direction. Considering the phase relations, the mean insertion loss is 11.5 dB. The narrow working range of a dynamic absorber places restrictions on engine speed variations. This is true especially for a type with little damping. From measurements of the variations in engine speed the obtainable reduction of the velocity level was deduced to be 6 dB in 68% of the measuring time.

CONCLUDING REMARKS

The simplified model for calculating the impedance of the engine foundation gives fairly accurate results which can be used as the calculation basis for a dynamic absorber. The self-adjusting rotary symmetric dynamic absorbers seem to be applicable where space restrictions demand for a compact design, however, structural fatigue problems may occur due to the high loading of the spring.

REFERENCES

C.M.Harris et al., "Shock and vibration handbook, second edition".
J.C. Snowdon et al., "Beamlike dynamic vibration absorbers", Acustica, pp 98-108, Vol. 44, 1980.
M.Ohlrich, "Proc. International conference on recent advances in structural dynamics", pp 225-234, Southampton, July 1980.