STRUCTURE-BORNE SOUND POWER TRANSMISSION THROUGH A POINT CONNECTION TO A FRAME STRUCTURE

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ABSTRACT

This paper discusses the basic theory of structure-borne sound source description and energy transmission through a point connection. Source description of an agricultural tractor diesel engine is determined with the help of vibration measurements. Cast iron frame is used as a receiver, which is connected to the engine with a 2,5 meter long steel beam. The vibration behaviour of the frame structure is measured. Excitation forces in the connection point of the engine and the beam are defined from measured data. The beam and frame structures are analysed with FEM and SEA models. Modelling results are compared with the measured data.

1 INTRODUCTION

In agricultural tractor applications, the diesel engine is typically rigidly connected to the tractor frame structure. Therefore the characterization of the engine as a structure-borne sound source and transmission of structure-borne sound to the frame are essential in improving the Noise, Vibration, and Harshness (NVH) properties of a tractor.

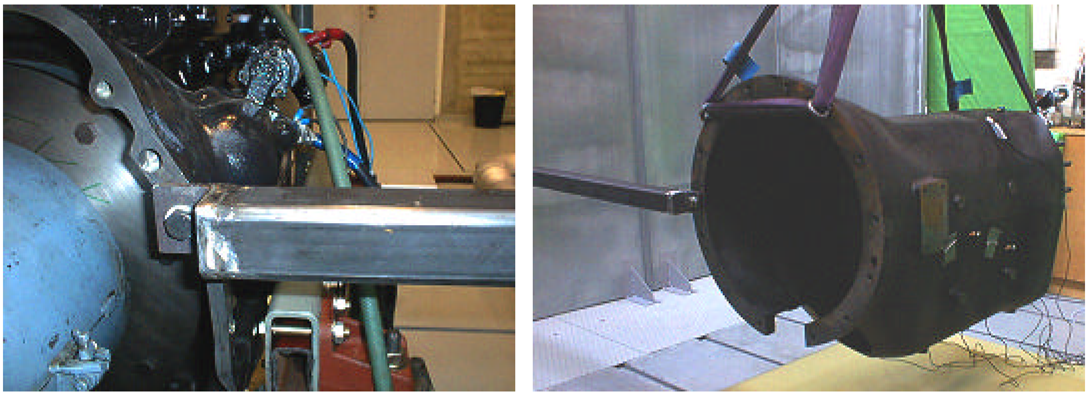

Figure 1. Bolt connection of beam to engine flywheel housing (left), and to tractor frame (right).

The cast iron frame of a Valtra tractor connected with a 2,5 meter long steel beam to a Sisudiesel engine flywheel housing is presented in Figure 1. This simplified point connection was used in this preliminary study.

2 BASIC THEORY

2.1 Characterization of Structure-Borne Sound Sources

Characterization of machines as sources of structure-borne sound is the major problem when estimating the structure-borne sound emission [1]. Characterization is needed when designers, product developers, vendors or purchasers need quantitative data about the vibration or sound levels of machines. This data is used when one source is compared to another, sources with set limits are compared, sound levels of installed machines are predicted or improvement of new low noise designs are quantified [2]. The source characterization must be a property of the source, independent of the receiving structure of the installed machine. It must describe the source's ability to generate structure-borne sound, be an independent property of the source, be able to be expressed as a single quantity, and give the data for the calculation of power generated when the machine is installed [2].

Since the paper [1] in 1987 progress has been made both in theory and measurement techniques. The development has produced a standardized method for the measurement of the vibration velocity of resiliently mounted machines [3]. In practice this velocity can be considered as their free velocity. It is independent and describes the source activity. If both translational and rotational velocities play a role in sound transmission then it cannot be given as a single quantity because of dimensional incompatibility [2]. The most important factor is that free velocity is not sufficient for the calculation of transmitted power, in addition mechanical mobilities of both source and receiver structures are needed [2]. The measurement of blocked forces is in practice difficult. That is why blocked forces are not typically used in the source characterization. The characterization of resilient elements with measured data can be made using standards [4]. This data is needed when estimating the power transmitted through these elements.

Various methods to characterize sources of structure-borne sound are reviewed in [2]. The source descriptor presented in [5] meets all criteria, but it does not work with multiple-point connected structures [2]. Three source descriptors for structure-borne sound sources, the characteristic power (CP), mirror power and maximum available power (MAP) are introduced in [2]. In a single contact point between the source and the receiving structure the complex power transmitted to the receiving structure is given by [2], [6]

$$\overline{Q} = \left|\overline{v}_{\mathrm{sf}}\right|^{2} \cdot \overline{Y}_{\mathrm{r}} / \left|\overline{Y}_{\mathrm{r}} + \overline{Y}_{\mathrm{s}}\right|^{2}, \tag{1}$$

where $\overline{v}_{\mathrm{sf}}$ is the rms free velocity, $\overline{Y}_{\mathrm{r}}$ and $\overline{Y}_{\mathrm{s}}$ are the receiver and source mobility respectively. Using the complex ratio of the receiver to source mobility $\overline{a} = \overline{Y}_{\mathrm{r}} / \overline{Y}_{\mathrm{s}}$ Eq. (1) can be written in a dimensionless form [2]

$$\overline{C}_{\mathrm{c}} = \overline{Q} / \left|\overline{S}_{\mathrm{c}}\right| = \left|\overline{a}\right| / (\left|1 + \overline{a}\right|^{2} e^{jq_{\mathrm{r}}}), \tag{2}$$

where $\overline{S}_{\mathrm{c}} = \left|\overline{v}_{\mathrm{sf}}\right|^{2} / \overline{Y}_{\mathrm{s}}^{*}$, q_{r} is the phase of the receiver mobility, and $\overline{C}_{\mathrm{c}}$ is the so-called coupling factor. From Eq. (1) the maximum complex power is obtained when the mobilities of the source and the receiver are complex conjugate ($\overline{Y}_{\mathrm{r}} = \overline{Y}_{\mathrm{s}}^{*}$) [2]

$$\overline{Q}(\overline{Y}_{\mathrm{r}} = \overline{Y}_{\mathrm{s}}^{*}) = \overline{S}_{\mathrm{a}} = \left|\overline{v}_{\mathrm{sf}}\right|^{2} \cdot \overline{Y}_{\mathrm{s}}^{*} / \left[2\mathrm{Re}(\overline{Y}_{\mathrm{s}})\right]^{2}. \tag{3}$$

The maximum available power (MAP) that is transmitted from source to receiver is obtained from [2]

$$\mathrm{Re}\left[\overline{Q}(\overline{Y}_{\mathrm{r}} = \overline{Y}_{\mathrm{s}}^{*})\right] = \mathrm{Re}(\overline{S}_{\mathrm{a}}) = \left|\overline{v}_{\mathrm{sf}}\right|^{2} / 4\mathrm{Re}(\overline{Y}_{\mathrm{s}}). \tag{4}$$

When the source and the receiver mobilities are equal in magnitude and phase ($\overline{Y}_{\mathrm{r}} = \overline{Y}_{\mathrm{s}}$) the so-called "mirror" power is obtained from Eq. (1), [2]

$$\overline{Q}(\overline{Y}_r = \overline{Y}_s) = \overline{S}_m = \left|\overline{v}_{sf}\right|^2 \cdot 1/4(\overline{Y}_s^*)\,. \qquad (5)$$

When the force is equal to the blocked force $(F_{bl} = v_{sf}/Y_s)$ and the velocity is the free velocity, then the characteristic power (CP) is obtained as [2]

$$\overline{Q}(F = F_{bl}, v = v_{sf}) = \overline{S}_c = \left|\overline{v}_{sf}\right|^2 \cdot 1/\overline{Y}_s^*\,. \qquad (6)$$

In [6] the result of Eq. (6) is called the source descriptor. These descriptors are generalized to multiple point contacts in [2].

2.2 Determination of Structure-Borne Sound Power Transmission

It is assumed that there is only force excitation and no moment excitation at the contact point between the structure and the machine. Therefore, the power P transmitted in this contact point to the receiving structure due to a point force is obtained from complex power $\overline{Q}$ [7]

$$P = \mathrm{Re}(\overline{Q}) = \tfrac{1}{2}\mathrm{Re}(\overline{F}\,\overline{v}_r^*)\,, \qquad (7)$$

where $\overline{F}$ is the point force applied on the receiving structure and $\overline{v}_r$ is the vibration velocity at the same contact point. The real part P of the complex power $\overline{Q}$, where the force and the velocity are in phase, is the active power flowing from the source into the receiving structure where it propagates further. The imaginary part, the reactive power, where the force and the velocity are $\pi/2$ out of phase, goes back and forth through the contact point.

Often in practise it is not possible to measure the transmitted power directly in multiple point contacts. The lack of source and receiver data including their mobilities often prevents the prediction of vibration and noise, which the installed machines inject into receiving structures. If also rotational degrees of freedom are included, then noise and vibration prediction in engineering applications is often impossible. In engineering applications, the power calibration method can be used [8]. The power can be estimated using the calibration of the receiving structure or making the calibration when the source machine is already installed [8].

3 MEASUREMENTS

For the validation of the finite element model, resonance frequencies of the tractor frame were determined from measured accelerance functions. Also damping values were determined from these measurement results. The frame was hanging freely from lifting belts during measurements (Fig. 1). Excitation was given by impact hammer equipped with force transducer (Brüel&Kjær 8200; amplifier Brüel&Kjær 2635) and acceleration was measured from 8 points simultaneously (3D-accelerometer Endevco 63B-100 with integrated electronics (IEPE)). Accelerometers were attached to the frame using beeswax, two accelerometers on each side of the frame.

Multi-channel measuring front-end (Hewlett-Packard 3566A/67A) was used to capture signals (sampling frequency 12.8 kHz) and make spectral analyses. Further analyses and calculations were done with LMS CADA-X-software.

Three orthogonal translational driving point accelerances were measured at the flywheel housing of the engine (Fig. 2, left). Similar measurements were made at the engine-end of the connection beam (Fig. 2, right). Both an impact hammer and an electrodynamic shaker (Brüel&Kjær 4808; amplifier Brüel&Kjær 2712) were used for excitation. Random signal, taken from the signal source of the measuring front-end, was used with the shaker. Driving point mobilities used in Eq. (1) were calculated from these measurement results.

The acceleration of the engine flywheel housing at the connecting point of the beam was measured when the engine was running at full load at maximum speed (2200 rpm) but disconnected from beam. This data was used in the determination of the free velocities (Eq. 1).

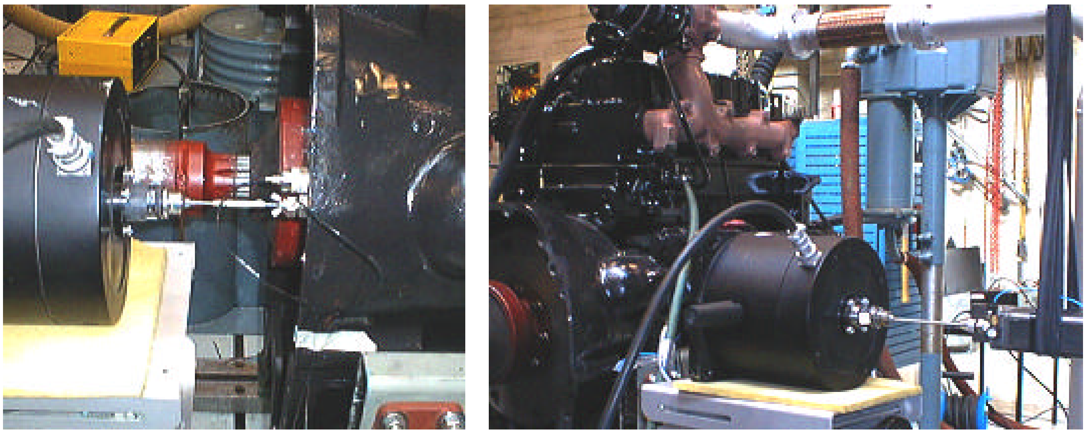

Figure 2. Driving point accelerance measurements at engine flywheel housing (left) and at the engine-end of connection beam (right).

The acceleration of the frame was measured simultaneously at 8 points when the frame was connected to the engine with the beam while the engine was running at full load at maximum speed. The vibration velocities were determined from this data and compared to modelling results.

4 MODELLING

The FEM-model created with I-DEAS -software included two parts: the frame and the beam. The frame structure was modelled with tetrahedral solid elements with middle nodes, altogether 10 nodes per element. The beam was modelled with two-node linear beam elements; cross section was identical with the real beam. Parts were connected rigidly together with two constraint elements. The constraint element included one node in the beam and two nodes in the frame. The entire model consisted of 8 333 elements and 16 978 nodes. The FEM-model was updated with the help of the measurement results.

Forced steady state vibration of the frame was calculated. The excitation force was obtained from the measurement results. The force was exerted into the end of beam at x-, y- and z-directions. The frequency range of the force was up to 1120 Hz, which is the upper frequency limit of the 1 kHz third-octave band. The vibration velocity response was calculated at 8 nodes corresponding to measurement points in the real frame.

At high frequencies a Statistical Energy Analysis (SEA) -model was used in the calculation of the vibration velocities of the tractor frame. This study included one third-octave frequency bands up to 6.3 kHz. The model consisted of three different beam elements and a cylinder element. The original model was updated due to problems at the contact point between the beam and the cylinder. For the modelling of the bolt connection further studies are needed.

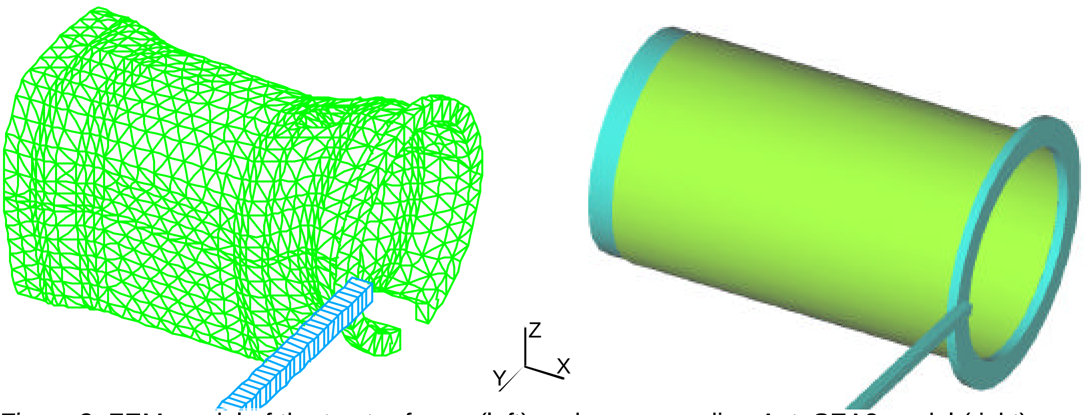

Figure 3. FEM-model of the tractor frame (left) and corresponding AutoSEA2-model (right).

The applicable frequency band of the SEA-model was investigated beforehand with FEM-models. According to the results SEA-model would be applicable above 200 Hz. It was also found out that the modal density of the connecting beam increases rapidly above 2 kHz because of the bending modes of the plate areas of the beam.

5 RESULTS

Translational driving point mobilities derived from measured accelerances at the contact point are presented in Figure 4. This data and free velocities measured from engine flywheel housing were used to calculate the forces exerted to the receiver (Fig. 4, bottom right).

Modelled perpendicular vibration velocities of the tractor frame correspond rather well with measured results (Fig. 5, left). With a verified model, it is easy to test the effects of different noise control measures as well as increase the knowledge about the behaviour of the object.

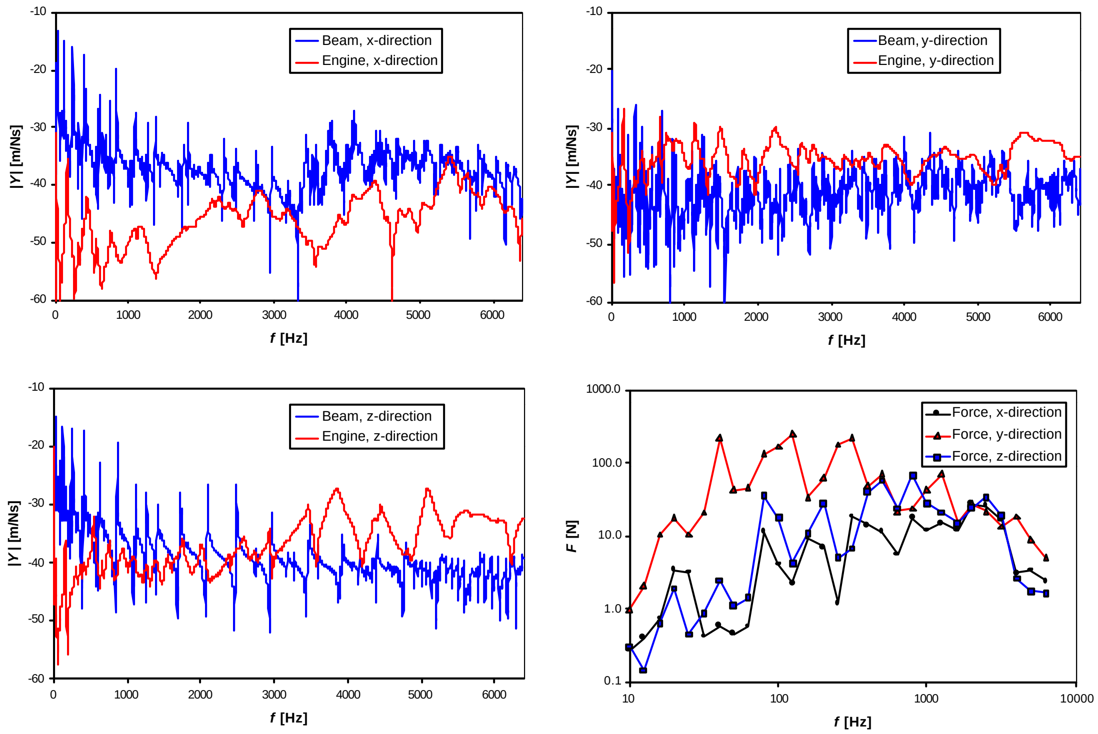

Figure 4. Magnitudes of measured driving point mobilities of the contact point; engine-end of connection beam and engine flywheel housing. 1/3-octave spectrums of forces exerted to the receiver; derived from the measured data (bottom right).

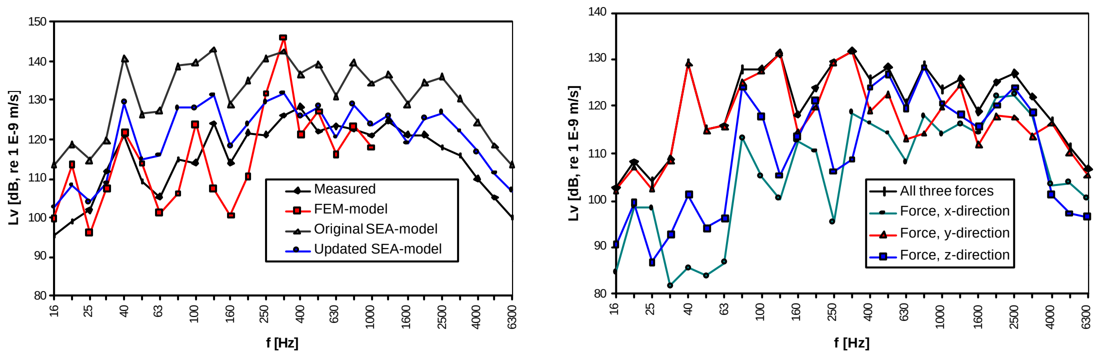

Figure 5. FEM- and SEA-results and measured perpendicular vibration velocities (left). Measured and FEM-results are averages of the 8 points, SEA-result is the average velocity of the cylinder. Perpendicular vibration velocities of different excitation forces, SEA-result (right).

6 SUMMARY

The diesel engine of an agricultural tractor was studied as a structure-borne sound source using laboratory measurements and FEM- and SEA-modelling. The structure-borne sound power transmission through a point connection between the engine flywheel housing and the connecting beam was analysed. The tractor frame was connected to the other end of the beam. The contact point was simplified with three translational degrees of freedom, due to practical limitations concerning measurements.

The results of the FEM-model coincided relatively well with the measured velocities considering that the real frame structure differed quite a bit from the used model. With the SEA-model some modifications were needed. With a verified model it is easy to demonstrate the effects of different noise control measures.

With the results of mobility measurements and free velocity measurements, it is possible to understand the source characteristics and the interaction between source and receiving structures [2], [5] & [6]. Due to problems concerning mobility matrix measurements more research is needed for the determination of allowed simplifications. When modelling is used additional problems are caused from the lack of knowledge about damping data of real structures.

This work continues with the study of surface connection between the diesel engine and the tractor frame. In addition, the source descriptors of structure-borne sound transmission from the engine will be estimated together with structure-borne sound power estimation according to the method presented in [8].

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