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QUANTIFYING THE NOISE EMISSION OF ENGINE OILSUMPS, VALVE COVERS, ETC. USING ARTIFICIAL EXCITATION

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1. SUMMARY

Oilsumps, valve covers and distribution covers belong to the main noise radiating parts on modern engines. These parts have in common that they undergo velocity excitation at their fixation to the main engine structure.

A method was developed to quantify the noise emission of the individual part using artificial excitation. This measurement technique allows the assessment of the parts and their mounting to the engine for efficient variant comparisons.

The difficulty lies in the simulation of near to rigid boundary conditions of those parts and in the effective velocity excitation. A combination of reciprocal acoustic excitation and inverse force identification is used. The objects are excited by volume velocity sound sources. On the basis of a reciprocity relationship and inverse force identification the effective noise emission is quantified for spatially uncorrelated velocity excitation.

A comparison of aluminium, plastic and rubber decoupled oilsumps is shown as an example.

2. OVERALL PRINCIPLE

The primary excitation is on the main engine structure. This will start to vibrate, also at the interface of the test component. The test components are essentially more flexible and lighter than the main structure, where they are mounted on. The vibration field at the interface is therefore almost entirely determined by the structural dynamic of the main structure. This leads to the following conclusions:

1. The best pure boundary condition simulation of the test object is rigid or clamped.
2. The noise radiation of the test object should be related to the velocity at the interface rather than the interface forces.

To simulate the velocity excitation we simplify the continuous interface vibration field to a set of discrete points at which the velocity is controlled. For a simulation of the true velocity field a half wave length discretisation is needed. But this is impractical; the number of points to consider would become too high.

Both to reduce the number of interface points and to come to a practical excitation simulation, we assume that the velocity excitation can be simplified to uncorrelated motion predescribed at a limited set of points equally spread around the interface.(Ref.1,3) This approximation is reasonable for the broadband engine excitation, analysing in third octave bands, as long as these bands contain several engine modes.

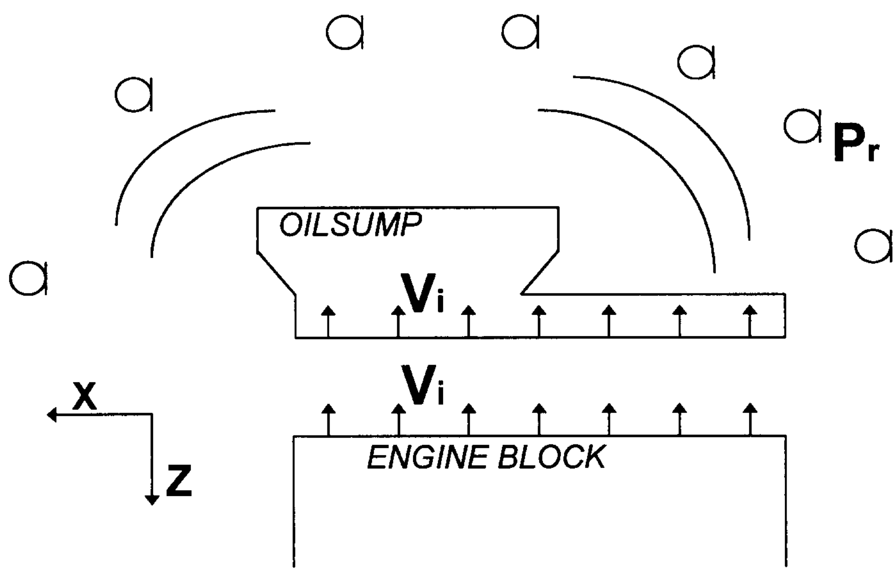

Fig. 1: Sketch of the object and discretisation.

The noise due to velocity at one interface point (in one direction) is quantified in a velocity-acoustic transfer function:

(1) $$H_{ri}^{PV} = \left.\frac{P_r}{V_i}\right|_{Q_{r=0}} \qquad (\mathrm{Pa.s.m^{-1}})$$

The sound pressure at any point in the sound field can then be quantified by:

(2) $$P_r^2 = \sum_i H_{ri}^{PV\,2} . V_i^2 \qquad (\mathrm{Pa}^2)$$

P_r = pressure at point r (Pa)
Q_r = volume velocity in point r (m^3/s)
V_i = velocity in point i (m/s)
x^2 = autopower spectrum or $|x|^2$

To assess test objects, independent of the operational excitation, an equal velocity amplitude $/V_i^2/$ is assumed. And to quantify the noise emission proportional to sound power we consider an even spread of pressure points in semi-free field at one meter distance from the test object.

The interface and hemisphere averaged pressure due to interface averaged velocity excitation becomes:

(3) $$/\,\tilde{H}^{PV}\,/ = \left(\frac{1}{r}\sum_r \frac{1}{i}\sum_i H_{ri}^{PV\,2}\right)^{1/2} \qquad (\mathrm{Pa.s.m^{-1}})$$

3. RECIPROCITY AND INVERSE FORCE IDENTIFICATION

The point velocity excitation is difficult to achieve in practice. But luckily, the objects and sound fields are generally linear, which means one can apply reciprocity:

(4) $$\left.\frac{P_r}{V_i}\right|_{Q_{r=0}} = \left.\frac{F_i^*}{Q_r^*}\right|_{V_{i=0}^*} \qquad (\mathrm{Pa.s.m^{-1}})$$

* denotes the reciprocal case
F_i = force at interface point i (N)

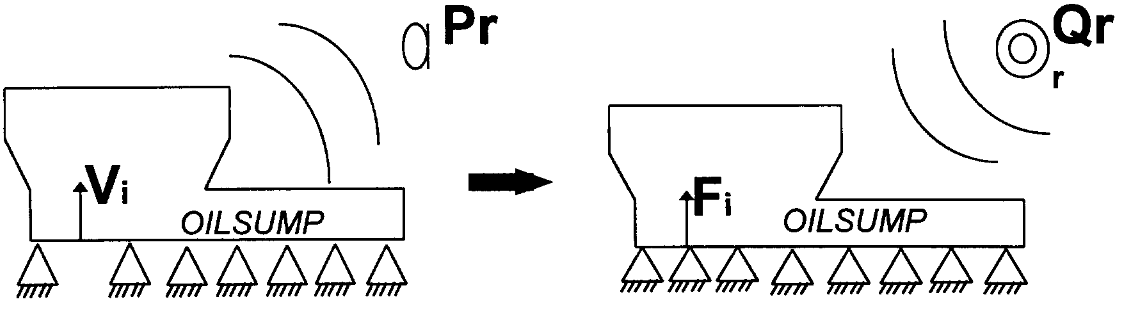

Fig. 2: Clamped object single point velocity excitation and reciprocal acoustic excitation.

This means that acoustic excitation with point sources in semi-free field of the clamped object will provide the same desired velocity acoustic transfer functions, as long as it is possible to measure the forces in the interface.

These forces are not measured directly because of instrumentation difficulties and the risk of modifying the system. Inverse force identification is used on the basis of acceleration measurements.

The test object is therefore mounted at its interface on a plate. The impedance of this baseplate at the interface is an order higher than the impedance of the object. A large number of accelerations are measured on the plate under acoustic excitation. After this the test object is removed and using structural excitation, the transfer functions (H_{ki}^{AF}) between the force at each interface point (i) and all acceleration signals (k) are determined.

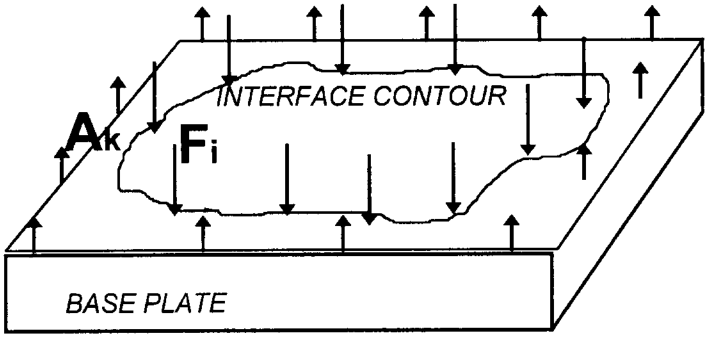

Fig. 3: Sketch of the baseplate, force excitation at the interface and accelerations on the base plate

All mechanical transfer functions are combined in a matrix. The inverse (or pseudo-inverse) of this matrix relates the measured accelerations to the interface forces.(Ref.2,3)

$$\underline{F} = \left[H^{AF}\right]^{-1} \cdot \underline{A} \qquad \text{(N)} \tag{5}$$

$\underline{A}$: vector of baseplate accelerations (m/s^2)

$\underline{F}$: vector of interface forces (N)

The measured volume velocity to acceleration transfer functions are thus translated to velocity acoustic transfer functions.

4. TEST CONFIGURATION

A thick damped aluminium plate is suspended close to the floor of a semi-anechoic test site. Eight accelerometers are mounted on the bottom of the plate. Two mid frequent volume velocity sources are positioned, alternating, at six points on a one meter hemisphere.(Ref.4)

Fig. 4: Photo of the test set-up; sound source above test object and baseplate

The oilsump variants are either glued, or bolted, directly onto the baseplate or the engine's bedplate is first glued onto the baseplate to provide the correct interface. The input accelerance functions of the baseplate and the oilsump can be seen in figure 5. The clamped BDC is sufficiently approximated, but still allowing some motion of the plate to be measured by the accelerometers.

5. RESULTS

A number of checks were performed on the intermediate results and on the final averaged velocity acoustic transfer.

The condition of the force identification matrix was checked. A condition limit was applied to improve the force estimations below 1000 Hz, where the number of excited modes in the baseplate is small. But this force estimation could be a study on its own, and is therefore not elaborated here.(Ref.2,3)

The next check is a comparison of the observed H^{PV} to a point source with the interface's velocity level and to a 100% radiation efficient oilsump with the interface velocity level. This confirmed that the correct order of magnitude is found.

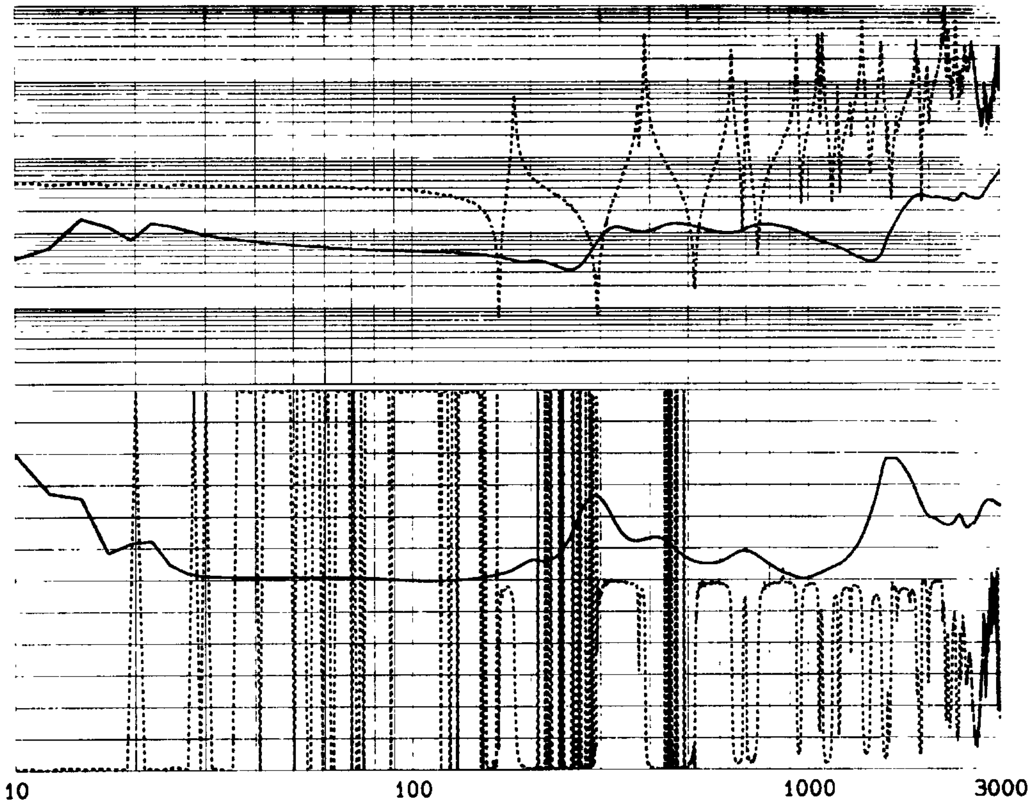

Fig. 5: Input accelerances, oilsump (dashed line) and baseplate (solid line)

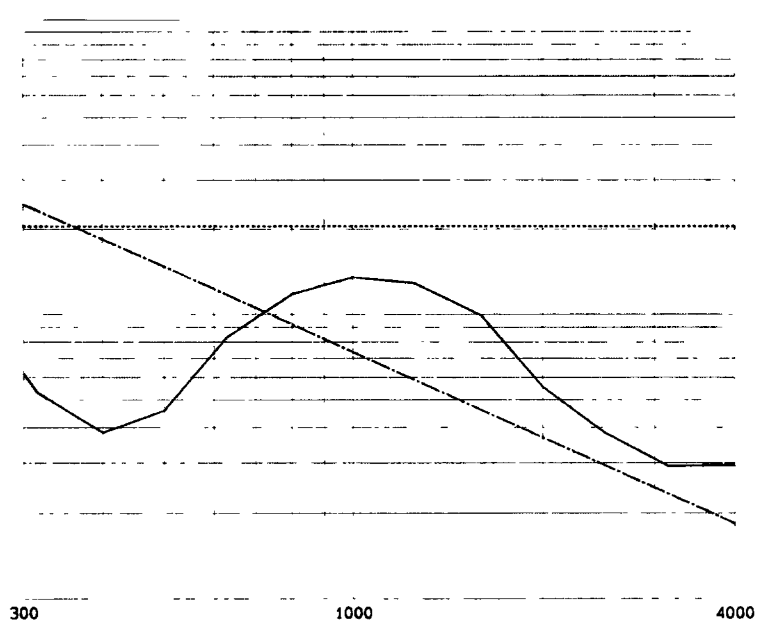

Fig. 6: First derivative of the averaged velocity acoustic transfer (Pa.s2/m) of an oilsump (solid line) and the point source reference curve (dashed) and the 100% radiation efficiency reference curve(dash-dot)

The averaged acoustic velocity transfer should not be sensitive to the exact location and number of interface velocity points. This is demonstrated in figure 7, where six or eight discretisation points were used.

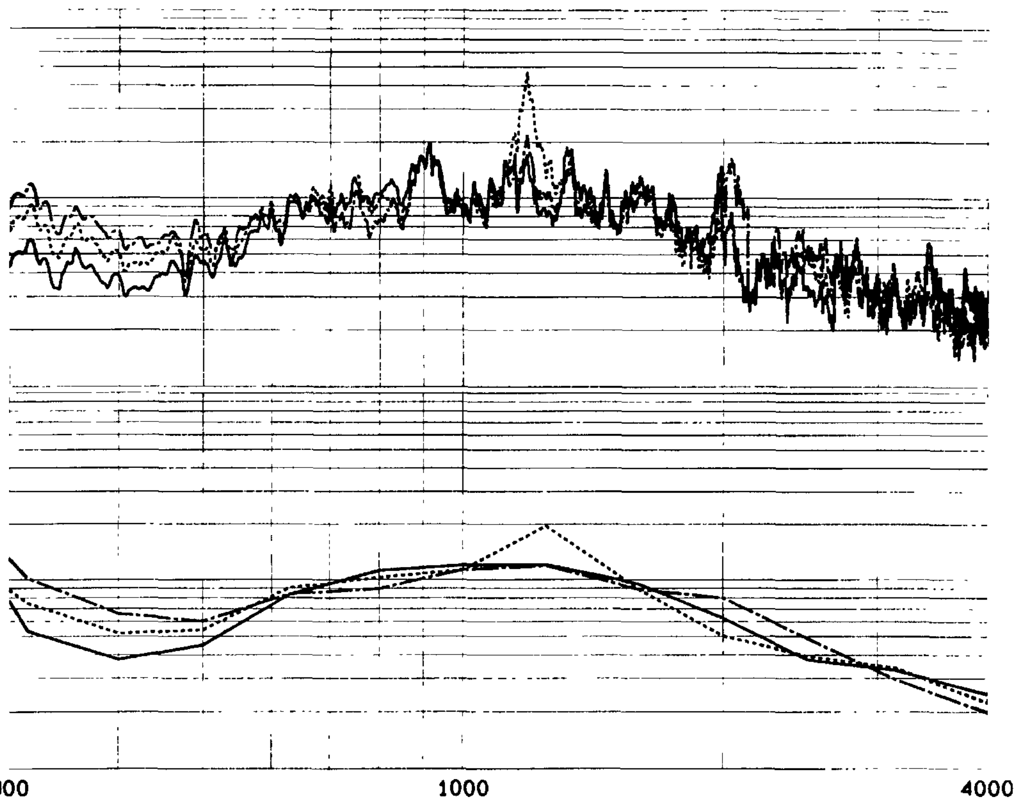

Fig. 7: First derivative of the averaged velocity acoustic transfer (Pa.s2/m), eight points(solid), six points(dashed), six other points(dashdot) Top window narrow band, lower window third octave bands

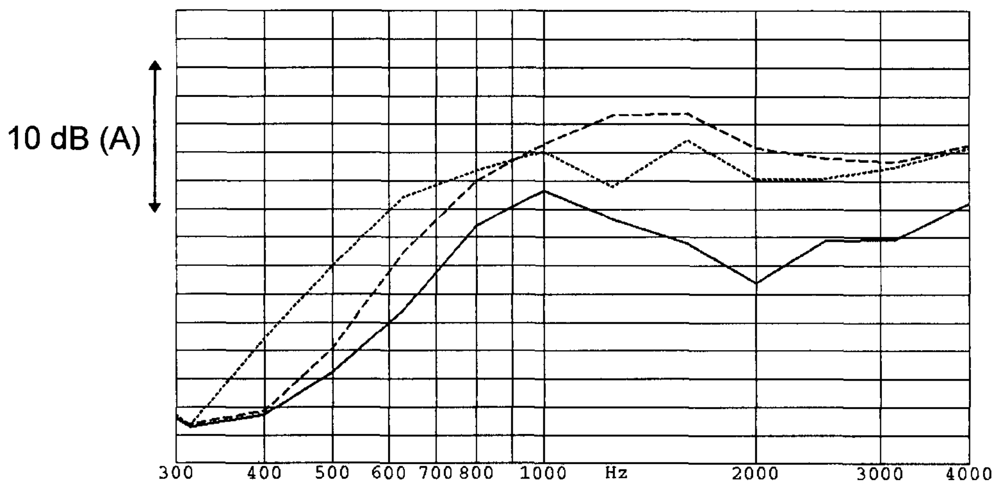

Fig.8: Averaged velocity-acoustic transfer (Pa.s/m)of the bare baseplate(solid line) and two oilsump variants(dashed/dash-dot)

Figure 8 shows three curves, the averaged velocity-acoustic transfer of two oilsumps and the average velocity-acoustic transfer of the baseplate without test object. The oilsump curve should be higher than the bare baseplate to be sure that the baseplate is not radiating too much.

The acceleration levels on the plate are higher with object than without for most of the targeted 300 to 3000 Hz frequency range. This is acceptable, but near to critical.
The low acceleration response under acoustic excitation (At 110 dB SPL) is part of the limitation. The other limitation is more principle; the interface of the oilsump to the engine, if it is closed by a perfectly damped material of minimal surface, will still radiate noise.(The damped aluminium baseplate is close to such a perfect termination)

Figure 8 shows the comparison of a fixed aluminium oilsump to a rubber gasket decoupled variant. Figure 9 shows the same aluminium oilsump compared to a fibre reinforced plastic variant. The results comply to both expected and testbed measurements. But the direct measurements of the noise from oilsump variants on engine testbeds have, so far, not been accurately enough to verify the observed two to eight dB range in variants.

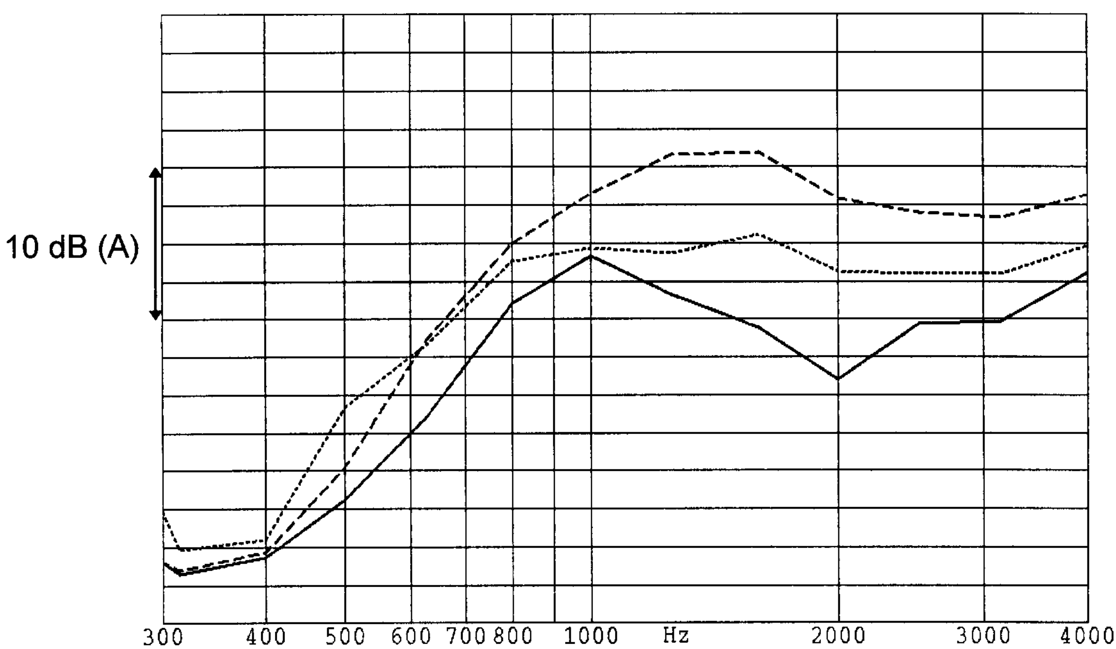

Fig. 9: Averaged velocity acoustic transfer (Pa.s/m)of the bare baseplate (solid line)the alu. oilsump(dashed) and the plastic variant (dash-dot):

6. CONCLUSIONS

The objective was a procedure for correct assessment of oilsumps, valve covers etc. including their mounting method.
The developed procedure approximates both the clamed boundary condition and the velocity excitation.
The average velocity-acoustic transfer quantifies the objects independently from the engine main structure dynamics and independent from the engine operating condition.
An estimation of the operational sound pressure due to the oilsump (or other) on a test bed can be obtained on the basis of the measured averaged acceleration of the interface.

$$|\tilde{P}|^2 = |\tilde{A}|^2 \;.\; |\tilde{H}^{PV}|^2 \qquad (6) \qquad (\mathrm{Pa}^2)$$

The practicality of the measurements has been proven, The analysis complexity is significant, but can be handled.

7. REFERENCES

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