Use of a new acoustical source descriptor for designing quieter heavy load vehicles

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ABSTRACT

Noise from the engine and the transmission is dominant in the ISO pass-by tests of current heavy road vehicles. In the Brite-Euram project PIANO several new methods have been developed to enable truck manufacturers to speed up the development phase of quieter trucks and to increase the cost-effectiveness of noise reduction measures.

This paper describes a new method for the acoustical source strength characterization of the engine and of the transmission. The acoustical source strength of each relevant part of the engine is modelled in terms of equivalent acoustical monopoles. An important practical feature is that this source strength is independent of the acoustical environment. Therefore, it can be determined on a test rig, but it remains relevant when the engine is installed in the vehicle. The other practical advantage is that the use of monopoles as equivalent sources facilitates the transmission path quantification. Transfer functions between the various parts of the engine or transmission and the pass-by noise microphones are measured reciprocally.

It is shown that this analysis method forms a very practical and robust method for accurate ranking of the contributions to the pass-by noise by different parts of an engine and for a quick evaluation of enclosure variants. After a brief description of the method and the discussion of some validation tests in the laboratory, results are shown of the application of the method for the determination of the acoustical insertion loss of the engine enclosure of a truck engine.

1. INTRODUCTION

The reduction of the pass-by noise of heavy trucks as measured in ISO tests, requires great efforts for a further reduction of the noise contribution from the engine and the transmission. The sound at an observation point is determined by the source strength and by the attenuation along the transmission path. The former is a property of the engine or transmission, whereas the latter is determined by the vehicle, including the possible presence of an enclosure. The vehicle manufacturers need to be able to distinguish the influence of these two factors on the sound at the receiver point. Therefore it is necessary to have methods that characterize the sources and the transmission paths separately.

Conventional source strength descriptors for airborne sound are sound pressure levels at a given distance from a machine and sound power levels. However, such descriptors are not usable for the above mentioned diagnostic analysis of sources and transmission paths. They will be very different in different acoustical environments, e.g. on an engine test rig and in the vehicle inside an engine enclosure. Moreover, it is hardly possible to define transfer functions which can characterize the transmission system properly and which also comply with these source descriptors.

Therefore within the Brite-Euram project PIANO, a new source descriptor has been defined which makes the diagnostic analysis of the partial source and transmission path contributions more practical.

The basic principle of the new method is that the sound radiation from an engine or from a part of it, like an oil sump, is defined in terms of fictitious elementary sources [1, 2]. These are assumed to be uncorrelated monopoles located on the radiating surfaces, see fig. 1a. The strength of these equivalent substitution sources can be determined indirectly, as will be discussed later.

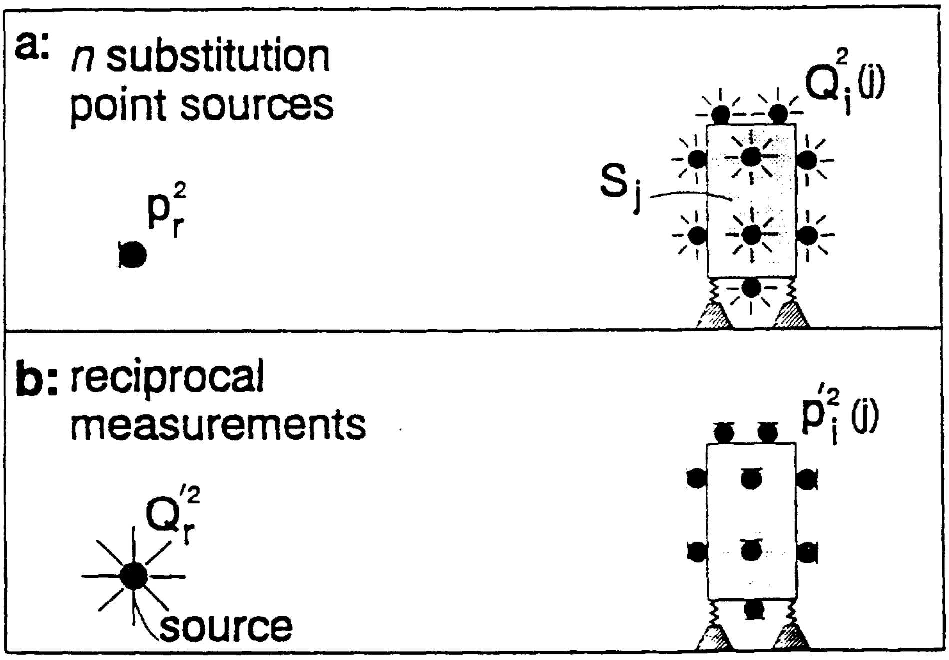


Figure 1 Method of uncorrelated monopoles.
a) subdivision into m partial source areas.
b) reciprocal measurement of transfer functions.

The basic equation for diagnostic system analysis is as follows

$$p_r^2 = \sum_{j=1}^{m} p_r^2(j) = \sum_{j=1}^{m} T_{j,r} \times Q_{eq}^2(j) \tag{1}$$

In eq. (1) the mean square sound pressure at a receiver position is written as the sum of m uncorrelated contributions. For each of the m partial radiators the equivalent source strength is given in terms of a mean square value of the volume velocity, i.e. $Q_{eq}^2(j)$. The transfer function from that partial source to the receiver position is $T_{j,r}$.

The idea when applying this method for trucks, is that the equivalent source strengths of parts of the engine and of a gear transmission can be determined on an engine test rig. The transfer functions can be determined on a vehicle with the engine not in operation. Application of the reciprocity principle makes these transfer function measurements practicable, see figure 1b.

2. DETERMINATION OF SOURCE STRENGTHS

For the determination of the equivalent source strengths, two methods have been investigated. One method uses sound intensity measurements close to the engine on a test rig. This method is especially suited for steady running sources. The other method uses a local enclosure, which is temporarily attached around part of the engine. This method is more suited than the intensity method, for sources under rapidly changing conditions, like a fast run up.

2.1 Source strength definition

The first step is to divide the radiating surface of the source into m sub-areas, which are considered as partial sources, see fig. 1a. On truck engines such parts can be, for example, valve and distribution covers, oil sumps, as well as different side of the engine block. The assumption is made that on each sub-area the sound radiation of the structure may be replaced by that of $n(j)$ uncorrelated monopole sources each with the same strength $Q_{eq,i}^2(j)$. The acoustical source strength of each sub-area is defined as the summed mean squared volume velocities of the $n(j)$ monopole sources, i.e.

$$Q_{eq}^2(j) = n(j) \times Q_{eq,i}^2(j) \tag{2}$$

2.2 Intensity method

From each of the sound radiating sub-areas of the engine, the radiated sound power can be measured on a test rig, using sound intensities just in front of the engine surface [1]. Now the equivalent mean squared volume velocity is found from equating the measured sound power with the estimated power radiated by the uncorrelated monopoles. It is assumed that for most of the fictitious point sources, the radiation resistance equals that for a monopole source against an acoustically hard baffle and radiating into a half-space [1]. Then it may be proven that

$$Q_{eq}^2(j) = n(j) \times Q_{eq,i}^2(j) \approx P(j)\frac{2\pi c}{\rho\omega^2} \tag{3}$$

where P denotes radiated power, c the speed of sound in air, ρ the density of air and ω the radian frequency.

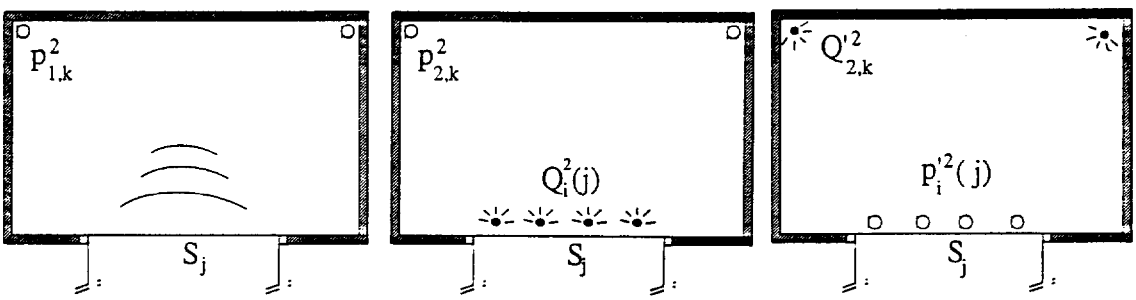
2.3 Local enclosure method

In case of rapidly changing sources, like vehicle engines on a test rig under fast run up conditions, sound intensity measurements become impractical. Because of the unsteady signal a scanning measurement is not possible. Measurements at a large number of discrete points would necessitate a large number repeatable run ups. In the local enclosure method rapidly changing signals can be handled much more easily. This method is illustrated in figure 2.

The partial source $S(j)$ under consideration, radiates sound into a temporarily attached local enclosure, the interior of which is effectively shielded from the other parts of the engine. The method consists of three steps. First, when the engine is running, the mean squared sound pressure $p_{1,k}^2$ inside the enclosure is measured at q different positions, see fig. 2a. Next it is assumed that the real source can be replaced by $n(j)$ uncorrelated monopoles, see fig. 2b. The transfer functions between monopoles and microphones may be written as

$$H_{i,k}^2(j) = \frac{p_{2,k}^2}{Q_i^2(j)} \tag{4}$$

where for the sound pressure the index number 2 is used to distinguish from measurements with the engine in operation (as in fig. 2a). Again it turns out to be practical to measure these transfer functions reciprocally (see fig. 2c). From each of the q microphone responses in figure 2a a raw estimation of the partial volume velocity may be obtained. However, a more smoothed estimation follows from averaging over all q available



a. sound pressure measurements with engine running
b. $n(j)$ uncorrelated monopoles
c. reciprocal measurement of enclosure transfer functions

Figure 2 Principle for determining the equivalent volume velocity using transfer functions within a locally attached enclosure.

microphone responses in figure 2a as follows:

$$\hat{Q}_{eq}^2(j) = \frac{n(j)}{q} \sum_{k=1}^{q} \left\{ p_{1,k}^2 \left(\sum_{i=1}^{n(j)} H_{i,k}^2(j) \right)^{-1} \right\} \tag{5}$$

Of course, estimation results of the source strength according this method and according to the sound intensity method (eq. (3)) ought to be consistent.

3. DETERMINATION OF TRANSFER FUNCTIONS

Following eq. (1), the transfer function between a sub-area of a partial source and a pass-by noise microphone is denoted by $T_{j,r}$. This transfer function is defined as the average transfer function over the $n(j)$ positions of the fictitious substitution monopoles on a sub-area $S(j)$ of the source. Invoking the well-known reciprocity relation in acoustics for monopole sources and receivers, the transfer function may be written as follows:

$$T_{j,r} = \frac{1}{n(j)} \sum_{i=1}^{n(j)} \left[\frac{p_r^2}{Q_i^2(j)} \right] = \frac{1}{n(j)} \sum_{i=1}^{n(j)} \left[\frac{p_i'^2(j)}{Q_i'^2} \right] \tag{6}$$

The right-hand side of eq. (6) represents the reciprocally measured transfer functions, see fig. 1b. $Q_i'^2$ is the source strength of the source at the 'receiver' position and $p_i'^2(j)$ is the squared sound pressure at the engine surface.

4. VALIDATION BY SIMULATIONS AND BY LABORATORY EXPERIMENTS

Prior to the PIANO project, the equivalent volume velocity of uncorrelated monopoles, as a source descriptor, was applied in experiments on a shipboard diesel engine [3]. The diagnostic purpose of this analysis was to determine the relative contribution of airborne sound transmission to the underwater sound radiated by the diesel engine. Moreover, detailed information was obtained on the relative importance of the various partial sources on the machine surface and of the corresponding transfer functions. Several tests were performed which resulted in the conclusion that this reciprocity method for path analysis is practically feasible and powerful.

The same conclusion was drawn from laboratory measurements on an engine simulator [1] and on a combustion engine [4].

However, the above mentioned experiments were with the engine or engine simulator installed in a relatively large space. With the application for truck engines in mind, the question had to be studied, whether or not the equivalent volume velocity as defined by eq. (3) remains the same when the engine is installed in completely different environments. In particular this should be the case on the engine test rig, where the source strength is determined and in the vehicle, e.g. inside a rather tightly fitting enclosure. In the PIANO project this question was studied at TNO (Delft, The Netherlands) with the aid of mathematical experiments and laboratory experiments.

4.1 Mathematical experiments

In the mathematical experiments sound radiation of baffled plates into a half-space and into a shallow cavity was studied. Some results can be found in ref. [2]. Basically the following hypothesis was tested:

$$\frac{P_{cav}}{P_{h.s}} \cdot \frac{4h}{\lambda} \approx 1 \quad \text{for } f < \frac{c}{4h} \tag{7}$$

In this formula the sound power radiated by the baffled plate with a given velocity distribution, is compared for a half-space and for a flat cavity of height h, with acoustically hard walls. The cavity height is smaller than a quarter of the sound wavelength. The ratio $4h/\lambda$ equals the ratio of the radiation resistances of a monopole source placed against an acoustically hard wall and radiating into a half-space and of a monopole source in a shallow and flat cavity. Therefore, if eq. (7) is valid for plates, this would imply that their radiation resistances alter in the same way as for fictitious monopole sources on their surface. In that case the source descriptor according eq. (3) would be appropriate for them.

Analytical models were used to test eq. (7) for the modal radiation of many plates. In addition finite element and boundary element calculations have been performed on the forced vibrations of a stiffened plate. Generally speaking in a majority of cases the prediction according to eq. (7) appeared very accurate. Typical results can be found in [2].

4.2 Laboratory experiments

In addition to this theoretical work a lot of practical testing has been performed. In the laboratory an engine simulator was used for this purpose [2]. This is a steel box-like structure (1 m × 0,63 m × 0,74 m) with a wall thickness of 6 mm. It has been stiffened internally with ribs (50 mm × 20 mm) and other beams. Underneath an oil sump of a truck engine has been installed and on top a stiffened steel plate of 2 mm thickness to simulate a valve cover. The sound was generated with an internally installed vibration exciter (1,7 kN).

Only a small selection of results can be shown here.

4.2.1 Consistency of equivalent source strength

In eqs. (3) and (5) two different methods are given for determining the equivalent mean squared volume velocity of a partial source. One is with the intensity method and the other with the aid of a temporarily attached enclosure. The consistency of the results between the two methods was checked with the aid of the engine simulator.

Figure 3 shows equivalent source strengths of the oil sump of the engine simulator, which were determined from three different experiments. During these experiments the engine simulator was driven with a signal which was derived from an acceleration signal measured on a truck diesel engine during a fast run up. In the speed range of 1000 – 2250 rpm, the rate of the speed change was approximately constant and the total time needed for this speed change was 2 s. Because of this transient nature, the equivalent source strength varies with time. The data shown in figure 3 correspond with time averages over 1/16 s at apparent engine speeds of about 1000 and 1625 rpm. The one-third octave analysis of the transient signals was made with a B&K type 2133 analyzer, using exponential averaging in the "fast" mode.

In one experiment the intensity method was applied, see eq. (3). The radiated sound power was determined from sound intensity measurements at 32 points rather close to the oil sump. In the two other experiments the local enclosure method was applied, see eq. (5). The enclosures were rectangular boxes made of chipboard panels and having a small amount of absorption added. The volume of the smaller box was 0,3 m^3 and that of the larger box 1,2 m^3. In both enclosures 4 microphones were placed in corners, as in figure 2a. In the reciprocity experiment microphones were positioned at 22 positions against the oil sump, as in figure 2c. The miniature sound source which was positioned in the box corners for these reciprocal experiments, has been described in [5].

It is seen in figure 3 that for 315 Hz $< f <$ 3150 Hz (the most important frequency range for exterior noise from trucks) there are only minor differences in the source strength estimates. This confirms that the equivalent mean square volume velocity is not just an artificial result of one particular method, but a realistic modelling of a real source strength. Further evidence for this has been reported in [6]. There also, measured and calculated data are reported for the A-weighted sound pressure levels at some distance from the engine simulator, as a function of engine speed. Calculations were made according to eq. (1), using speed varying source strengths, which were determined in three different experiments, like those in figure 3. The predicted and measured sound pressure levels at 2,3 m distance from the engine simulator were equal within 1,5 dB(A) for the apparent speed variation from 1000 – 2250 rpm. Because the sound spectrum of the engine simulator was made equal to that of a genuine truck engine, this close agreement seems a representative result with respect to the intended application.

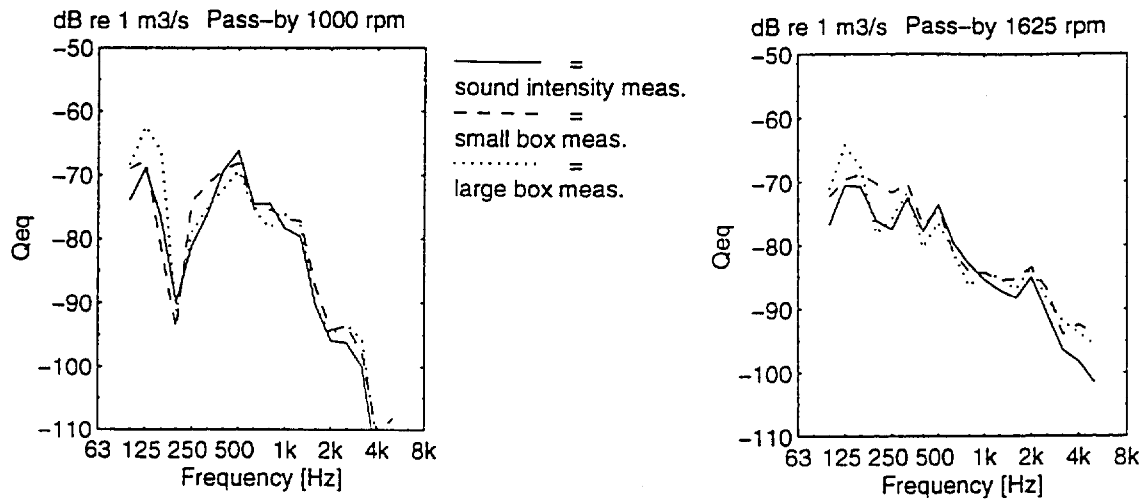


Figure 3 Equivalent volume velocity levels (1/3-octaves) of truck engine oil sump, determined from three different experiments.

4.2.2 Insertion losses of engine enclosures

In another series of laboratory experiments the new source descriptor method, using eqs. (1), (3) and (6), was applied on the engine simulator and with several variants of an engine enclosure. For practical applications in trucks there is great interest in a quick and reliable method for the acoustical evaluation of enclosures and shieldings. A good measure for the performance of the enclosure is the so-called insertion loss or the change of the insertion loss, i.e.

$$IL_p = L_p(\text{without enclosure}) - L_p(\text{with enclosure}) \tag{8}$$

or

$$\Delta(IL_p) = L_p(\text{enclosure 2}) - L_p(\text{enclosure 1}) \tag{9}$$

With the newly proposed method, eq. (1) is used to predict the sound pressure levels at the pass-by microphone, which form the terms on the right-hand sides of eqs. (8) and (9). The equivalent source strengths are considered to be unaffected by the enclosures or shieldings. Therefore, for another enclosure variant only the new transfer functions $T_{j,r}$ need to be measured. Then calculations using eqs. (1) and (8) or (9) give the desired insertion loss values or changes.

Table 1 shows laboratory results for two variants of an enclosure around the engine simulator and for the "bare" engine simulator. The measurements have been performed under steady conditions. One enclosure variant was fully closed and the other only partially; its bottom plate was left out. The sound spectrum of the bare engine simulator was made equal to that of a truck engine.

From Table 1 a comparison can be made between measured results and results calculated with the newly proposed reciprocity method. A very good agreement is seen between the measured and calculated results. This was also the case for other experiments, in which local apertures were made in the enclosure and where additional local sources were attached on the engine surface. In all cases the prediction results based on eq. (1) appeared

Table 1 A-weighted sound pressure levels at a few metres distance from the engine simulator and after conversion to representative diesel engine spectra.

	Side position		Front position	
	measured [dB(A)]	predicted [dB(A)]	measured [dB(A)]	predicted [dB(A)]
L_A without enclosure	116.5	116.5	115.2	115.0
L_A with enclosure	91.0	91.0	90.5	89.8
L_A for enclosure without bottom plate	112.4	110.8	112.0	111.5
insertion loss, enclosure	25.5	25.5	24.7	25.2
insertion loss, enclosure without bottom plate	4.1	5.7	3.2	3.5

rather insensitive to the precise number and location of substitution monopoles. Therefore, at the start of the industrial implementation of the method, there was good reason to expect that the reciprocity experiments combined with the new source descriptor, form a valid and very practical tool in studying shielding and enclosure variations.

5. INDUSTRIAL APPLICATION OF THE METHOD

Within the framework of the PIANO project, two industrial partners were involved in the implementation of the method in an industrial environment. Both CRF (Turin, Italy) [7] and DAF (Eindhoven, The Netherlands) have investigated several important practical aspects. These concern matters as

- the installation and the operational conditions of an engine on a test rig;
- the definition of partial sources and transmission paths;
- the amount of spatial averaging needed over source areas to obtain stable and representative transfer function data.

Good insight was obtained in how engines can be operated in steady conditions on a test bed, whereas source data can be found which is still relevant for fast run up conditions. This research result will facilitate the adoption of the new method considerably, because the intensity based method for source strength determination is simpler than the alternative method with locally attached enclosures.

Main application results:

Equation (1) forms the basis for the practical applications of the method. It is used to distinguish the relative contributions of different parts of the source and the relative strengths of transmission paths from different parts of the engine to the pass-by-noise. The "Q-terms" on the right-hand side of eq. (1) represent the local source strengths of parts of the engine, like for example, a valve cover or a distribution cover. The analysis according to the newly developed procedure reveals, whether or not redesign of such components can have a favourable effect on the pass-by noise of typical vehicle configurations.

On the other hand, if no modifications of the engine as a source are considered, the transmission of the sound can be altered by enclosure and shielding modifications. This can be analysed by studying the "T-terms" on the right-hand side of eq. (1). Special post-processing and presentation procedures were developed, adapted to the specific purposes of diagnostic analysis or vehicle design improvement. Because of the commercial interest of such results the data reported here are very limited. However, the usefulness of the method for the evaluation of insertion losses of enclosures can be easily illustrated.

The insertion loss is the reduction in radiated sound due to the application of a certain enclosure (see 4.2.2). Figure 4 shows the result from an experiment by DAF. Shown are results for pass-by noise microphones on both sides of the vehicle (left- and right-hand side columns). The seven rows are for seven vehicle positions, spaced two metres from each other. Position 4 is opposite the engine (i.e. at the shortest distance). One line in each figure represents the difference in radiated sound, which was directly measured during the run up of two vehicle variants. The other line represents the difference based on two calculations according to eq. (1). For these calculations the source data were used from steady state test bed data and the sets of transfer functions were measured on the "track". The results (especially those on the right-hand side) show the great accuracy of the newly developed

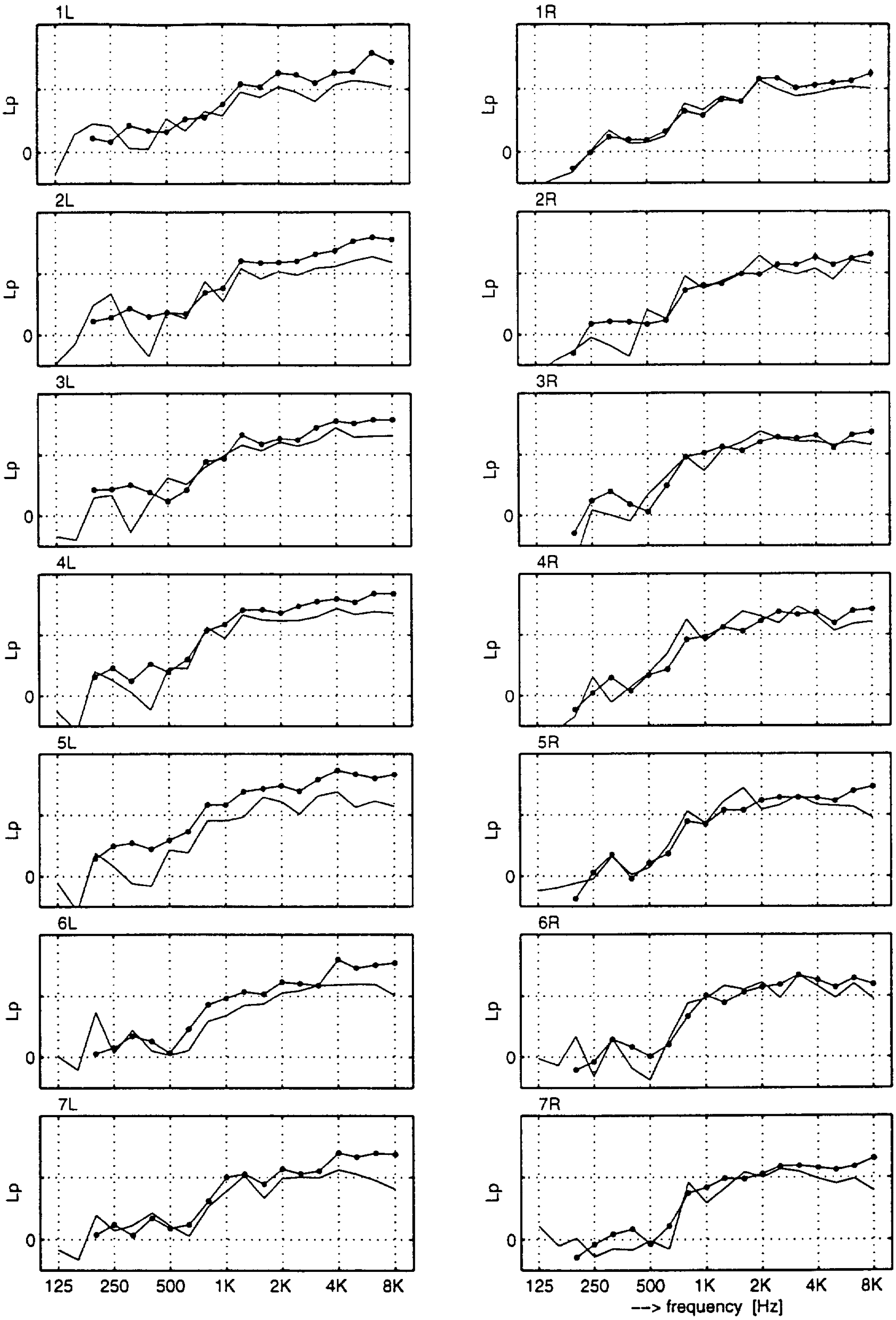


Figure 4: Pass-by noise measurements and calculations by DAF for two truck variants. One without and one with an engine enclosure. Insertion losses (10 dB/div.) are shown for seven microphone positions on both sides of the vehicle. One line shows the measured difference in pass-by noise between the two vehicle variants. The other line (with the dots) shows results calculated with eq. (1) and using steady state source data measured on an engine test rig.

indirect method, in the frequency range which is most relevant for the pass-by noise of trucks.

6. CONCLUSIONS

A new concept has been investigated for the acoustical source strength characterization of truck engines. Compared to more conventional methods the newly proposed method characterizes sources with a quantity which is largely invariant for the installation environment. Moreover, the corresponding transfer functions to the pass-by noise microphones can be easily measured using a reciprocity technique. The clear distinction between source strengths and transfer functions, facilitates the diagnostic analysis of existing products and of prototypes. It also facilitates the development of cost-effective noise control strategies, which are either aimed at affecting the source properties or at affecting the transmission system properties.

Acknowledgement

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