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RANKING OF PARTIAL NOISE SOURCES AND NOISE CONTROL MEASURES

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Abstract

Sound radiation of an object often consists of several sources. These partial sources may differ in size, material and sound radiation properties. The basis for successful noise control is the ranking of these sources. Traditionally this task is solved using sound intensity measurement or by excluding separate noise sources using partial enclosures. The same approach can be used to assess the effectiveness of noise control measures. Alternative methods are calculation of surface velocity distribution from the measured sound pressures or approximation of radiated sound power level using the measured velocity distribution. In this paper the feasibility of a rather simple and straightforward method is studied. The surface velocity distribution is measured with a laser vibrometer. The measured data is used as excitation for boundary element calculation to estimate the sound power level of a partial source. The potential limitations and errors of the method are studied using FEM and BEM modeling. At high frequencies the errors turned out to be smaller than the accuracy of standardized sound power measuring methods. At low frequencies the errors are higher but the A-weighting reduces their significance. The method was applied to measure the sound power level of a camshaft cover of a large diesel engine and thereby to assess the efficiency of proposed low noise covers. A comparison with intensity method is made and the supremacy of the studied method is shown.

1. INTRODUCTION

Sound radiation of an object often consists of several sources. These partial sources may differ in size, material and sound radiation properties. There are also different sound generation mechanisms, which have certain typical characteristics. The basis for successful noise control is ranking these sources.

There are several possibilities to do noise source ranking and ranking of noise control measures. A traditional method is selective elimination of separate sources. Almost as widely used is the sound intensity method. A more demanding technique is acoustic holography where the acoustic field further away from the source is measured and numerically transferred near to the source or even converted to the velocity of the vibrating surface. Conversely the measured vibration velocity distribution of a source can be used to calculate the radiated sound power level for different areas of the source.

A large diesel engine (Figure 1) is a complex noise source. There are several excitation mechanisms and the sound radiation of the different parts of the engine varies widely. The sound power of the engine depends on several factors, such as the size of the engine, number of cylinders, engine load and speed, type and load of turbocharger and type of fuel. The source ranking of an engine studied here has indicated that the crankcase and camshaft covers are the dominating noise sources. A variety of cover designs were tested to reduce the noise generation. According to the intensity measurements all tested alternative covers seemed to perform equally which was a bit surprising. The measurements were carried out in a reverberant environment in an engine test cell with highly reflecting walls which may introduce some uncertainty to the results. Therefore a closer study was initiated.

W6L32
- Nominal power: 3000 kW
- Cylinder bore: 320 mm
- Stroke: 400 mm
- Length: 5110 mm
- Width: 2207 mm
- Height: 3703 mm

Figure 1. The in-situ measurements were carried out on Wärtsilä W6L32 diesel engine.

2. NOISE SOURCE RANKING METHODS

A straightforward traditional method for ranking noise sources is selective elimination of separate sources by changing the sound radiation of different parts of the major sources. This can be accomplished for example with wrapping or with other treatments or by disconnecting the parts if possible. One problem with this method is that it is difficult if not impossible to make such treatments that would work equally on the whole frequency range. If a partial enclosure is used it may be prone to structure-borne sound and it also has influence on the sound radiation of the neighboring parts.

The sound intensity measurement method is a powerful tool for noise source ranking compared to sound pressure measurements. When partial noise sources of a complicated structure are ranked in a normal reverberant factory hall, the measurements are typically done in the near field. The phase error between the measurement channels is one limiting factor which defines the dynamic capability index of the measurement system and thereby specifies the highest allowable pressure-intensity indicator [1, 2]. This means that dominant partial noise sources are reliably measured but in quiet areas problems occur. The near field measurement itself is sensitive to errors because the sound intensity field is complicated containing areas of negative intensity. Thereby non-ideal sweeping or poorly chosen measurement point grid causes errors. Therefore the intensity measurement of silent parts of the source easily gives misleading results [3].

A commercially available Near-field Acoustic Holography (NAH) or Statistically Optimal Near-field Acoustic Holography (SONAH) product is Spatial Transformation of Sound Fields (STSF), which is developed and marketed by Brüel & Kjær [4]. A set of stationary reference transducers and a scan microphone array is used to obtain a complete model of the sound field by measuring over a two dimensional region close to a stationary sound source. Using this model to back-propagate to a plane close to the surface of the source allows high resolution source localization.

Near-field Acoustic Holography calculates the pressure, particle velocity, and active and reactive acoustic intensity in the near field region. The scan area dimensions define the minimum analysis frequency and measurement point spacing defines the maximum analysis frequency. With NAH at least 1/3 wave length scan area size is needed, while SONAH works well with 1/10 wavelength scan area size. The measuring point spacing must be less than ½ wave length. STSF may be applied also to analyze acoustical noise sources.

The velocity distribution of a sound source can be measured using accelerometers or a scanning laser vibrometer. In practice the latter is a more realistic choice if good spatial resolution is needed. The velocity distribution measurement i.e. operational mode shape in itself may in some cases give valuable information of the source.

When the velocity (v) distribution of the sound source is measured then acoustical boundary element method can be used to estimate the radiated sound power (P) based on

$$P = \sigma \rho c A \left\langle v^2 \right\rangle \tag{1}$$

where σ is radiation ratio, ρ is fluid density, c is speed sound in fluid, A is area of the vibrating surface and $\langle v^2 \rangle$ is perpendicular mean square velocity of the surface.

The frequency range of BEM results is affected by the resolution of the measurement grid. As a rule of thumb the maximum measurement point distance may not exceed the bending wave length divided by 6. The wave length being the one corresponding to the desired upper limiting frequency. Another limit is introduced by the calculation time. With a standard 32 bit PC the model size is in practice limited to 10 000 degrees of freedom which in this case is the number of velocity measurement points. As an example if the upper limiting frequency is 5 kHz this means that the measurement area can not be larger than 1.3 m^2. In some cases, with thick structures, the fact that the radiation factor is 1 well above the coincidence frequency lowers the needed spatial resolution. This means that either larger sources or higher frequencies can be analyzed.

Velocity distribution measurement together with BE-calculations was chosen to be the tested method due to the nature of the problem concerning the ranking of the alternative noise control measures of the covers of the diesel engine.

3. VERIFICATION OF THE USED METHOD

In vibro-acoustics modeling noise generation is typically carried out using FEM calculated perpendicular velocity distribution of the source surface as a boundary condition for the sound radiation calculation using BEM [5]. With an existing source the calculated velocity distribution can be replaced with measured data. This approach is seldom referred to in literature as a means of estimating the sound power of partial sources. Visser [6] used the laser vibrometer in order to obtain a boundary condition for the BEM computation to verify a developed BEM solver. Di Sante et.al. [7] analyzed the acoustic power generation of a belt drive by using vibration data obtained with a scanning laser vibrometer as inputs for a boundary element code.

3.1 Baffled steel plate

For testing the method a baffled steel plate was analyzed. A 2.5 mm thick steel plate was attached to the wall of a semi-anechoic laboratory (Figure 2). The plate was separately excited with an electro-dynamic exciter and with a diffuse sound field. Scanning laser vibrometer was used to measure the velocity distribution of the plate. The measured data was used as a boundary condition for BEM to calculate the radiated sound power. The sound power level of the steel plate was also measured using sound intensity method. A good correlation was detected between the estimated sound power spectra.

With experiments simulating machinery noise generation it is essential to note that the type of excitation has a significant effect on the sound radiation. This is illustrated in Figure 3. In this test example with point force and diffuse sound field excitations a noticeable difference between the radiation index spectra is observed.

Figure 2. The test set-up in the laboratory.

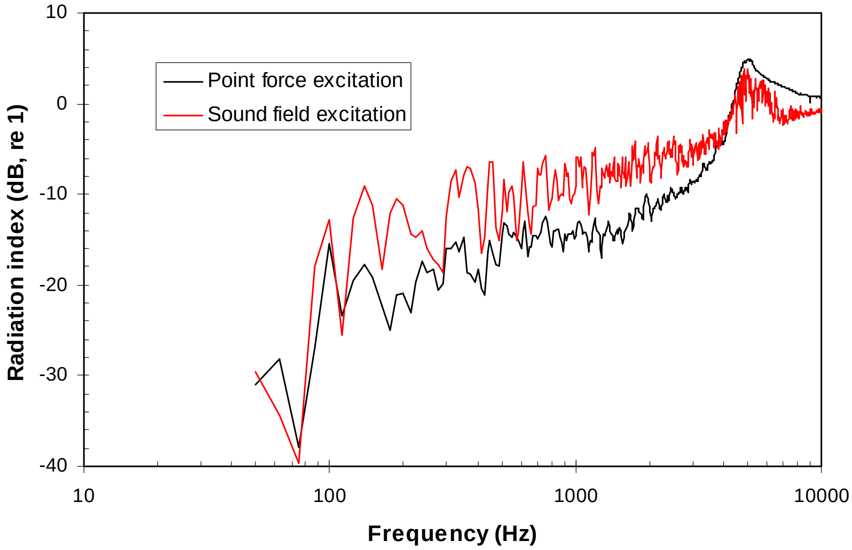

Figure 3. Radiation index spectra of different excitation mechanisms.

3.2 Numerical simulations

Numerical simulations were used to simulate the proposed sound source ranking method with a baffled steel plate. In the first stage a 10 mm thick homogeneous steel plate (1 m × 1 m) with a point force excitation was modeled. The first variation was to change the thickness of the middle part of the plate to 2 mm (0.4 m × 0.5 m). Second variation was to change the excitation to a multipoint phase shifted point force excitation to achieve more diffuse excitation mechanism. In all the three cases the velocity distribution of the plate was calculated using FE-software. The baffled direct BEM was used to calculate the radiated sound power of the whole plate, the middle part of the plate and the edges of the plate. The radiated sound power level of the whole plate was compared to the summed sound power level of the other two parts. The analysis results are presented in Figure 4. Level difference is defined with the total plate result as a reference.

The homogeneous plate results are really good at frequencies above 800 Hz. An overestimate of the combined sound power between 300 Hz and 750 Hz is caused due to the fact that acoustic interaction between the plate areas vibrating out of phase is disturbed. Probably similar effects occur also with the two other variations.

A more accurate way to verify the results is to calculate the center part sound power from the total model and compare the result with the separate center part sound power. This result is presented in Figure 5 for the non-homogeneous case with diffuse like excitation.

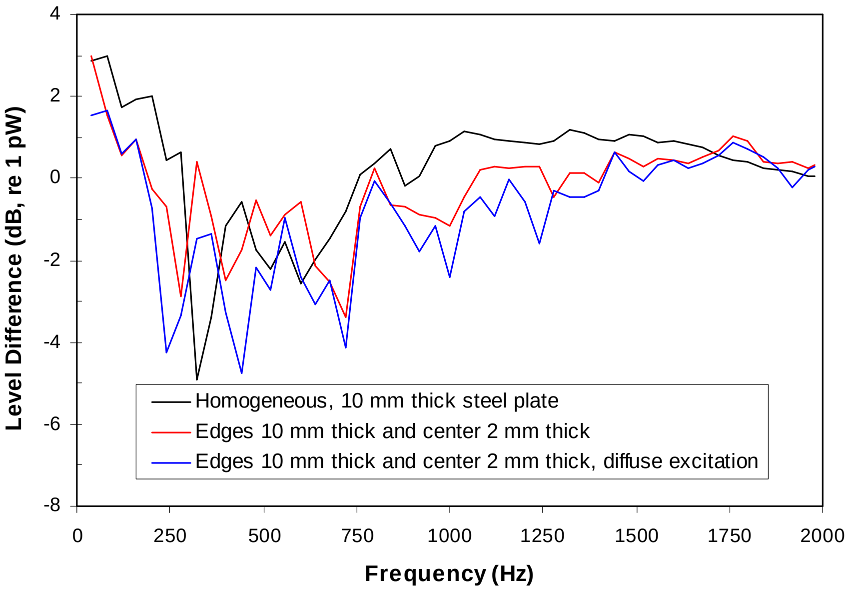

Figure 4. Deviation of the sound power levels calculated in parts from the result calculated as a whole.

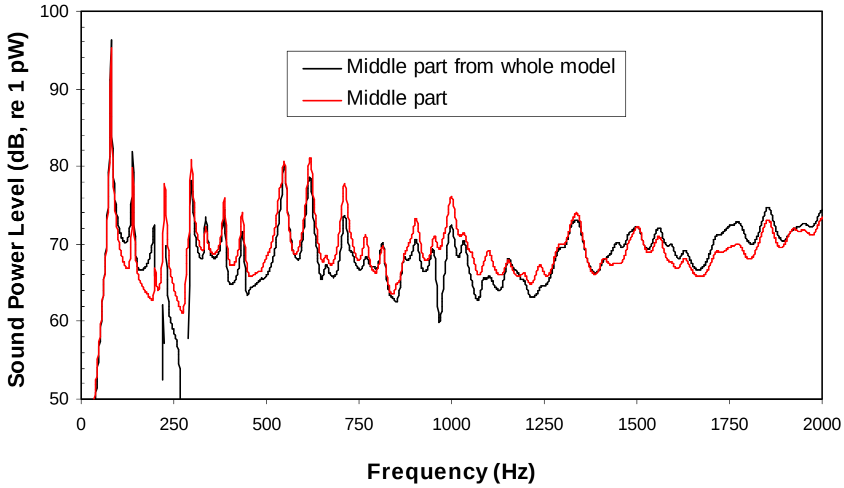

Figure 5. Sound power level spectra; separate middle part and middle part from the total model.

4. CASE STUDY, COVERS OF A DIESEL ENGINE

Improving of crankcase and camshaft cover design of a large diesel engine was a part of an engine noise reduction project. Several prototype covers with different vibro-acoustic properties were tested in laboratory conditions and in a full-scale engine test. In the final measurements all covers on the maneuvering side were changed to redesigned covers.

4.1 Measurements and modeling

The covers were tested on a Wärtsilä W6L32 diesel engine. The nominal speed of the engine is 750 rpm and the output power is 3000 kW. All measurements were carried out at full load at nominal speed. Engine was run in a manufactory test cell.

Three types of measurements were carried out during the engine test: sound pressure, sound power and vibration velocity distribution of the covers. Sound pressure level was measured to monitor the effect of different covers to total sound pressure level in the engine room.

Sound power radiated from the side of the engine (Figure 6) was measured using sound intensity method. The measured area was divided into eight sub-surfaces that were scanned with the intensity probe for each cover configuration. The effect of different covers can be seen directly by studying the sound power of the maneuvering side of the engine, since it is dominated by the noise from the covers. However it is difficult to draw conclusions for ranking the alternative prototypes.

Figure 6. Measurement object equipped with prototype covers.

Velocity distribution of one specimen of each tested type of the covers was measured using scanning laser vibrometer. Measured velocity distribution was used in BEM-calculation as excitation to calculate the radiated sound power level. Calculation times were around some hours with a standard PC.

4.2 Results

The results of two alternative methods are presented in Table 1. Based on the intensity measurement results the reduction of sound power level (L_{WA}) of all covers was in range from

7.2 to 9.7 dB whereas the BEM results vary from 16.2 to 26.1 dB. The intensity results are partly influenced by the fact that the scanned area included also small parts of the engine block. According to the limitations of the intensity method and previously presented modeling examples the BEM results are claimed to be more correct.

A-weighted sound power level spectra of the standard crankcase cover and a prototype cover is presented in Figure 7. The results are determined using laser measurements and BEM calculations.

The BEM results confirm that the vibro-acoustic properties of all the prototype covers are far better than the current structure. Therefore a new cover type may be selected based on material and manufacturing costs.

Table 1. Sound intensity measurement results compared to the combined results of laser measurements and BEM calculations.

Structure	Sound power level determined using sound intensity measurements L_{WA} (dB)	Sound power level determined using laser measurements and BEM calculations L_{WA} (dB)
Standard cover	0	0
A: Steel cover	-7.2	-16.2
B: Sandwich cover[1)]	-8.2	-25.9
C: Sandwich cover	-7.9	-20.4
D: Enclosure	-9.7	-26.1

[1)] Viscoelastic damping material optimized to operating temperature.

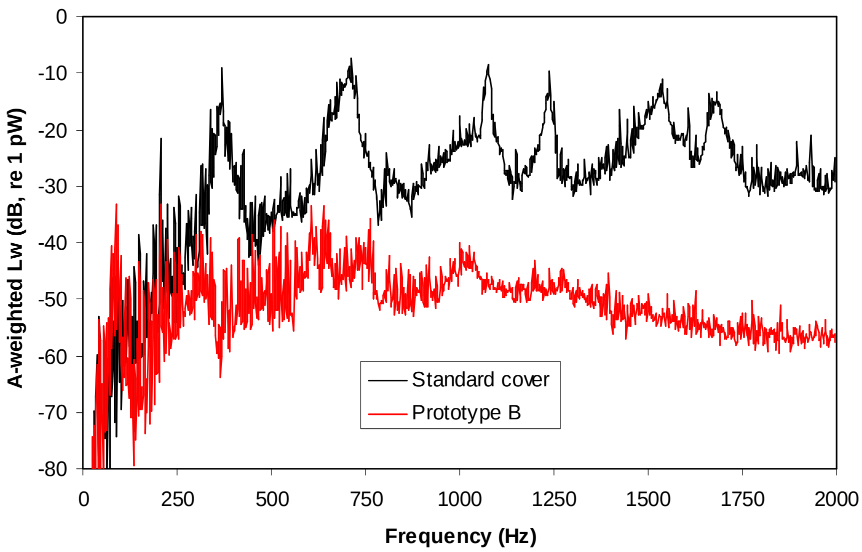

Figure 7. A-weighted sound power level spectra of standard cover compared to prototype B.

5. SUMMARY

Ranking of partial noise sources and ranking of noise control measures are basic tools for successful noise control. A straightforward approach is to use sound intensity measurement.

With a diesel engine crankcase cover prototypes intensity measurement gave doubtful results. The measured insertion loss was about equal for each prototype. Therefore another ranking method was utilized.

The modeling of the noise radiation can be carried out using measured velocity distribution of the source surface as a boundary condition for the sound radiation calculation using BEM. This method was verified with a simple test case and numerical simulations. The errors proved to be within ± 3 dB at the principal frequency range.

In the final stage improvement of crankcase and camshaft cover design of a large diesel engine was studied using four prototype covers. According to the sound intensity measurements the prototypes were 7 to 10 dB quieter than the original cover. The alternative method showed that correct values would be from 16 to 26 dB. The accuracy of the method could be improved by examining a larger section of the radiating surface but the measurement and calculation time together with highest studied frequency puts a limit to this.

The usability of the method would be more evident if the prototype covers would have been only slightly better than the original structure. Especially if only a single cover could be changed the effect of surrounding surfaces would corrupt the validity of the intensity measurement for ranking the alternative prototypes.

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