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by

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reprinted from

Journal of LOW FREQUENCY NOISE, VIBRATION AND ACTIVE CONTROL

VOLUME 30 NUMBER 2 2011

MULTI-SCIENCE PUBLISHING COMPANY LTD.

An approach to an NVH investigation of vehicle hydraulic pumps

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Received 22nd December 2010

ABSTRACT
This work presents a method for analyzing the noise and vibration problems caused by hydraulic pumps in vehicles. This methodology is based on experimental research and coherent analysis of observed systems. Experimental investigations are conducted in order to determine dominant excitation of hydraulic systems. Vibrations and sound pressure levels of pumps were investigated with respect to the determination of the influences on pump NVH. Measurements were performed in different operating conditions. Pressure of oil in the hydraulic system was varied. A new method of assessment and determination of dominant NVH pump excitation is presented. Experimental results are used as input and output data for coherent analysis. Performed analyses of coherent functions showed dependence of pump velocity and sound pressure level especially in the low frequency region to 4.5 kHz. Results showed that the number of measuring channels can be reduced, especially in high frequency domain. Results can be used in experimental design.

Key words: vehicle, hydraulic pump, NVH, noise, vibration, coherent analysis

1. INTRODUCTION

Contemporary vehicles are faced with undesirable fluid borne, structure borne and airborne noise. Fluid borne noise is typically generated in the fluid reservoir or in the hydraulic lines and can propagate through mounting brackets. Structure borne noise is transmitted to the driver via the body structure through the pump mount, engine mounts, lines and system mounting brackets. The structure borne noise is heard as whine, the frequency is driven by the natural frequency and harmonics of the pump rotational Vane Passing Frequencies or Blade Pass Frequency, depending on the type of the pump [1, 2, 3].

The noises generated by the pumps of power steering systems and body control systems are examples of this kind of problem. Vane pumps are used in the majority of steering systems in automotive applications and radial piston pumps are used in body control systems. These pumps generate noise due the vane passing frequency. Basically, the hydraulic pump noise can be classified as a whine [2, 4, 6, 5].

The noise and vibration problems in hydraulic pumps of hydraulic power systems, as well as steering systems, body control systems, mainly whine noise, have similar generation mechanisms. For a diagnostic procedure, it is necessary to find the frequencies and amplitudes for which these noises become annoying for the passengers. In most cases, the whine noises are related to different running conditions of the power steering system and the diagnostics must be done for each one separately. However, the whine problems are directly related to the engine rotation and the frequencies of noise and vibration can be found knowing the construction and operation characteristics of the hydraulic pumps, [4].

The moan problem can be noted as a noise and/or a vibration. This kind of problem occurs when the vehicle engine is idling and when the wheels are being steered. In this case, the system needs hydraulic assistance and the pump is charged.

The whine noise arises for engine revolations above idling speed. In most cases, this category of noise becomes annoying for the vehicle passengers when the engine is accelerated to around 2000 rpm. Above this frequency, the engine noise masks the whine noise and the analysis become more difficult. Whine noise is also noted at the harmonics of the vane passing frequency. An important characteristic of this type of noise is that the harmonics do not appear to have the same contribution to the annoyance sensation inside the vehicle cabin [1, 2].

The aim of this work is to determine the dominant hydraulic system parameters on pump noise. The experimental investigations are conducted in order to get input data for suitable NVH modeling of hydraulic system.

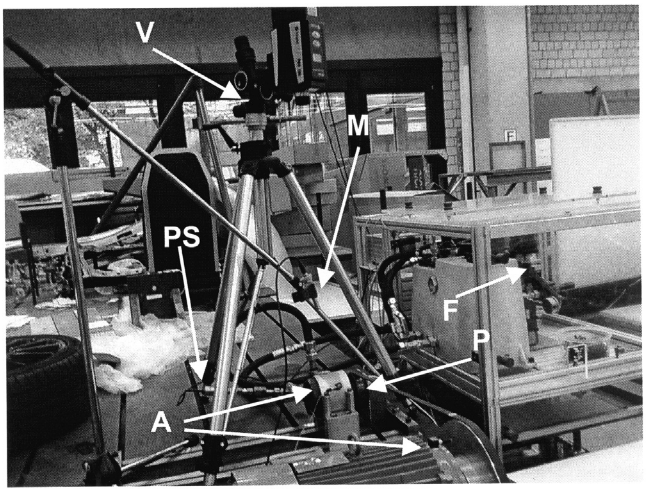

Figure 1 Measurement rig

2. EXPERIMENTAL WORK

Experimental research was conducted in the laboratory. Research was conducted in order to get input data for coherent analysis and to determine the dominant excitation of hydraulics systems. The measurement set up was designed to investigate the level of influence of hydraulic system parts.

The measurement rig used in this research is shown in Figure 1.

An electric motor drives the pump by belt transmission, a thermal control unit, a system to control pressure and oil reservoir. Hydraulic pump (P), is driven by an electric motor *SEW Eurodrive*, type DV132S4TF/IG1. Motor and pump accelerations are measured by three axial transducers *PCB* 356A16, (A). Outlet pump pressure is measured by a pressure sensor *Shaevitz*, Type P1221-0002-03 M0, (PS). Pump volume flow is measured by *Kracht* flow meter, type VC3F1DS, (F). Sound Pressure Level (SPL) is measured by a microphone *Microtech Gefel,l* type MK 250. (F). Pump velocity in the radial direction is measured by a digital vibrometer *Polytec,* type PDV100, (V). Pump speed is measured by an optical sensor *Baumer,* type: FZAM 181155 V152, (OS). NVH Measuring signals are acquired and stored by a *Pak Mueller* system for data acquisition and HVH analysis, [7].

Measurements are performed in order to get proper data for further NVH investigation. A radial piston pump is used. Radial piston pumps are commonly used in body control systems Measuring protocol is defined in order to obtain necessary data for analysis of dominant factors which have an influence on SPL of the pump. Acceleration transducers could not give a complete analysis in the human hearing frequency domain. Acceleration transducers have flat characteristics in the frequency domain to 4.5 kHz. The human ear can nominally hear sounds in the range 20 Hz to 20 kHz. The upper range tends to decrease with age, most adults being unable to hear above 16 kHz. The region 2 kHz to 5 kHz is very important for understanding of speech, [8]. The digital vibrometer is used to get data about pump behavior in the high frequency domain, as well as the pressure transducer and the volume flow transducer. Details can be seen in Figure 2.

Figure 2 Measuring pump detail

Measurements were preformed under different pressure condition: 30, 50, 60, 70 and 80 bar. Pump speeds were: 1000, 2000, 3000 and 4000 min^{-1} and variable from 500 to 4000 min^{-1}.

Sampling frequency was 48000 Hz, maximal frequency was 18750 Hz, number of samples was N=65536 -, frequency resolution was Δf=0.732 Hz, duration of signal was T=1.38 s, and a Hanning filter was applied. Sampling frequency was 2.56 times maximal frequency, [7]. Overlapping of 50% of measured signal, averaging procedure was applied.

Some experimental results are given in Figures 3 - 5.

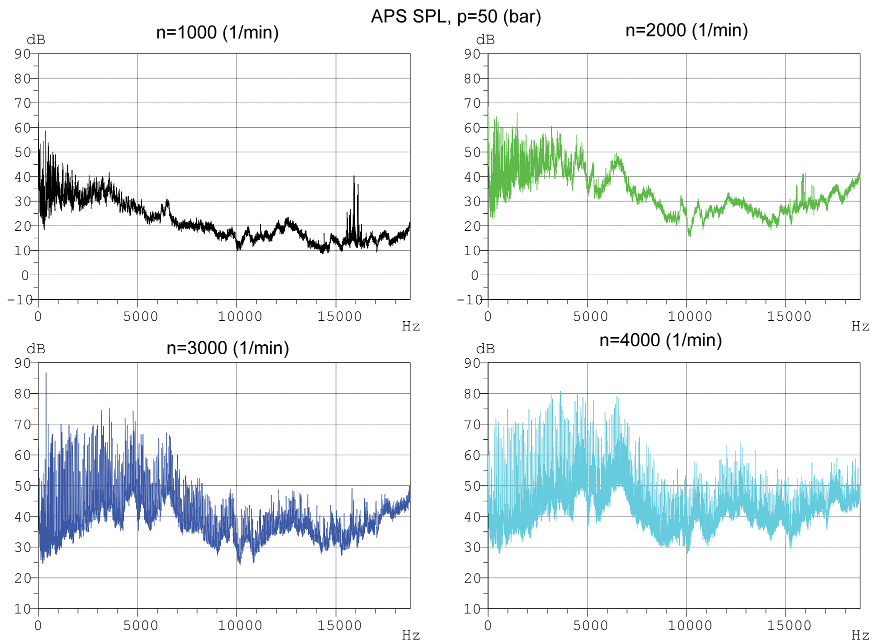

Figure 3 Auto power spectrum of Sound Pressure Level at constant pumpspeed and constant pump pressure p=50 bar a) n_p=1000 min^{-1}, b) n_p=2000 min^{-1}, c) n_p=3000 min^{-1}, d) n_p=4000 min^{-1}

In Figure 3, the auto power spectrum of SPL is given. Under constant pressure (50 bar), the overall SPL increases with increase in pump speed. At higher pump speeds, the auto power spectrum of SPL has more variation.

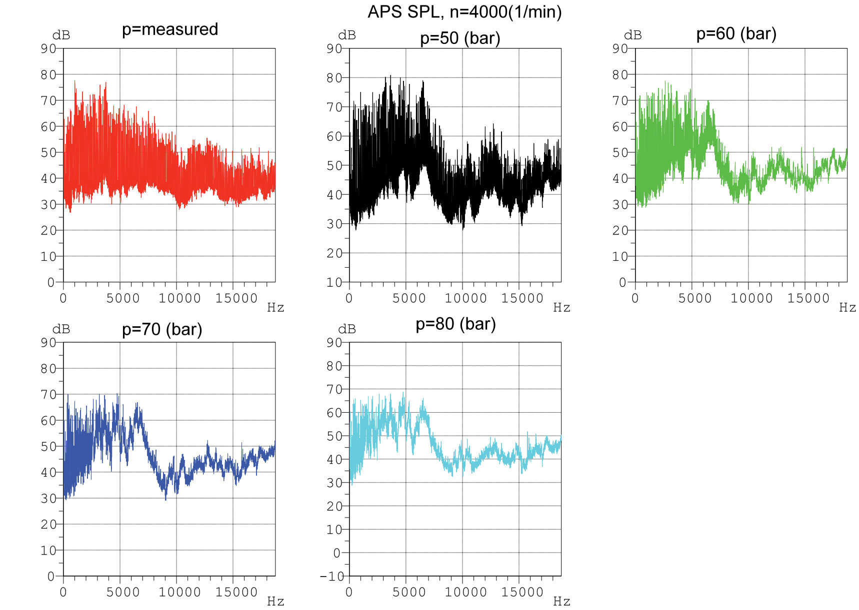

Figure 4 Auto power spectrum of sound pressure level at fixed pump pressure, pump number of revolution was $n_p = 4000$ min^{-1}: a) p = 50 bar, b) p = 60 bar, c) p = 70 bar, d) p = 80 bar

At constant pump speed, the auto power spectrum of the SPL decreases with increase of pump pressure, Figure 4. Signal oscillations are firmly marked in the frequency region to 10 kHz and are less so as pressure increases. Low pump pressure shows more pulsation in the Auto power spectrum of SPL. The resonant frequency is around 6.5 kHz. This peak is more pronounces under high pressure, Figure 4.

In Figure 5, a water flow diagram of the auto power spectrum of pump velocity in the radial direction is given.

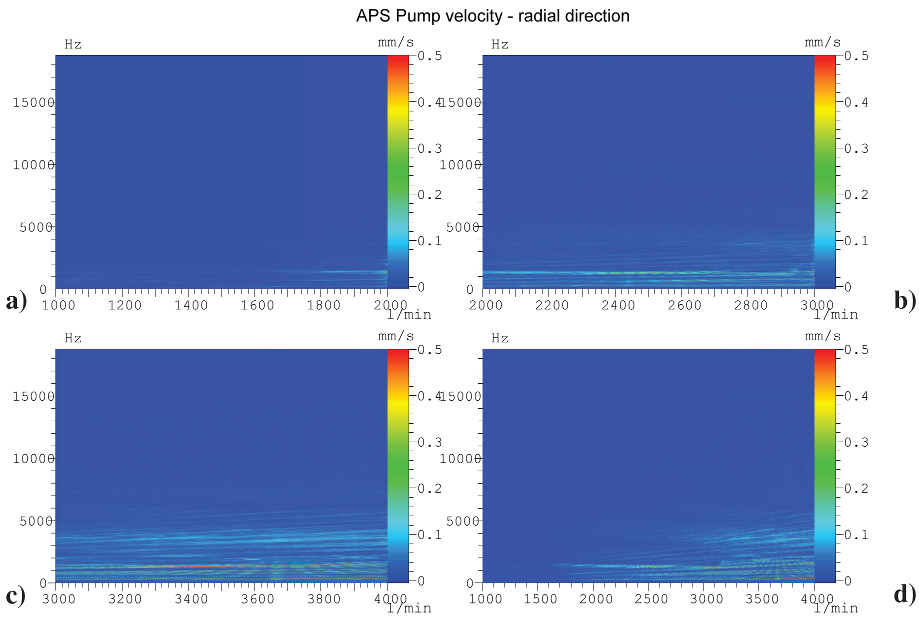

Figure 5 Auto power spectrum of pump velocity in radial direction at variable pump revolution number: a) $n_p = 1000 \div 2000$ min^{-1}, b) $n_p = 2000 \div 3000$ min^{-1}, c) $n_p = 3000 \div 4000$ min^{-1}, $n_p = 1000 \div 4000$ min^{-1}

Increase of pump speed caused increase of the flow velocity, especially in the frequency region up to 4.5 kHz.

3. DATA ANALYSIS

The aim of the work was to investigate the influence of different hydraulic system parameters on the SPL of the pump. Analysis of coherent functions is conducted according to computational algorithms given in chapter 10.3, pages 249-263 of reference [9].

In order to perform coherent analysis, software for calculation of partial and multiple coherence functions was developed according to [9]. The measuring system, given in Figures 1 and 2, can be presented as Multi Input One Output system, Figure 6. According to Figure 6, input data are:

- x_1 – pump acceleration in axial direction,
- x_2 – pump acceleration in tangential direction,
- x_3 – pump acceleration in radial direction,
- x_4 – pump pressure,
- x_5 – engine acceleration in axial direction,
- x_6 – engine acceleration in tangential direction,
- x_7 – engine acceleration in radial direction,
- x_8 – pump velocity in radial direction.

Output data, y, is

- x_9 – sound pressure level.

Figure 6 Multi Input One Output system in frequency domain till 4.5 [kHz]

Input signals x_i (i=1,…,q; q=1,2,…,8) are mutually correlated, $0<\gamma_{ij}^2<1$. Conditioned spectral density functions when linear effects of r! are removed and are determined according to computational algorithms given in chapter 10.3 [9]:

$$S_{ij\cdot r!} = S_{ij\cdot(r-1)!} - L_{rj} S_{ir\cdot(r-1)} \tag{1}$$

where $S_{ij.(r-1)!}$ are conditioned cross spectra between x_i(t) and x_j(t) when linear effects of $x_{(r-1)!}$ are removed from both x_i(t) and x_j(t), [9].

The frequency response function L_{ij} is calculated, [9]:

$$L_{ij} = \frac{S_{ij\cdot(i-1)!}}{S_{ii\cdot(i-1)!}} \tag{2}$$

Partial coherence functions are determined, according to [9]:

$$\gamma^2_{iy\cdot(i-1)!} = \frac{\left|S_{iy\cdot(i-1)!}\right|^2}{S_{ii\cdot(i-1)!} S_{yy\cdot(i-1)!}} \tag{3}$$

Results of coherent analysis in low frequency region are shown in Figures 7-10.

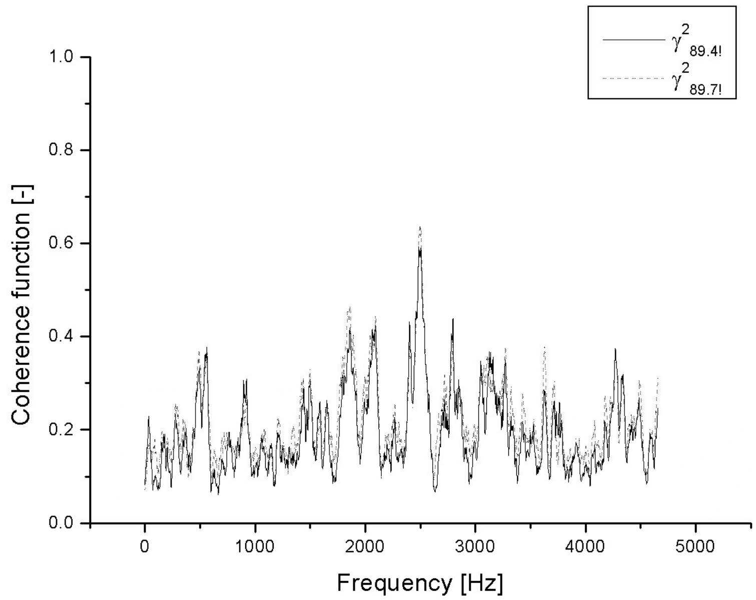

Figure 7 Partial coherence functions (Pump velocity in radial direction/ pump SPL), under constant number revolutions of pump $n_p = 3000$ min^{-1}, $\gamma^2_{89.4!}$ – is partial coherence function when linear effects of pump accelerations and pump pressure are removed, $\gamma^2_{89.7!}$ – is partial coherence function when linear effects of pump accelerations pump pressure and engine accelerations are removed,

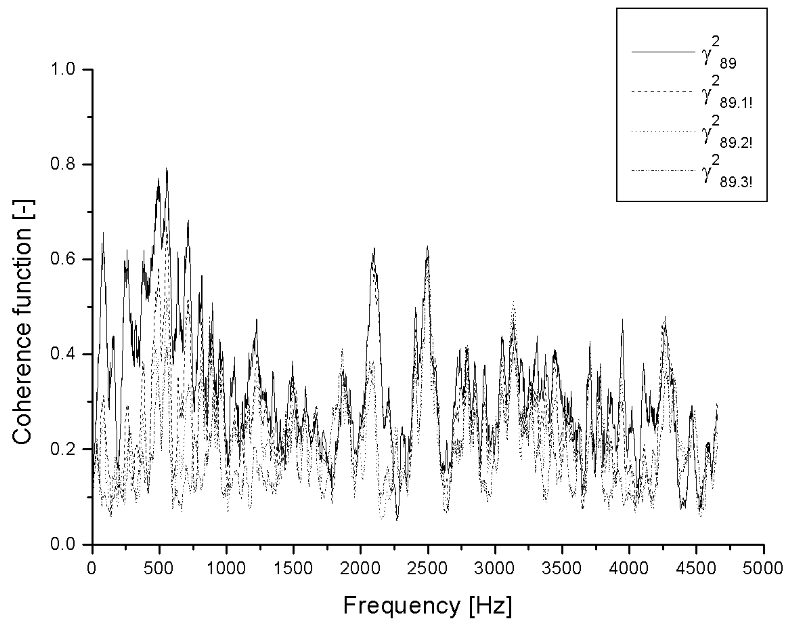

Figure 8 Coherence functions (Pump velocity in radial direction / pump SPL) , $n_p = 3000$ min^{-1}, $\gamma^2 89$ – is ordinary coherence function, $\gamma^2_{89.1!}$, $\gamma^2_{89.2!}$ and $\gamma^2_{89.3!}$ – are partial coherence functions when linear effects of pump accelerations in axial, radial and tangential directions are removed respectively,

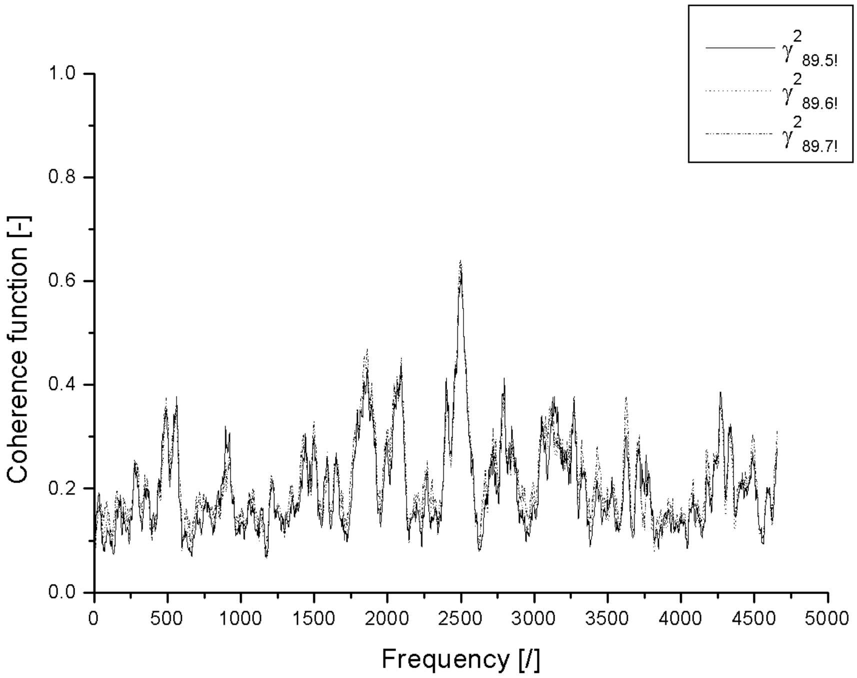

Figure 9 Coherence functions (Pump velocity in radial direction / pump SPL), $n_p=3000$ min^{-1}, $\gamma^2_{89.5!}$, $\gamma^2_{89.5!}$ and $\gamma^2_{89.5!}$ are partial coherence functions when engine accelerations in axial, tangential and radial directions are removed respectively

Coherence functions between pump pressure and SPL are at a middle level in the region around 2.5 kHz, while the magnitude of partial coherence functions has the same level as the ordinary coherence function.

Partial coherence functions are low, when the linear effects of pump accelerations are removed, Figures 8 and 9.

Ordinary coherence function between pump velocity in the radial direction and SPL is high, and partial coherence functions are lower than the ordinary coherence function. Partial coherence function between pump velocity in radial direction and SPL, when the pump accelerations in axial and tangential directions are removed, is lower than in case when the pump acceleration in axial direction only is removed.

It can be concluded that values of acceleration of the pump in all three directions and accelerations of the engine in axial and tangential directions have no influence on the coherence function between pump velocity in radial direction and SPL. Partial coherence function $\gamma^2_{89.7!}$, when linear effects of pump accelerations, pump pressure and engine accelerations are removed is lower than the previous one, Figure 9.

The ordinary coherence function between pump acceleration in the radial direction and pump pressure is higher than the partial coherence function when linear effects of pump accelerations in the axial and tangential directions are removed.

The ordinary coherence function is higher than the partial one. Resonances of the ordinary coherence function are the same resonances of the partial coherence functions.

Partial coherence functions are lower than ordinary ones. Significant differences can be observed between ordinary coherence function and partial coherence function when the linear effect of pump acceleration in the axial direction is removed.

The ordinary coherence function is lower than the partial coherence functions. Partial coherence function $\gamma^2_{89.7!}$ is lower than other partial coherence functions.

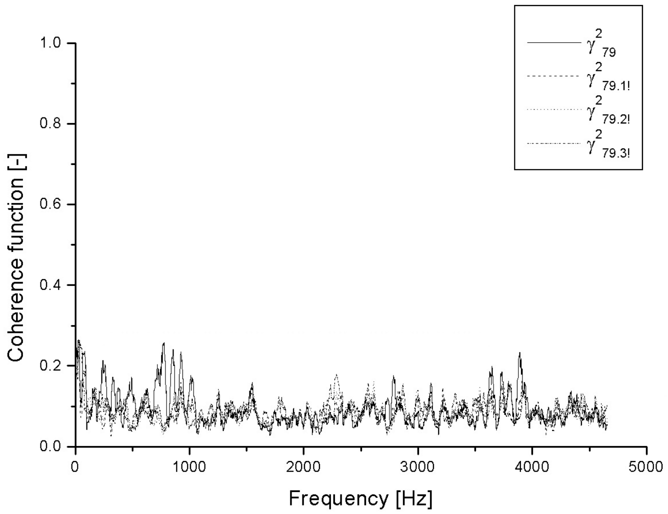

Figure 10 Coherence functions (Engine acceleration in radial direction / pump SPL) , $n_p = 4000\ min^{-1}$, $p = 80$ bar

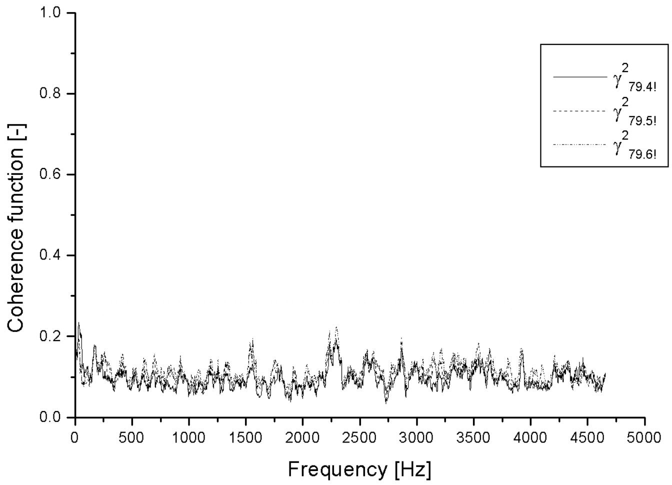

Figure 11 Coherence functions (Engine acceleration in radial direction / pump SPL) , $n_p = 4000\ min^{-1}$, $p = 80$ bar

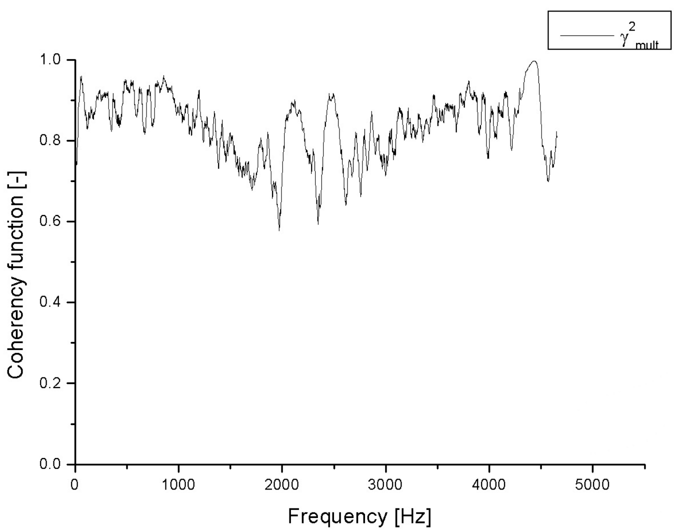

Figure 12 Multiple Coherence function (Pump velocity in radial direction / pump SPL), $n_p = 4000\ min^{-1}$, $p = 80$ bar

Based on results of partial coherent analysis ofthe system presented in Figure 6, with eight inputs and one output, dominant parameters could be observed and the system can be reduced with less input data especially in the high frequency region. Previously mentioned, coherent analysis is performed in the frequency region to 4.5 kHz, because of transducers characteristics.

Coherent analysis showed that the model given in Figure 6 can be substituted by model 2x1 given in figure 13. Accelerations of engine and pump cannot be considered as input data, because they have no influence on SPL. It is valid in laboratory conditions.

There are no differences in amplitude level between partial coherence functions when linear effects of pump accelerations and engine accelerations in axial and tangential direction are removed and have the same level as given in Figures 10 and 11.

In Figure 12, the multiple coherence function is given. The system has a low level of input noise.

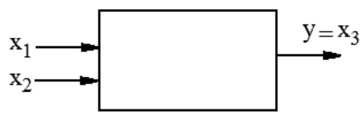

Figure 13 Multi Input One Output system in high frequency domain

A system with two inputs and one output, Figure 13, could be used in the high frequency domain with respect to the characteristics of applied transducers. Input data are: x_1 – pump velocity in radial direction and x_2 – pump pressure, output data is $y=x_3$ – pump SPL.

Results given here, present one way of contribution to NVH pump modeling in order to get valid data for further model development, Figure 14 -Figure 15.

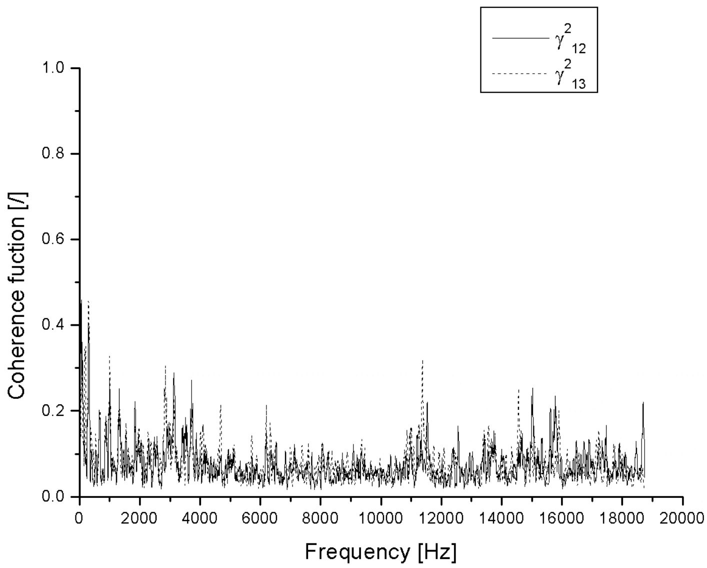

Figure 14 Ordinary coherence functions: γ^2_{12} - ordinary coherence function between pump velocity in radial direction and pump pressure, γ^2_{13} - ordinary coherence function between pump velocity in radial direction and pump SPL, $n_p=4000$ min^{-1}, p=80 bar

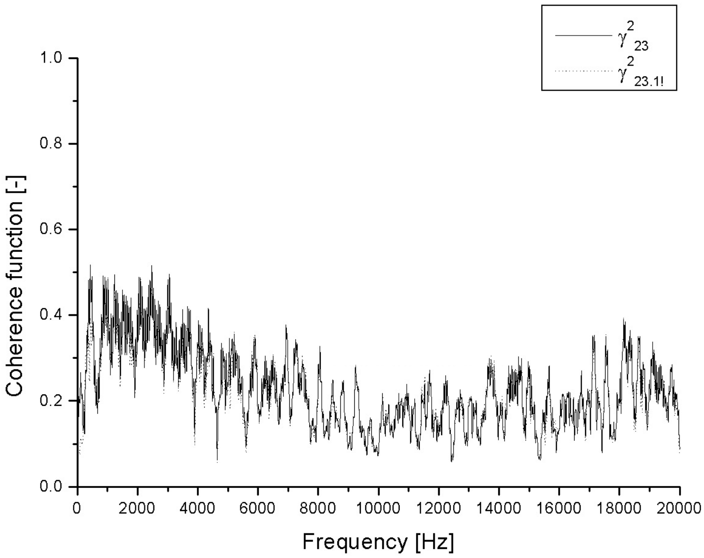

Figure 15 Coherence functions: γ^2_{23} - ordinary coherence function between pump pressure and pump SPL, and $\gamma^2_{23.1!}$ – partial coherence function between pump pressure and pump SPL when pump velocity is removed, $n_p = 4000$ min^{-1}, $p = 80$ bar

Results showed that the existing measuring rig could be updated by digital pump pressure control unit implementation. The measurement rig also can be used in developing active noise control.

For example, an investigation of new duct applications for pump specification, investigation at operating conditions in vehicle and developing NVH model of pump and ducts mounting.

4. CONCLUSIONS

Results obtained showed that performed coherent analysis can be applied in order to get sufficient input data for NVH modeling of vehicle hydraulic systems.

- Results of coherent analysis showed that:
- Partial coherence functions are low when linear effects of pump accelerations are removed which means that vibration of the pump cannot be neglected and are a dominant influence factor on pump SPL.
- Ordinary coherence function between pump velocity and sound pressure level is high, and partial coherence functions are lower than ordinary coherence function.
- When oil pump pressure is increased all values of coherence functions increase.
- Multiple coherence functions are significant in the low frequency region up to 4.5 kHz.
- The same analysis method can be applied on vane and radial piston pumps.
- Coherent NVH analysis showed that a reduced measuring rig can be applied. Depending on volume flow characteristics in the high frequency domain, a system with two input and one output signals can be used if the volume flow is constant.
- Method can be very useful in the initial experiment phase.

ACKNOWLEDGEMENT

Experimental research was conducted at Cologne University of Applied Sciences, Faculty of Automotive Engineering and Production, NVH Laboratory, under supervision of Prof. Dr. – Ing. Klaus Becker. Research was financially supported by DAAD grant.

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