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Årg 25 Nr 3 Dec/Jan 2007/2008



Vibrationsdagen 2007, Köpenhamn.

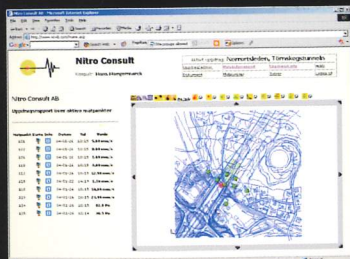


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Vibrationsdagen 2007

Vibrationsdagen 2007 hölls i år för andra gången i Köpenhamn. Programmet hade med viss hjälp av styrelsen i huvudsak sammanställts av Ole Døssing. Till denna endags-begivenhet den 22 november infann sig ett 40-tal personer i Dansk Arkitektcenters utställningslokaler på Gammel Dok som är ett före detta lagerhus utmed kanalen som i norr begränsar den gamla befästningsanläggningen och Christiania. Möteslokalen inrymde lägligt nog, så här inför julhelgen en mycket välsorterad bokhandel fylld med rikt illustrerad litteratur över byggnadsverk och arkitektur.

Ole som ledde mötet, presenterade bland andra Ann Sofie Wessberg, Metron Miljökon-sult AB, anso@metron.se, ny ordförande efter Tomas Odebrant i SVIB AU 3 Byggnadsvibrationer och Sören R. K. Nielsen, soren.nielsen@civil.auc.dk, Aalborgs Uni-versitet som övertagit rodret i SVIB AU 5 efter Thomas Abrahamson. Christian Madshus medförde också en kort presenta-tion av Linda Hårvik, NGI, som på årsmötet valdes in i styrelsen efter Håkon Heyerdahl. Närvarande på mötet var också Torbjörn Kloow, toby@acoutronic.se, Acoutronic AB, som i flera år varit föreningens revisor, och som valdes till dess skattmästare.

Av eftermälena att döma blev Vibrationsda-gen ett mycket uppskattat evenemang.

Årsmötesförhandlingarna blev även denna gång en rätt besvärlig historia eftersom revisorerna inte ansåg att föreningens räkenskaper kunde godkännas innan skat-teverkets beslut inhämtats. Som resultat av

revisorernas berättelse och rekommenda-tion "Vi föreslår därför att vi avvaktar med att fastställa resultat- och balansräkningarna, samt att styrelsen beviljas ansvarsfrihet för verksamhetsårets räkenskaper och för-valtning" uppfattade sekreteraren och jus-teringsmännen att räkenskaperna skulle bordläggas och godkännas i efterhand per korrespondens och att styrelsen skulle beviljas ansvarsfrihet. Men så var inte fallet. Revisorerna menade att styrelsen INTE skulle beviljas ansvarsfrihet och har seder-mera begärt att protokollet ändras. Detta är bakgrunden till att protokollet ännu inte är klart för publicering.

De som återstod att notera var skattever-kets bedömning av de begärda avdragen för moms samt huruvida skatteverket skulle godkänna rättelsen av förra årets deklara-tioner. I efterhand kan konstateras att skat-teverket med vissa ändringar av beloppen godkänt de begärda avdragen och reserva-tionerna samt godtagit den begärda ändrin-gen av förra året deklaration.

När underlagen färdigställts kommer det att cirkuleras för godkännande per e-post.

Alf Olsson



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SVIB styrelse

Ordförande

Ole Døssing, civ ing
od@rovsing-dynamics.com
Rovsing Dynamics A/S
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Vice ordförande

Ylva Vidhög, civ ing
yvi@forsmark.vattenfall.se
Forsmarks Kraftgrupp
+46 173 821 30

Sekreterare

Alf Olsson, ing
alf.olsson@elstandard.se
SVIB Normtjänst AB
+46 70 717 29 15,

Skattmästare

Torbjörn Kloow
toby@acoutronic.se
Acoutronic AB
+46 70 597 02 80

Övriga ledamöter

Linda Hårvik, civ ing
linda.harvik@ngi.no
Norges Geotekniske Institutt
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Markku Juntunen
markku.juntunen@vtt.fi
VTT
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Urban Emborg
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Arbetsutskott

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kjell.ahlin@swipnet.se
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Ann Sofie Wessberg
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AU 4 Vibrationers inverkan på människan

Johan Granlund, civ ing
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AU 5 Forskning och utbildning inom vibrationsområdet

Soeren R. K. Nielsen,
soren.nielsen@civil.auc.dk
Aarhus Universitet
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Alf Olsson

Postgiro

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SVIB Kansli och redaktion besöksadress:

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On the Comparison of Test Specification and Measured Field Data

Kjell Ahlin
Department of Mechanical Engineering
Blekinge Institute of Technology
SE-371 79 Karlskrona
Sweden
kjell.ahlin@bth.se

It is not always obvious how to compare a given test specification with measured field data. The paper presents a systematic method based on Shock Response Spectrum and Fatigue Damage Spectrum. A random on random test specification for tracked vehicles, AECTP 400, is used as an example.

INTRODUCTION

A common situation for the test engineer is that there exists a test specification and also measured field data. How can the two be compared? In many cases the test specification is made to mimic the real environment, but usually there is no way to directly make the comparison. A typical test specification for munitions transported as secure cargo in tracked vehicles, AECTP 400, is shown in figure 1.

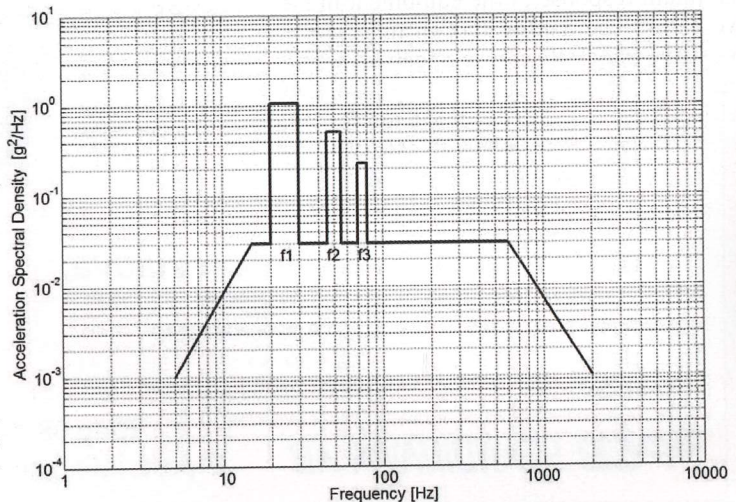


Figure 1. AECTP 400, method 401. Test specification for munitions transported as secure cargo in tracked vehicles.

The bandwidth of each narrow band is 10 Hz. The frequencies of the band should be harmonically related and swept, so that

$$15 \text{ Hz} < f_1 < 150 \text{ Hz}$$

$$30 \text{ Hz} < f_2 < 300 \text{ Hz}$$

$$45 \text{ Hz} < f_3 < 450 \text{ Hz}$$

(There are versions of the test specification where the bandwidths of the frequency bands are 5, 10 and 15 Hz, respectively)

All the levels in the spectrum are defined and the sweep rate should not exceed one octave per minute. The test time is one hour per axis.

The test specification is of course meant to mimic the background vibration and the track laying frequency with two harmonics.

COMPARISON METHODS

There are basically two different ways to do the comparison between the environment defined by the test specification and the environment measured in the field.

The first way looks at the damage to the transported goods or equipment from a maximum point of view. The method tries to find the maximum response of the equipment in the two cases. The natural tool in this case is to use the Shock Response Spectrum, SRS. There is an ISO standard, ISO 18431-4, that defines how to calculate the SRS for the field data. The catch here is how to calculate the SRS for the test specification. How this may be done is explained in the next clause.

The other way looks at the damage from a dose point-of-view. The proposed tool in this case is the Fatigue Damage Spectrum, FDS, which may be regarded as SRS with dose. FDS is defined below and a method to estimate the FDS for the test specification is given.

COMPARISON USING SHOCK RESPONSE SPECTRUM

A typical model for the SRS calculation, shown in Figure 2, assumes that the signal to be analyzed is applied to an array of independent SDOF systems at a common base and of interest then is the peak response of each of the masses.

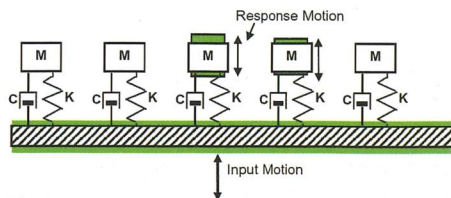


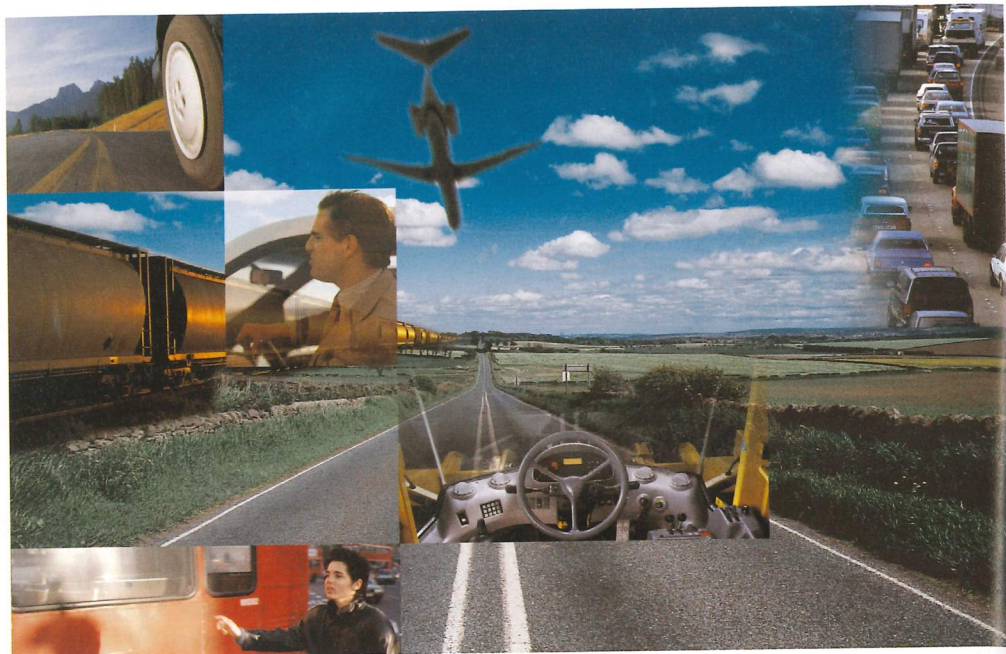
Figure 2. Typical model used for the calculation of the shock response spectrum (from Strether Smith). The motion is assumed to be applied to a common base of an array of damped and independent SDOF systems.

In ISO 18431-4 the Shock Response Spectrum is calculated by the use of digital filters, one filter for each SDOF system in figure 2. The filter coefficients are calculated using the ramp invariant method, introduced for this use by Dave Smallwood [1].

The SRS for absolute acceleration response may be calculated for the field data by a simple MATLAB script:

```
function [y,f] = srs(x,fs,fmin,fmax,fno,Q)
%SRS Shock Response Spectrum Ramp
Invariant Version Maximax
% [y,f] = srs(x,fs,fmin,fmax,fno,Q)
% y maximax is calculated
% f frequency vector Hz
% x data vector
% fs sampling frequency Hz
% fmin low frequency Hz
% fmax high frequency Hz
% fno number of frequencies
% Q Q value
y = zeros(fno,1);
f = zeros(fno,1);
%
```


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```

k1 = log(fmax/fmin)/(fno-1);
k2 = pi/Q/fs;
k3 = 2*pi/fs*sqrt(1-1/(4*Q*Q));
%
for n = 1:fno;
    f0 = fmin*exp(k1*(n-1));
    A = k2*f0;
    B = k3*f0;
    a = [1, -2*cos(B)*exp(-A), exp(-2*A)];
    b = [1-exp(-A)*sin(B)/B, 2*exp(-A)*(sin(B)/
        B-cos(B)), exp(-2*A)-exp(-A)*sin(B)/B];
%
    z = filter(b,a,x);
    y(n,1) = max(abs(z));
    f(n,1) = f0;
end

```

So, the calculation of the SRS for the measured field data is quite straight forward. To get the SRS corresponding to the test specification we first have to consider how to estimate the SRS of a vibration with a given Power Spectral Density, PSD_{in} . For each filter $H_n(f)$ in the SRS filter bank we can calculate the output Power Spectral Density, PSD_{out}

$$PSD_{out}(f) = PSD_{in}(f) \cdot |H_n(f)|^2 \quad (1)$$

We get the RMS value σ_n of the filter output by integrating the output PSD

$$\sigma_n = \sqrt{\int PSD_{in}(f) \cdot |H_n(f)|^2 df} \quad (2)$$

which is easily done, as we know the filter functions. This is a better estimate than the one usually given which assumes that the input PSD is constant with the level given for the resonance frequency under consideration. Then the PSD level is multiplied with the filter band width.

With a given RMS value, the n :th maximum value max_n can be estimated by the following equations

$$c_n = \sqrt{2 \ln(f_n \cdot T)}$$

$$C_n = c_n + \frac{0.5772}{c_n}$$

$$max_n = C_n \cdot \sigma_n \quad (3)$$

where f_n is the resonance frequency for the n :th filter and T is the total time under consideration. The equations may be found in [2].

We show how the method works with a simulation. We consider the case in the test specification when f_1 is at 75 Hz.

We create a time signal of a certain time length with the given PSD by adding a random phase to the given Fourier amplitude and perform an inverse Fourier transform. The given PSD and the calculated spectrum for the simulation are given in figure 3.

The PSD is calculated using a Hanning window and averages with overlap, which is the reason for the randomness in the spectrum.

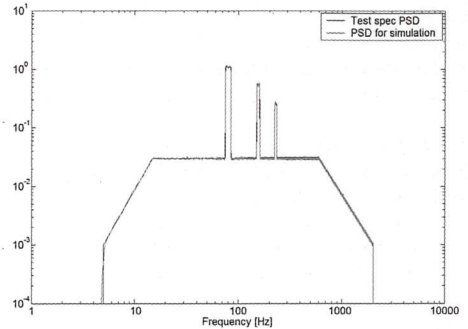


Figure 3. Comparison of specified and simulated Power Spectral Density.

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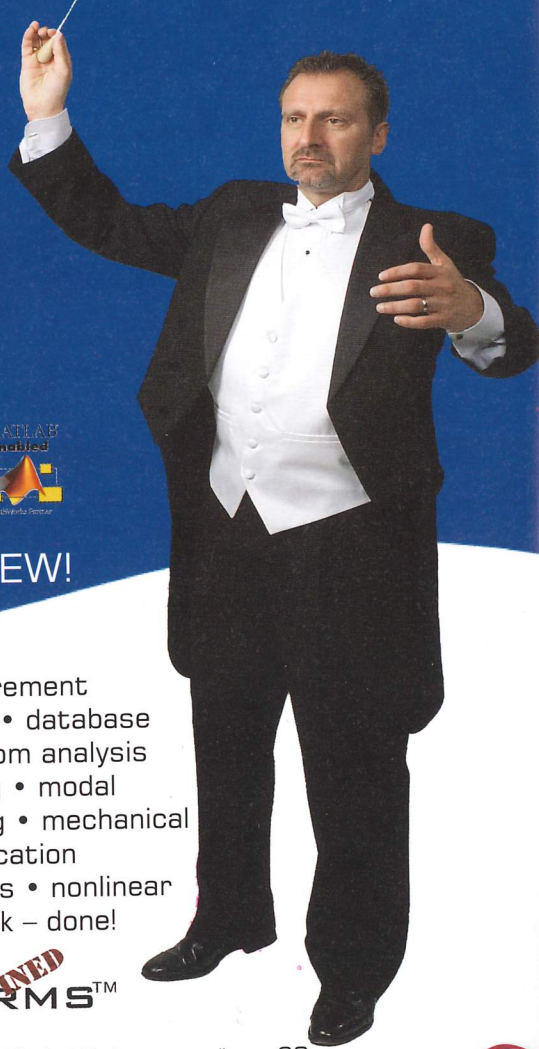


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We can now use the simulated time signal to calculate the SRS and compare with the estimated SRS from the specification PSD. The result is shown in figure 4. We find a good agreement.

We now use the method to estimate all SRS for the test specification by stepping the gliding frequencies one Hz at a time. We calculate the time spent in each position to mimic a logarithmic sweep, adding up to one hour. The result is shown in figure 5.

The maximum envelope for all SRS in figure 5 is our final result, the curve to be used as a comparison with field data SRS.

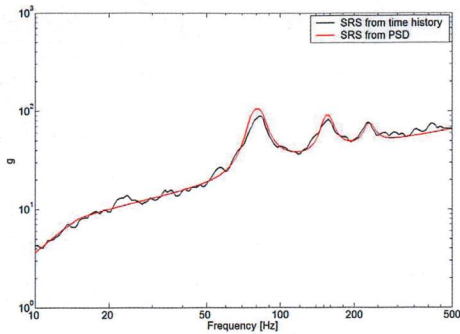


Figure 4. Comparison of calculated SRS for simulated signal and estimated SRS from specification PSD.

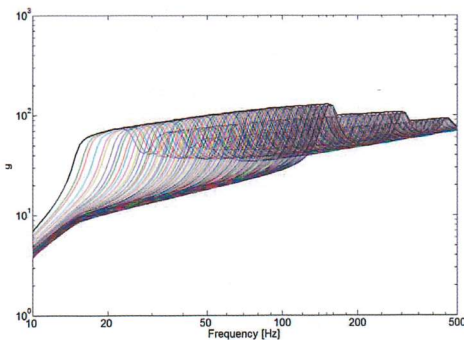


Figure 5. Estimated SRS for test specification, corresponding to one hour test time.

COMPARISON USING FATIGUE DAMAGE SPECTRUM

For the comparison of test specification and measured field data from a dose point-of-view, the Fatigue Damage Spectrum, FDS, is used. The FDS may be regarded as a Shock Response Spectrum with dose, see figure 6. As for the SRS calculation, the vibration signal to be investigated is applied to a set of SDOF systems. For each SDOF system the damage according to Miner's rule is estimated, using a simplified S/N curve, just defined by the exponent. To make a proper damage estimate, a lot of information has to be entered into the calculations, such as geometry of the object in question, the constants defining the proper S/N curve, etc. But, as the FDS only is used to make a comparison, we can forget all this and just set all unknown constants to one.

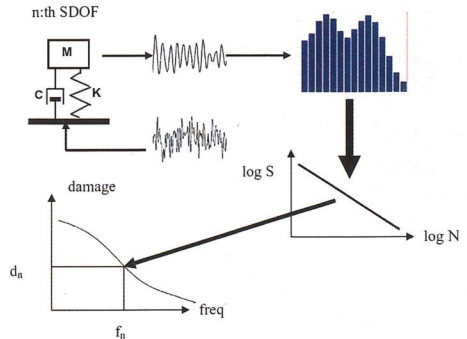


Figure 6. Principle for Fatigue Damage Spectrum, FDS, Calculation.

The steps in figure 6 are as follows:

1. The response to the given input vibration in the n :th SDOF system, usually as relative displacement, is calculated.
2. For the SDOF system narrowband output signal, a histogram, showing the number of peaks per peak amplitude interval is calculated.

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3. The histogram is applied to an S/N curve and Miner's rule is applied to calculate the damage.
4. The resulting damage represents one point in the FDS, corresponding to the resonance frequency of the n :th SDOF system.
5. As for SRS, the procedure is repeated for all interesting resonance frequencies.

The parameters to give to the calculation are the Q -value of the SDOF systems and the S/N curve exponent.

As the response of the SDOF system is a narrowband signal, the histogram may be replaced with a list of the peak values.

A simple MATLAB script to calculate FDS is given here:

```
function [y,f] = fdstdispl(x,fs,fmin,fmax,N,Q,b)
%FDSDISPL Fatigue Damage Spectrum,
%           relative displacement
%           [y,f] = fdstdispl(x,fs,fmin,fmax,N,Q,b)
%           y      FDS output
%           f      frequency axis for FDS
%           x      input data vector, accelera-
%                  tion
%           fs     sampling frequency in Hz
%           fmin   low frequency for calculation
%           fmax   high frequency for calcula-
%                  tion
%           N      number of frequency points
%                  for calculation
%           Q      Q value for calculation
%           b      Wöhler S/N exponent
```

```
k1 = log(fmax/fmin)/(N-1);
```

```
%
for n = 1:N;
fn = fmin*exp(k1*(n-1));
[B,A] = dispfilt(fs,fn,Q );
d = filter(B,A,x);
de = abs(extr(d));
y(n) = sum(de.^b)/2;
f(n) = fn;
```

```
end
```

The function `dispfilt` calculates the filter coefficients for a relative displacement output for an SDOF system. The function `extr` extracts the peak values in the response signal.

As for the SRS case above, there is a simple way to estimate the FDS when the input PSD is given. The principle is the same, but the estimate of the maximum is replaced with an estimate of the damage, when the RMS value of the SDOF output (and the S/N exponent b) is given. This is simple, providing the assumption of a Gaussian output is valid. A comparison (like the one in figure 4) between FDS calculated on a simulated time history and the estimated FDS given the PSD, is shown in figure 7. As the procedure seems to work well, we can estimate the full FDS for one hour sweep as we did for the SRS. The result is shown in figure 8. The difference to the SRS case is that the curve we have to use for comparison with field data is the sum per frequency as we deal with dose.

CONCLUSIONS

In this paper, a method to compare test specifications with measured field data has been proposed. The comparison is made in two ways, with a maximum approach using the Shock Response Spectrum, and with a dose approach, using the Fatigue Damage Spectrum. The key feature of the method is the introduction of a simple way to estimate Shock Response Spectrum and Fatigue Damage Spectrum from the given test specification. The method is demonstrated on the random on random test specification AECTP 400 for tracked vehicles.

REFERENCES

- [1] Smallwood, D.O.: "An Improved Recursive Formula for Calculating Shock Response Spectra." 51st Shock and Vibration Bulletin, 1980
- [2] NC Nigam., "Introduction to Random Vibration", MIT Press, Cambridge, MA, 1983.

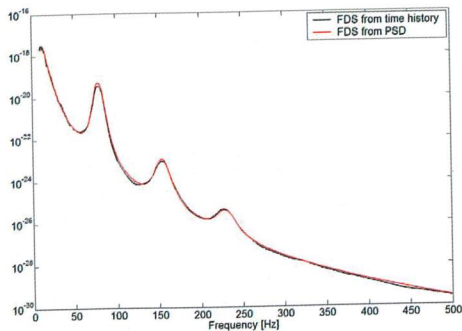


Figure 7. Comparison of FDS calculated from simulated time history and estimated from given PSD.

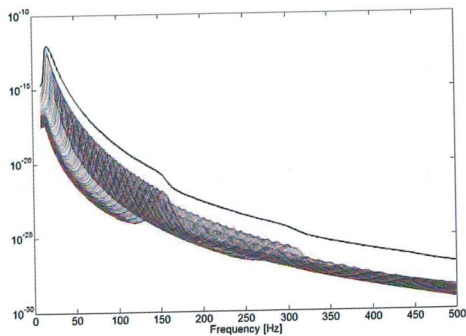


Figure 8. Resulting FDS corresponding to one hour test time. The solid curve can be used for comparison with FDS for measured field data. Framtiden för ljud- och vibrationskonsulter



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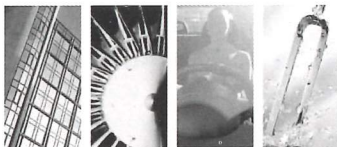
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On Nonlinear Parameter Estimation with Random Noise Signals

Andreas Josefsson
Martin Magnevall
Kjell Ahlin

Department of Mechanical Engineering, Blekinge Institute of Technology
SE-371 79 Karlskrona, Sweden
andreas.josefsson@bth.se

NOMENCLATURE

H	Frequency Response Function
G_{xx}	Single sided, auto-spectrum of signal x
G_{xy}	Single sided, cross-spectrum of x with y
$\mathcal{F}(\cdot)$	Fourier Transform
F, X	Force and displacement, frequency domain
f, x	Force and displacement, time domain
γ_m^2	Multiple Coherence function

ABSTRACT

In the field of nonlinear dynamics it is essential to have well tested and reliable tools for estimating the nonlinear parameters from measurement data. This paper presents an identification technique based on using random noise signals, as initially developed by Julius S. Bendat. With this method the nonlinearity is treated as a feedback forcing term acting on an underlying linear system. The parameter estimation is then performed in the frequency domain by using conventional MISO/MIMO techniques.

the model structure and thus methods for nonlinear characterization and localization are studied. The paper also demonstrates the various ways the method can be formulated for multiple-degree-of-freedoms.

The implementation of the method is illustrated with simulated data as well as a practical application, where the method is used to create a dynamic model of a test-rig with a significant nonlinearity.

To apply this method successfully it is necessary to have some pre-information about

1. INTRODUCTION

The overall aim with parameter estimation is to find suitable parameters to a mathematical model, based on measurements of the inputs and outputs of a system. The mathematical model can, for instance, be based on a beforehand known model structure. This is known as parametric modeling, which will be studied in this paper. Furthermore, the systems studied contain significant nonlinearities that must be included when creating the model, hence nonlinear parameter estimation.

Systems used in engineering can generally be separated into two distinct groups; linear or nonlinear. A system, H , which does not satisfy the following properties, is defined as nonlinear:

$$H\{c \cdot x\} = c \cdot H\{x\} \quad (1.1)$$

$$H\{x_1 + x_2\} = H\{x_1\} + H\{x_2\} \quad (1.2)$$

A useful tool for detecting nonlinearities is the frequency response function. For a significant nonlinearity the amplitude and/or the resonance frequencies will be dependent on the excitation amplitude. Thus, the FRF's gives a linear approximation at the specific excitation amplitude and fails to fully describe the dynamics.

This work will study a method where random noise signals are used and the parameter estimation is done in the frequency domain. The methods are described in Chapter 2 and Chapter 3. This will be followed by a brief analysis of the errors introduced when doing the parameter estimation. Finally an experi-

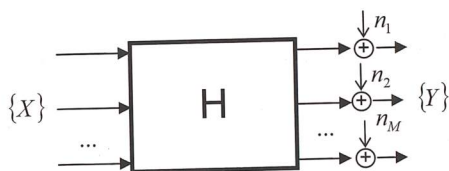
mental test is performed, in Chapter 5, where a real nonlinear structure is studied.

2. USING RANDOM NOISE SIGNALS FOR NONLINEAR SYSTEMS

One of the most promising identification techniques for nonlinear parameter estimation was initially developed by Bendat [1]. The method estimates the parameters in the frequency domain with conventional Multiple-Input-Multiple/Single-Output (MIMO/MISO) procedures. The latter is often used in linear theory and is demonstrated below.

For a general linear system, with inputs and outputs, and noise on the measured outputs, as shown in Figure 2.1, a least-square solution that minimize the noise on the outputs (H_1 -estimator) can be determined at each frequency as

$$\mathbf{H}_1 = \mathbf{G}_{YX} \cdot \mathbf{G}_{XX}^{-1}$$



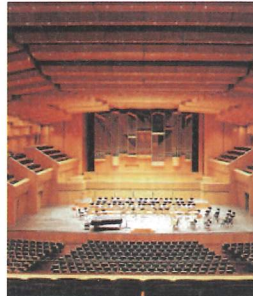
In Equation (2.1), $\mathbf{G}_{YX}$ is the cross-spectral density matrix with size $(M \times N)$, $\mathbf{G}_{XX}$ is the auto-spectral density matrix with size $(N \times N)$ and $\mathbf{H}$ is the estimated transfer matrix with size $M \times N$. Furthermore, a multiple coherence function can be calculated for each output as

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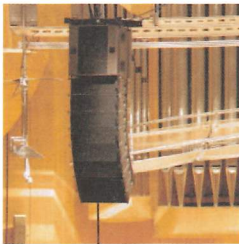
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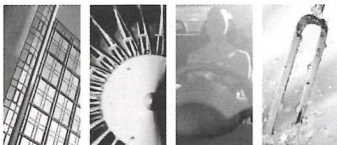


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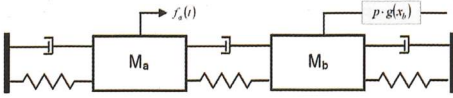
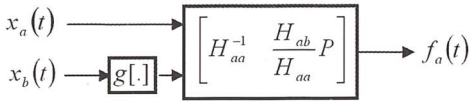


Figure 2.2: A nonlinear system with two degrees of freedom and a nonlinear element connected to the second dof. The nonlinearity is described with a nonlinear operator $g(\cdot)$ with a coefficient p .

The system shown in Figure 2.2 can be modeled in frequency domain as

$$\begin{pmatrix} H_{aa} & H_{ab} \\ H_{ba} & H_{bb} \end{pmatrix} \begin{pmatrix} F_a \\ -P \cdot \mathcal{F}(g(x_b)) \end{pmatrix} = \begin{pmatrix} X_a \\ X_b \end{pmatrix}$$

Alternatively, equation (2.3) can be rewritten, for each row, as



$$X_a \cdot H_{aa}^{-1} + \frac{H_{ab}}{H_{aa}} \cdot P \cdot \mathcal{F}(g(x_b)) = F_a$$

$$X_b \cdot H_{ba}^{-1} + \frac{H_{bb}}{H_{ba}} \cdot P \cdot \mathcal{F}(g(x_b)) = F_a$$

There are several possible ways to formulate this problem with MIMO/MISO. With the original Reverse-Path technique (RP) [1], Equation (2.4)-(2.5) can be used and the analysis is accomplished as illustrated in Figure 2.3. With Nonlinear Identification Through Feedback Of Outputs (NIFO) [2], Equation (2.3) is used directly and the linear system and the nonlinear coefficients are estimated as shown in Figure 2.4.

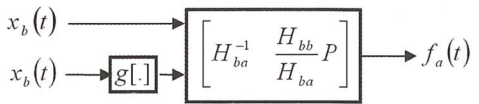


Figure 2.3: Reverse-Path (RP): Two MISO analyses are performed and using the reciprocity of the underlying linear system H_{aa}^{-1} , H_{ba}^{-1} and the nonlinear coefficient P can be identified. Here, $g(\cdot)$ is a nonlinear operator.

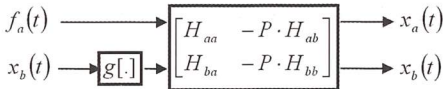


Figure 2.4: Nonlinear Identification Through Feedback of Outputs (NIFO): A single MIMO analysis is performed and using the reciprocity of the underlying linear system H_{aa} , H_{ba} and the nonlinear coefficient P can be identified. As in Figure 2.3, $g(\cdot)$ is a nonlinear operator.

In the above figures, all inputs and outputs quantities are given in the time domain, to emphasize that the input signal $x_b(t)$ must pass through the nonlinear function, for instance $g(x)=x^3$, before the spectral densities can be calculated. The transfer functions are then given by Equation (2.1) and, in both cases, two multiple coherence

functions can be calculated with Equation (2.2).

Notice that the transfer function $P(f)$ can be both frequency dependent and complex. However, in this example $P(f)$ will be, after a sufficient number of averages, real-valued and constant throughout the frequency range. It is also possible to use several nonlinear inputs and obtain a polynomial fit to the true nonlinearity, as may be the case with a gap- or clearance-nonlinearity.

As shown in the above example, the NIFO-technique is more compact and easier to work with - especially for larger systems with several nonlinear elements - since

MIMO is used as shown in [2,3]. With the reverse-path technique the system must always be rewritten with force as output as in Equation (2.4)-(2.5).

In an ideal case, without any noise, both methods will give the same estimates since they are mathematically identical. However, in the presence of bias errors and - for a real measurement - external noise, the estimates are not identical as will be discussed in chapter 4.

3. CLASSIFY NONLINEARITIES AND FINDING NONLINEAR NODES

With the method studied in previous chapter the type of nonlinearity and its location in the system must be known for the analysis to work which may seem like a major drawback. However, as will be shown in this chapter, the MIMO/MISO analysis can actually be a good tool to determine the model structure.

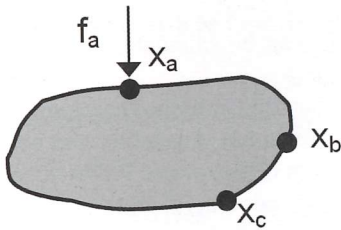


Figure 3.1: A blackbox system used in simulations with three degree of freedoms. A nonlinear element is connected either to a single node, or is dependent on the relative motion between two nodes.

A simulated blackbox system with three degrees-of-freedom is studied, as shown in Figure 3.1. The system does not fulfill

the properties given by Equation (1.1)-(1.2) since a nonlinear element is connected somewhere. By studying the FRF data it was possible to see the presence of a hardening spring nonlinearity.

Random data was simulated at all three dofs and, for the analysis, a reverse-path model was created for the system as illustrated in Figure 3.2. Here, three polynomials is used to describe the nonlinearity but the input signal $q(t)$ is unknown. The nonlinear element may be connected to ground from any dof or may depend on the relative motion between two dofs.

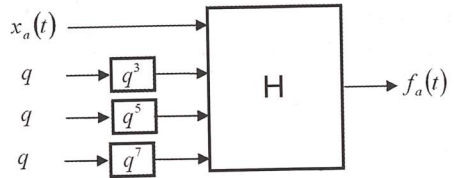


Figure 3.2: A reverse-path model created for the unknown system shown in Figure 3.1. The input signal $q(t)$ is unknown and depends on the nonlinearity.

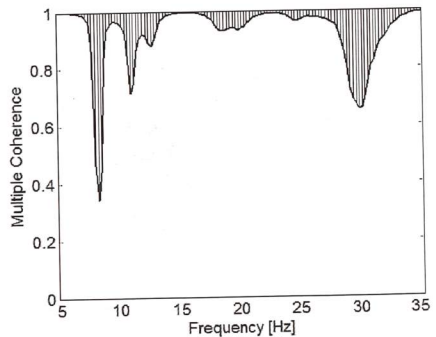


Figure 3.3: Multiple Coherence Function. The area shown in the figure was calculated for each combination of inputs and an error function was defined as $1/Ae$.

To solve this problem all possible combinations of inputs was tested and the multiple coherence function was used to evaluate each combination. If the model is correct the multiple coherence function should be close to one for all frequency. The area, A_e , shown in Figure 3.3, was therefore calculated for each combination of inputs. The error function is then defined as $1/A_e$, i.e the highest value for the overall best multiple coherence.

As shown in Figure 3.4, this method correctly detects that the nonlinear element is connected between x_b and x_c . With the nonlinear nodes identified the test was repeated but with different nonlinear functions as shown in Figure 3.5. Nonlinear functions that depend on the velocity, as well as the unphysical x_2 , were included for comparison. The true nonlinearity in the black-box system was a cubic function that depends on the relative motion between x_b and x_c . As illustrated with this example, random data can in many cases be used to classify nonlinearities and finding nonlinear nodes.

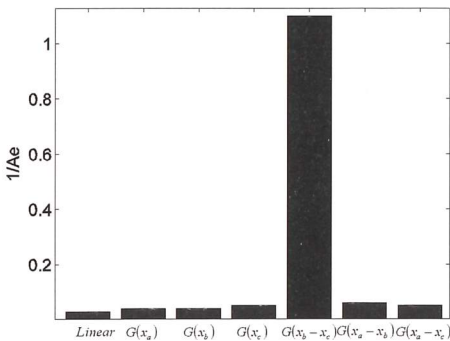


Figure 3.4: The method correctly detects that the nonlinear element is connected between x_b and x_c . The true nonlinearity in the black-box system was a cubic function that depends on the relative mo-

tion between x_b and x_c .

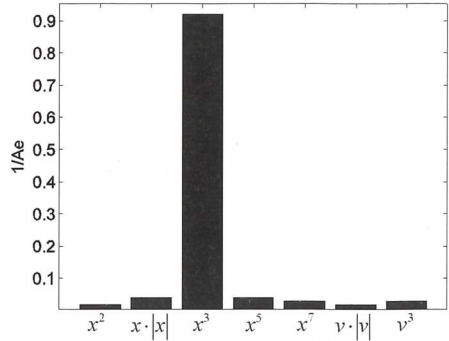


Figure 3.5:

The presence of a cubic stiffness could clearly be seen. Nonlinear functions that depend on the velocity, as well as the unphysical x_2 , were included for comparison.

4. ERRORS IN THE ANALYSIS

When using random noise signals the estimated spectral densities will contain random and bias (systematic) errors. The effect from random errors can be controlled with a sufficient number of averages and normally, overlap processing is applied for more effective use of data. The bias error, on the other hand, does not necessarily decrease with averaging since this error comes from consistently estimating an incorrect spectrum, for instance due to aliasing or leakage.

The time-leakage effect can give a considerable random error. This error occurs when the signal is truncated and an output signal is measured which is not totally correlated with the input signal. In the beginning of the output signal, for a certain time block, there is data which depends on the input signal from the previous time block. This effect will be even more significant for lightly damped structure. The leakage error can be reduced by increasing the block length. The

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bias error is a function of the truncation and the window used and may also be reduced by increasing the block length.

When using MIMO/MISO to identify non-linear systems it is essential to use a sufficient amount of data due to a relatively high sensitivity to these errors, as can clearly be seen in simulations.

Bias errors can also be introduced by contaminating noise in the measurement. The H1-estimator, Equation (2.1), assumes that the noise on the input channels in the MIMO/MISO model is small and can be neglected, while the H2-estimator, Equation (4.1), neglects the noise on the output channels.

$$\mathbf{H}_2 = \mathbf{G}_{YY} \cdot \mathbf{G}_{XY}^{-1} \quad (4.1)$$

However, the H2-algorithm is less useful for multiple-input models since it is necessary to calculate an inverse to a non-square matrix if the number of inputs and outputs are not equal. For this reason, the H1-estimator must be used instead. The effect this will have on the parameter estimation can be demonstrated with a simulated example.

A duffing system is used with the following parameters: $M=150 \text{ kg}$, $C=800 \text{ N/ms}$, $K=4e6 \text{ N/m}$ and a cubic term $P=2e12 \text{ N/m}^3$. The excitation force was set to an amplitude where the resonance would increase approximately 15%-20% relative to the underlying linear system.

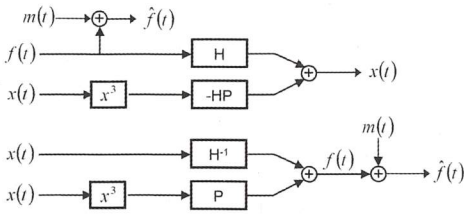


Figure 4.1: Analysis of a duffing system with a NIFO model (above) and the reverse-path technique (below). A small amount of noise is added to the force channel as shown in the figure.

Two possible MISO-models (NIFO and Reverse-Path) are shown in Figure 4.1. When doing the analysis a small amount of noise was added to the force channel as shown in the figure. The noise was designed with an rms-value equal to approximately 5%-10% of the rms-value of the force signal.

The simulation was done with a sampling frequency of 512 Hz and enough data was simulated so that the analysis could be done with 216 in blocksize (fft) and 200 averages with 50% overlap.

As shown in Figure 4.2, the NIFO formulation gives relatively large bias error while the reverse-path formulation performs better since the H1-estimator minimizes the effect from noise at the output. This is an undesirable effect since in reality the dominating noise may be on the force signal due to force-dropouts and a very weak force signal at the resonance.

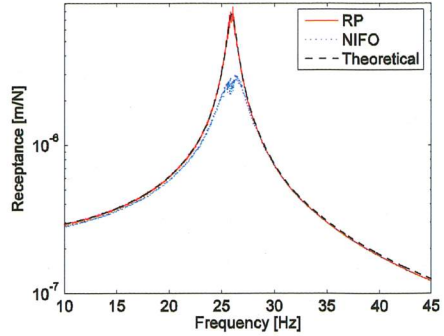


Figure 4.2: Estimated linear system - comparison between the two models shown in Figure 4.1. After averaging a large bias error is left with NIFO while the reverse-path formulation performs better since the H1-estimator minimizes the effect from noise at the output.

5. EXPERIMENTAL TEST

In this chapter the methods described previously is used to model a real structure with a small but significant nonlinearity. The test structure is described in section 5.1 and the experimental setup is explained in Section 5.2 followed by results from the nonlinear analysis in Section 5.3.

5.1 THE NONLINEAR TESTRIG

The structure under test consists of a clamped beam supported at point A with two clamped thin beams as shown in Figure 5.1. The design comes from a similar test-rig that was developed by J. Ferreira [4] and has since then been used to verify various identification methods [4, 5].

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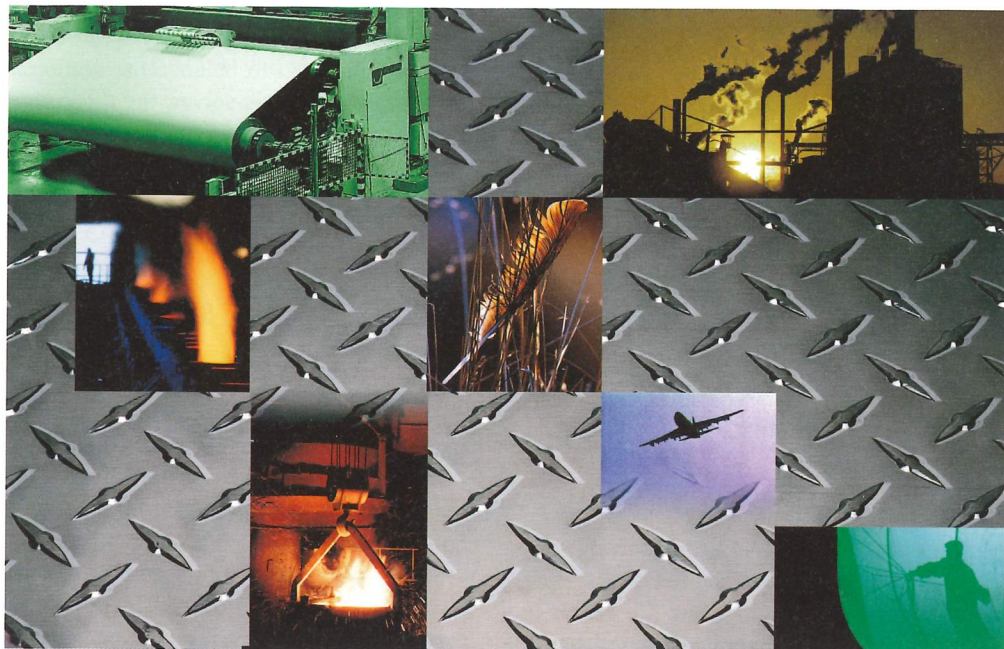
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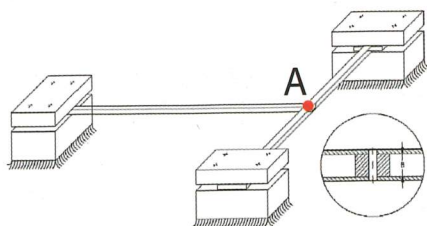


Figure 5.1: The structure used in the experiment consists of a clamped beam with two fixed transversal beams connected at point B. The clamped beam has a cross section of (15x8) mm with a free length of 420 mm and the two transversal beams has a cross section of (13x1.4) mm and free length 242 mm.

The properties of the beam are interesting since it can be described as a linear system (the clamped beam) with a local nonlinearity at the free end. Due to the thin geometry of the transversal beams the linear assumption is no longer valid. As illustrated in Figure 5.2, the geometry is changed as the beams deflect and the length along the center line of the beams (L') becomes longer than the original length L . This will introduce axial forces (N) which will increase the stiffness of the beam [6]. However, the material is still elastic and when the force is removed, the system will return to its initial position without any change in material properties. Hence, this effect is known as geometric nonlinearity since the nonlinearity is only related to changes in geometry.

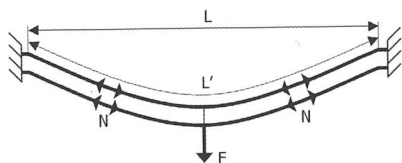


Figure 5.2: The transversal beams will behave nonlinearly at higher displacements. As the force is increased the beams become longer than the original length L , and an axial force is introduced along the beams. The stiffness will therefore increase. This effect is known as geometric nonlinearity.

Using ABAQUS® [7], a nonlinear static model could be created. ABAQUS® uses an iterative scheme where the force is applied incrementally and a Newton-Raphson solver is used to balance the nonlinear equations. Using this solver, the theoretical nonlinear force-displacement relationship at point A could be calculated as

$$F(x) = 1.76 \cdot 10^4 \cdot x + 2.97 \cdot 10^9 \cdot x^3 \quad (5.1)$$

5.2 EXPERIMENTAL SETUP

The complete setup for the experimental test is shown in Figure 5.3. The structure is firmly fixed to a rigid table and a rigid steel module is used to fix the shaker. The free end of the cantilever beam was selected as a response point since the nonlinear element is connected to this point. With the identification method described previously it is necessary to measure the response at the nonlinear degree of freedoms. However, the reference point (force) can be selected at any point along the cantilever. For practical reasons the shaker is fixed to the same point and thus the linear point-receptance and the nonlinear coefficients will be estimated.



Figure 5.3: The complete setup for the experimental test. The structure is fixed to a rigid table and a shaker is used to excite the structure.



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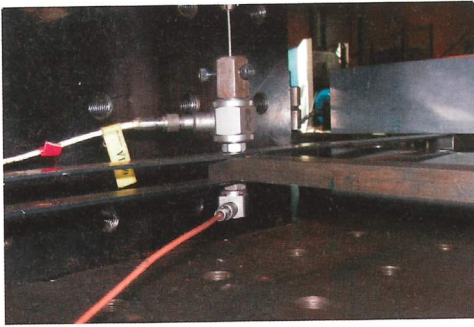


Figure 5.4: A detail view of the experimental setup. A force transducer and an accelerometer are used at the selected measurement point.

A force transducer was mounted on the top of the beam and the accelerometer was mounted under the bottom beam as shown in Figure 5.4. The force signals were created in MATLAB [8] and Signal Calc Mobilyzer [9] was used to acquire the raw time data. All data analysis was done in MATLAB.

Initially, no evident nonlinear phenomena could be seen in the measurements due to force dropouts which only increased with higher excitation amplitude and thus avoiding the high displacement at the resonance. It was therefore necessary to create voltage signals which added more power at the resonances and carefully make sure that the force spectrum obtained was constant over the desired frequency range.

5.3 ANALYSIS OF EXPERIMENTAL DATA

The structure behaves linearly at low excitations and it was therefore necessary to find a level of the excitation where the nonlinear effects are clear. A force amplitude which made the resonance move approximately 15% was selected for nonlinear analysis.

In the following two figures, two nonlinear analyses are compared with the raw FRF. The latter is the FRF obtained with conventional linear theory (SISO). Thus, the raw FRF is calculated without any compensation for nonlinear effects.

The nonlinear effects can clearly be seen in the raw measured FRF. A disturbance occurs at approximately $2 \cdot f_0$ and $3 \cdot f_0$. This disturbance can also be seen in the coherence function. A part of the response that is obtained at the resonance frequency will, due to the nonlinearity, create higher frequency response which the linear model fails to describe.

Two reverse-path models were created. First, a cubic function was used to describe the nonlinearity and the result can be studied in Figure 5.5. In the second model, shown in Figure 5.6, a cubic- and a square-polynomial are used to describe the nonlinearity. It could be seen that the true nonlinearity was a non-symmetric function and thus a cubic function was not enough to describe this behavior.

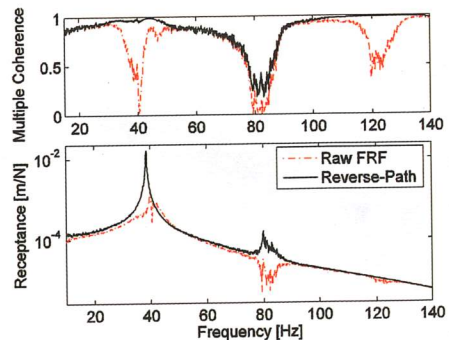


Figure 5.5: The Raw FRF (linear theory) compared with the linear system estimated with reverse-path technique. A cubic function is used to describe the nonlinearity. As shown, the FRF and the coherence are improved at the resonance and at three times the resonance frequency.

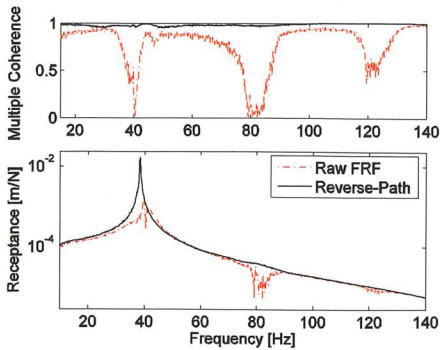


Figure 5.6: The Raw FRF (linear theory) compared with the linear system estimated with reverse-path technique. A cubic- and a square- function are used to describe the nonlinearity. As shown, the FRF and the coherence are considerably improved over the frequency range.

Figure 5.7 shows the estimated linear system with different methods. The true linear FRF was measured at very low excitation where all nonlinearities are unaffected. As

shown in the figure, the estimated linear system with NIFO techniques gives larger amplitude errors. One possible explanation for this was given in chapter 4; if the force signal is contaminated with noise, the reverse-path technique will perform better.

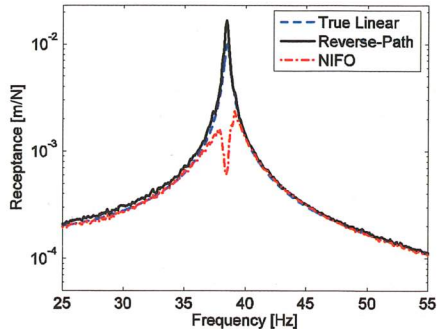


Figure 5.7: The estimated linear system with different methods. The true linear FRF, used as reference, was measured at very low excitation. As shown, the estimated linear system with NIFO techniques gives larger amplitude errors.



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The real parts of the nonlinear coefficients are shown in Figure 5.8. By definition, for a zero-memory nonlinear system, the imaginary parts of the nonlinear coefficients should be relatively small after a sufficient number of averages. For this experiment, the imaginary part was approximately ten times smaller than the real part after averaging. This may indicate a small amount of nonlinear damping which was not considered in the modeling.

The estimated cubic coefficient was $2.02 \cdot 10^9$ N/m³, which was considered to be reasonable in comparison with theoretical value from section 5.1. Due to the non-symmetric force, a square coefficient was identified as $4 \cdot 10^6$ N/m². When studying the test rig it could be seen that the slender beams were not perfectly aligned with the cantilever beam, and thus creating a non-symmetric force. When this effect was included in the nonlinear static model, a better correlation between the experimental results and the simulated results could be obtained.

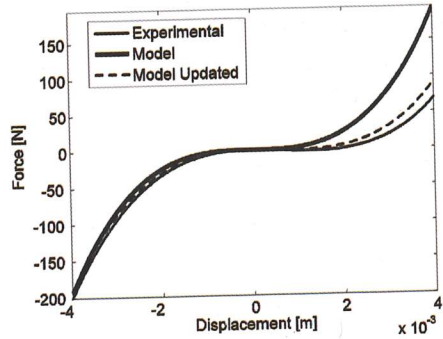
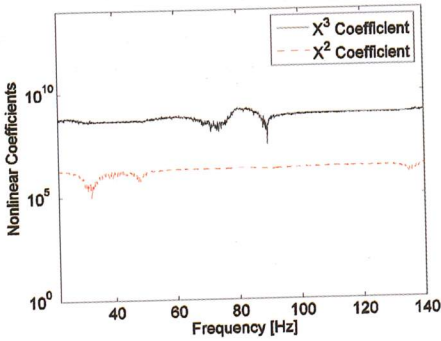


Figure 5.8: The nonlinear coefficients (left) and the estimated nonlinearity compared with the model (right). The nonlinear static model could be updated with the experimental parameters.

6. CONCLUSION

Nonlinear parameter estimation can be performed with random noise signals by using conventional MIMO/MISO techniques. The methods described in this work treat the nonlinearity as a feedback forcing term acting on an underlying linear system. Thus, the true linear system and the nonlinear coefficients can be estimated by measuring the input and output quantities.

The approach that is used in this work requires some pre-information about the type

of nonlinearity present, and its location in the system. If this information cannot be obtained before the analysis it may be possible to try a set of different combinations and then utilize the multiple coherence function as shown in Chapter 3.

When using MIMO/MISO there are several possible ways to formulate a nonlinear problem. Two methods have been studied in simulations and experiments - referred to as NIFO and Reverse-Path. As shown in

this work, it is essential to create nonlinear models which best compensate for the errors introduced with contaminating noise.

An experimental test was done on a structure with a significant nonlinearity and the results, presented in section 5.3, shows that the underlying linear system and the coefficients describing the nonlinearity could be identified. When comparing the results with linear theory a clear improvement could be seen: The estimated linear system is free from disturbance and with a correct resonance frequency. Thus, a simple but very useful application of the methods may be to make a better estimate of the linear system, for instance when comparison are made to a numerical linear model.

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7. ACKNOWLEDGE

This work has been sponsored by The Knowledge Foundation in Stockholm, Sweden.

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Denna klara uppmaning fick USAs delegation på klimatmötet på Bali varefter de gjorde en helomvändning och svarade –

"We will go forward and join consensus,"

Detta visar att det går att påverka även den mest hårdhudade motståndaren och att "consensus" är en förutsättning för framgång i miljöfrågor.

Sekretariatet för IEC TC 104 ligger vid sidan av SEKs aktuella fokus och kommer i fortsättningen inte att finansieras av SEK på annat sätt än med resebidrag som utgår till alla SEK kommittéer. SEES initiativ att ta över ansvaret för sekretariatet ligger däremot helt i linje med SEKs inriktning för hur IEC arbetet skall bedrivas och är därför välkommet. Problemet är att det inte gått att uppnå full finansiering. Fyra företag har ställt sig bakom förslaget och lovat bidra till dess finansiering i 3 år framåt.

TC 104 är en tvärteknisk kommitté som utarbetar horisontella standarder för generell tillämpning. Arbetets betydelse för uppbyggnaden och utformningen av provningslaboratorier och forskningsinstitutioner över hela världen, för industrins utveckling av nya produkter och system, för upprätthållande av samhällets service och kontroll av produkter och deras tålighet mot miljöns inverkan på dess väg från tillverkaren till användaren och av dess livslånga funktion kan inte nog understrykas eftersom detta är grundläggande, horisontella och internationellt vedertagna metoder och anvisningar.

Provningen av miljötåligheten är ett av de första stadierna i utvecklingsfasen av en produkt. Klimatisk- och mekanisk provning föregås endast av specifikationsfasen och följs av verifikationsstadiet. Det är först när produkten fullgjort sitt uppdrag som den blir en belastning för miljön och först då blir den föremål för den nu aktuella miljödebattens uppmärksamhet. När man nu slagit fasts att jordens atmosfär inte kan ta emot en ökande belastning kommer uppmärksamheten snart att riktas mot de faktorer som går att påverka nämligen produkters miljötålighet, deras livslängd och bestämningen av relevanta miljöfaktorer. I skenet av den belysningen får TC 104 arbetet inom en snar framtid förnyad aktualitet.

IEC har i år särskilt uppmärksammat de horisontella standardernas betydelse för hela organisationens strävan efter samstämda och entydiga standarder för provning av elektrotekniska produkter samt för det övergripande målet att undvika dubbelarbete. Ett 10-tal av det totalt 81 standarderna som TC 104 utarbetat är i dag antagna som horisontella standarder. Övriga är så kallat "basic safety standards".

Sekreterarna, Magnus Davidsson för SEES och Alf Olsson för TC 104 har nyligen beskrivit innebörden och omfattningen "projektet" att flytta IEC sekretariatet från SEK till SEES. Handlingarna som består av en projektbeskrivning, en rapport från arbetet med miljöprovnings- och miljöklassificeringsarbetet samt ordförande Kjell Ahlins reserapport från mötet med TC 104 i Kina den 17-21 september i år distribuerades till intressenterna den 18 december 2007.

Målsättningen är att behålla sekretariatet i Sverige och att finansiera arbetet under tre år med Alf Olsson som sekreterare och under den tiden stärka och komplettera den svenska kommittén TK 104 samt att rekrytera en efterträdare på sekretariatsposten för TC 104. Lyckas inte detta återstår för SEK att återlämna uppdraget till IEC. Sekretariatet kan därefter hamna var som helst i världen och så mycket är känt att det är inte de traditionella huvudaktörerna i arbetet som står först i kön att få nya uppdrag av IEC.

Ledamöterna har därför med stort intresse följt turerna i det svenska agerandet och kommitténs ordförande, Hermann Ruoss, Tyskland, har speciellt tackat de svenska kollegorna inom SEES för initiativet.

Projektledarna har därför begärt att Elektroindustriföreningen, ELIF, som har en lång tradition av att stödja exakt denna typ av standardisering ställer upp och deltar i finansieringen.

If you're not willing to maintain the secretariat of TC 104, please get out of the way - Var den Österrikiska uppmaningen till den svenska nationalkommittén av IEC. Svaret dom fick var:

SEES will maintain the secretariat, go forward and achieve consensus!

Detta måste nu med eftertryck följas upp med ett starkt engagemang och erforderliga resurser.

Alf Olsson

Secretary IEC TC 104 Environmental conditions, classification and methods of test

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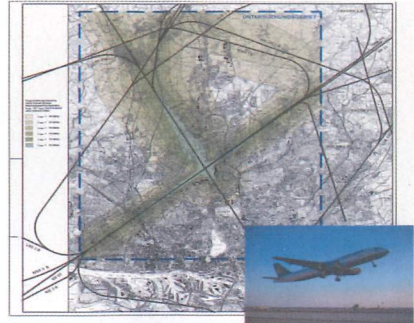
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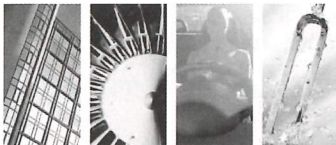
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Vad är en konsult? Namnet härstammar från latinets "consultus" som betyder råttslärd och "consulo" som betyder rådslå, arbeta för, hjälpa, handla och rådfråga (källa Nationalencyklopedin).

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Det finns fördelar och nackdelar med olika konsultföretagsformer. En tendens i branschen är att många storkunder att väljer att teckna ramavtal med ett fåtal konsultföretag med många kunskapsdiscipliner. Då kan det vara en fördel att jobba i ett företag med 3700

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P.s. När grundades Ingemansson och när grundades ÅF? Skicka in rätt svar per e-post till mig. Fem rätta svar premieras med varsin svensk trisslott cirka två veckor efter denna tidnings distribution. Jag är själv enväldig domare.

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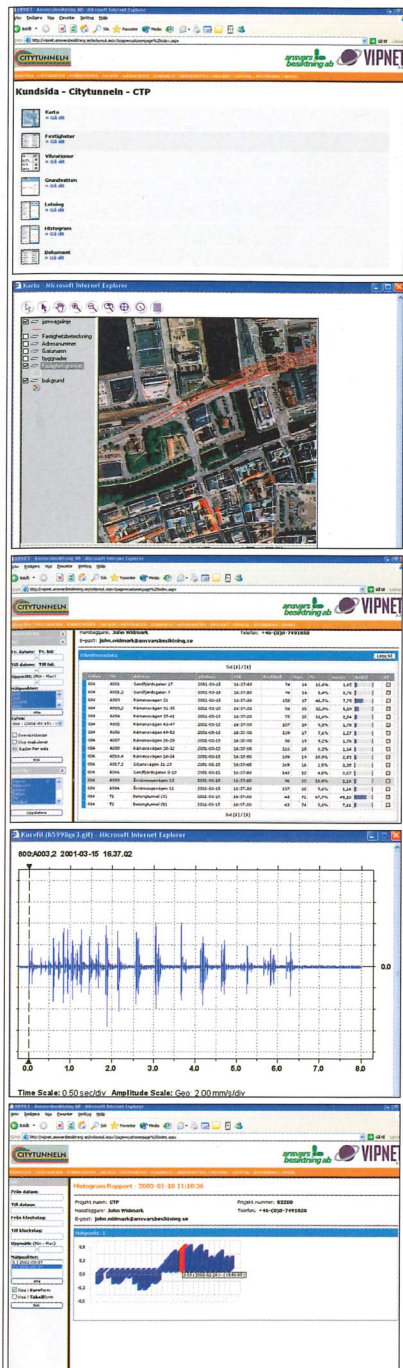
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Till styrelsen valdes Linda Hårvik och Torbjörn Kloow. Ann Sofie Wessberg och Soeren R. K. Nielsen tillträdde som nya ordföranden i SVIB AU 3 resp. AU 5.

Short CV - Linda Hårvik



Date of Birth: 9 November 1961

Nationality: Norwegian

Position: Head Communication and Organisation Division Director, NGI

Education: M.Sc., 1984, Diploma in Geotechnical Engineering, University of Trondheim, Norwegian Institute of Technology, Department of Civil Engineering. "Master of Management", MP from the Norwegian School of Management, BI in 2005 within organizational psychology and change management.

Previous Positions:

2000-2006 Division Director, Geomechanics Division, NGI

1997-2000 Division Director, Analysis Division, NGI

1995-1997 Deputy Division Director, Analysis Division, NGI

1995-1997 Responsible Information Technology and senior Engineer, Analysis Division, NGI

1984-1995 Project Engineer, Analysis Division, NGI

1984 Research Assistant, University of New Brunswick, Department of Civil Engineering, New Brunswick, Canada

1981-1984 Research Assistant in calculus and linear algebra, Norwegian Institute of Technology, Trondheim

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1981-1984 Research Assistant in calculus and linear algebra, Norwegian Institute of Technology, Trondheim

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Tiden går så fort – Ja denna klyscha besannades nu under hösten 2007, då vi på Acoutronic har 20 årsjubileum. För att fira detta riktigt ordentligt och tacka för den tid som varit, bjöd vi in ett antal av våra kunder på fest. God mat och gott vin serverades på restaurang "Fredsgatan 12" i Stockholm och "28+" i Göteborg, där vi dessutom utbildades i vinprovningens ädla konst.



Mingel från Stockholm (fr. vänster): Torgny Hållstrand, Mikael Sjöberg, Per-Otto Walter, Staffan Hedlund, Torbjörn Kloow, Alf Olsson.



Vinprovning i Göteborg (fr vänster): Anders Granhäll, Erik Bäckström, Hans Håkansson, Anders Rylander, Peter Granfeldt, Tomas Jerson.

Förutom mingel och vin serverades fantastiskt god mat och jag vill ge stort beröm till kockarna på Fredsgatan 12 och 28+, samt lärarna i vinprovning. För att utöka utbildningsdelen höll undertecknade en kort historiektion, där det bl a talades om:

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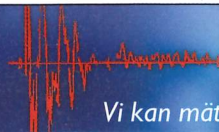
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Gothenburg office:
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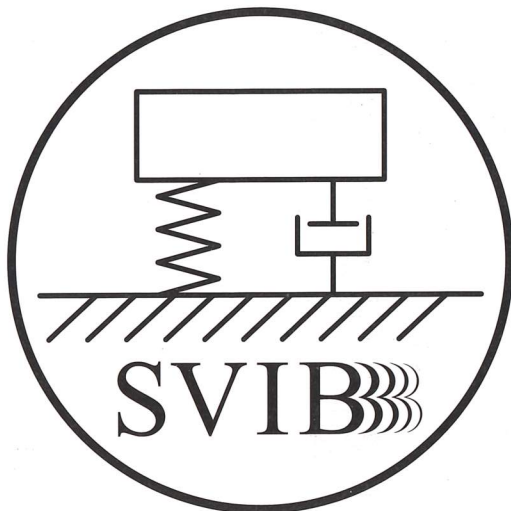
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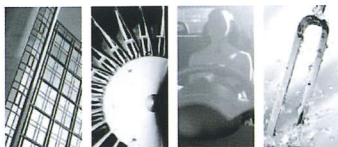
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