IDENTIFICATION FOR CONTROL SYSTEM DESIGN OF VEHICLE SUSPENSION

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ABSTRACT

One of the main trends of present days' modern vehicle production is the developing and correcting the vehicle's own convenience, in safety operating. One of segments is the suspension, since this can ensure the necessary conditions for other safety systems like ABS, ESP, steering system, etc. Simultaneously the suspension has to hold the body of the vehicle in comfort, solving that problem is still difficult. Nowadays the passive suspension systems can not fulfil these conflicting requirements anymore, it is becoming more urgent to introduce the active and the semi-active suspensions in the practical use. The direct contact with other safety systems makes the suspension a justified part of the X-by-wire system. This way we can include more functionality in the suspension, since we can access additional information for being used as input data for the active suspension system.

The knowledge of the passive behaviour of the suspension is the first step on the way for creating the best control solution. We must know the disadvantageous reactions in the different driving situations and the mathematical relations of the oscillation for generating a theoretic model. With the use of this model we can select the best system design without realizing the controlled system, so the faulty model's construction can be spared.

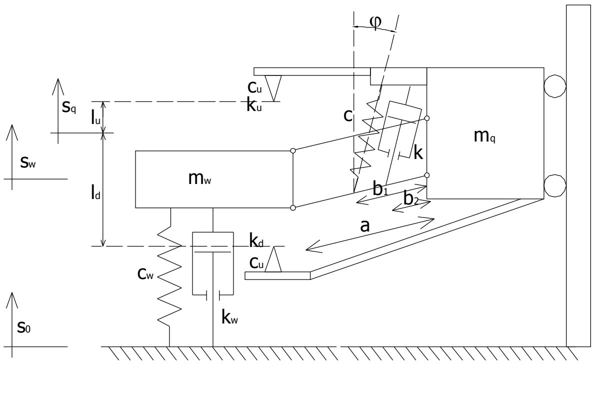

Figure 1 - The quarter-car model

This paper focuses on the steps of the identification of the passive suspension. For these steps we must execute our suspension analysis in laboratory conditions on a quarter-car model since the excitations have to be known

and the model must be simple and easy to survey. This model is shown on 'Figure 1'. There is a possibility that we can define the model with mechanical tools (white box model). The complexity of the problem lies in a few mechanical determinants on whose behalf we can not define the mechanical model precisely. In the quarter-car model we find more constraints and frictional force where the model is connected to the measuring system.
It's very difficult to model these influences with mechanical tools, so we should choose the statistical tools (black box model). With this method we can get enough information for a general identification, because based on the input and output signals relations stemming from sufficient measurement we can determine the probable output characteristics. Conversely the target of our identification is the control design of this system. We must specify a few components of this system because one of these parts will be replaced by an active element. We have to know the effect of these parts and we must simulate the process of its parameter in time. We can define some parameters like masses, distances, etc. relatively easy as well.
This way we must find the intermediate state between the two methods. In our procedure we analyse the system's response excited by simple statistical signals. From these input-output values we can estimate the transfer function of different model structures. The mechanical description will help us to define the correct model structure and to identify the connection between the physical parameters and the transfer function's ones. Thus we get the parameters of the system's elements concerning this quarter-car construction.

Keywords: suspension, identification, simulation, modelling, parameter estimation, quarter-car

1. INRODUCTION

The first step of every control design is the modelling of the target, the vehicle part and the simulation of it. The model created by this method is the base of control algorithm. In our case it's means the modelling of the suspension of vehicle. We establish a simulation for the modelling of the suspension where we can change the parameters of the elements. On the one hand in this case we can simulate suspensions with different element characteristics. On the second hand we have the possibility to change one of passive part to an active one and analysing its behaviour. To determine the parameters we have to identify our passive quarter-car.

2. THE MECHANICAL MODEL

As the first step of identification we have to set up a starting, mechanical model. This model will be the base of the final model contained the element parameters. In this task we simplify something. First we neglect the special geometrical structure of the given suspension. The cause of this is that we have to execute the identification steps independently of suspension's specialities. Second we discard arising friction is the system. This effect will be taken into consideration later. After these steps we get the 2DoF vibrating system as the starting base of our investigation shown in Figure 1.

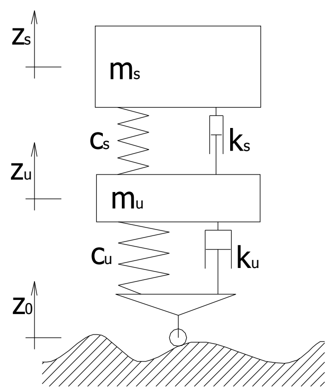

Figure 1: Quarter-car model

We take one by one the motion equations of the mechanical model. These equations contain the parameters that we want to define:

$$m_s \cdot \ddot{z}_s = k_s \cdot (\dot{z}_u - \dot{z}_s) + c_s \cdot (z_u - z_s)$$
$$m_u \cdot \ddot{z}_u = c_u \cdot (z_0 - z_u) + k_u \cdot (\dot{z}_0 - \dot{z}_u) - k_s \cdot (\dot{z}_u - \dot{z}_s) - c_s \cdot (z_u - z_s)$$

The goal is the expression of the displacements that are the variables of the differential equations. For realise this we use the L-transformation on the equations before. The results:

$$m_s \cdot Z_s \cdot s^2 = k_s \cdot (Z_u - Z_s) \cdot s + c_s \cdot (Z_u - Z_s)$$
$$m_u \cdot Z_u \cdot s^2 = c_u \cdot (Z_0 - Z_u) + k_u \cdot (Z_0 - Z_u) \cdot s - k_s \cdot (Z_u - Z_s) \cdot s - c_s \cdot (Z_u - Z_s)$$

Because:

$$G_{u0}(s) = \frac{Z_u}{Z_0} \qquad and \qquad G_{s0}(s) = \frac{Z_s}{Z_0}$$

The transfer functions:

$$G_{u0}(s) = \frac{Z_u}{Z_0} = \frac{b_3 \cdot s^3 + b_2 \cdot s^2 + b_1 \cdot s + b_0}{a_4 \cdot s^4 + a_3 \cdot s^3 + a_2 \cdot s^2 + a_1 \cdot s + a_0} \qquad , where$$

$$b_3 = k_u \cdot \frac{1}{m_u} \qquad b_2 = \left(\frac{k_u \cdot k_s}{m_s m_u} + c_u \frac{1}{m_u} \right)$$
$$b_1 = \left(\frac{c_u \cdot k_s}{m_s m_u} + \frac{c_s \cdot k_u}{m_u m_s} \right) \qquad b_0 = \frac{c_u \cdot c_s}{m_u m_s}$$
$$a_4 = 1 \qquad a_3 = k_u \frac{1}{m_u} + k_s \cdot \frac{1}{m_s} + k \cdot \frac{1}{m_s}$$
$$a_2 = c_u \frac{1}{m_u} + \frac{k_u \cdot k_s}{m_s m_u} + c_s \cdot \frac{1}{m_u} + c_s \cdot \frac{1}{m_s}$$
$$a_1 = \left(\frac{c_u \cdot k_s}{m_s m_u} + \frac{c_s \cdot k_u}{m_s m_q} \right) \qquad a_0 = \frac{c_u \cdot c_s}{m_u m_s}$$

$$G_{s0}(s) = \frac{Z_s}{Z_0} = \frac{b_2 \cdot s^2 + b_1 \cdot s + b_0}{a_4 \cdot s^4 + a_3 \cdot s^3 + a_2 \cdot s^2 + a_1 \cdot s + a_0} \qquad , where$$

$$b_2 = \frac{k_u \cdot k_s}{m_s m_u} \qquad b_1 = \left(\frac{c_u \cdot k_s}{m_s m_u} + \frac{c_s \cdot k_u}{m_u m_s} \right) \qquad b_0 = \frac{c_u \cdot c_s}{m_u m_s}$$
$$a_4 = 1 \qquad a_3 = k_u \frac{1}{m_u} + k_s \cdot \frac{1}{m_s} + k \cdot \frac{1}{m_s}$$
$$a_2 = c_u \frac{1}{m_u} + \frac{k_u \cdot k_s}{m_s m_u} + c_s \cdot \frac{1}{m_u} + c_s \cdot \frac{1}{m_s}$$
$$a_1 = \left(\frac{c_u \cdot k_s}{m_s m_u} + \frac{c_s \cdot k_u}{m_s m_q} \right) \qquad a_0 = \frac{c_u \cdot c_s}{m_u m_s}$$

It’s easily reasonable that we have enough, independent expression to define the element parameters. We have to find only the transfer parameters to calculate the model ones.

3. PARAMETER ESTIMATION OF TRANSFER FUNCTIONS

If we postulate our system an unknown one we have to imagine a so-called ‘Black-box’. Its input is the road function; its output is the displacement of any mass. In these cases the transfer function is definable from the output-answer of system exited by some input. The Matlab System Identification Toolbox is a very helpful tool for this task. We can analyse and transform the measured data and we can establish relation between them. The structure of transfer function can be estimated and its parameters can be defined with it.

In our case the system isn’t really unknown. With the motion equations we have given exact estimation for the transfer function’s structure before. In this way the two identification method can be connected to each other: We execute the black-box identification steps on the base of the transfer function’s structure of mechanical model. We want to reach with this that the neglected effects be integrated into the parameters of the mechanical model. Thus we get the fictive ‘in-system’ parameters of the elements that simplify the system model yet describe it correctly.

4. MEASUREMENT OF THE SYSTEM’S INPUT – OUTPUT DATA

We have to do some measurements on the quarter-car for the input data of identification. We generate stochastic road-function as the input excitation of system. We use the hydraulic pulsator of the Department of Automobiles for this task shown in Figure 2.

Figure 2: The hydraulic pulsator

It’s practical to measure the displacement of the wheel and the body because there is the road-wheel and the road-body transfer function in the spotlight of our analysis. Thus we set up inductive displacement sensors on the mentioned elements. For the verification we consume the accelerations too, so we have to register them. We use dSpace AutoBox to acquiring data for thinking to the future. This hardware will be good base of the control system design hardware. The construction of the measuring system is shown in Figure 3.

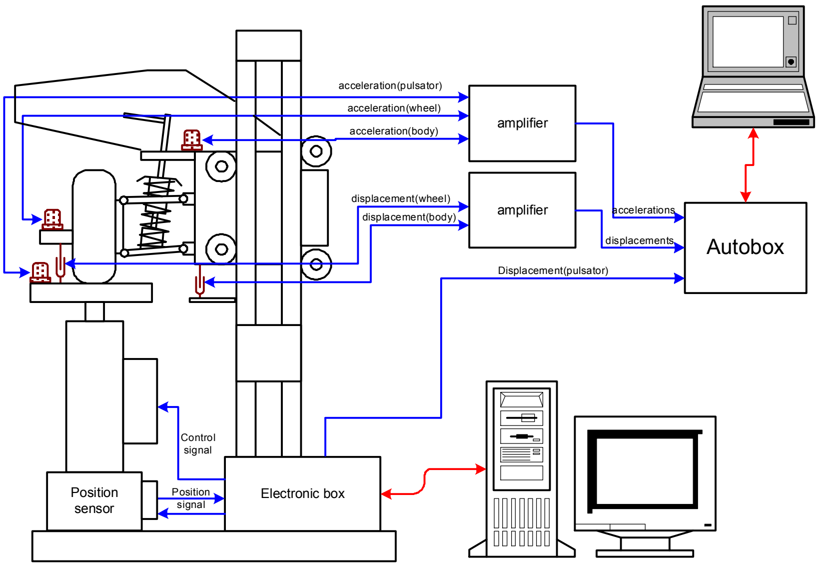

Figure 3: Measuring system

5. IDENTIFICATION INITIAL STEPS

Following we talk about the determination process of road-wheel transfer function. We will show the parameter estimation steps on this example.

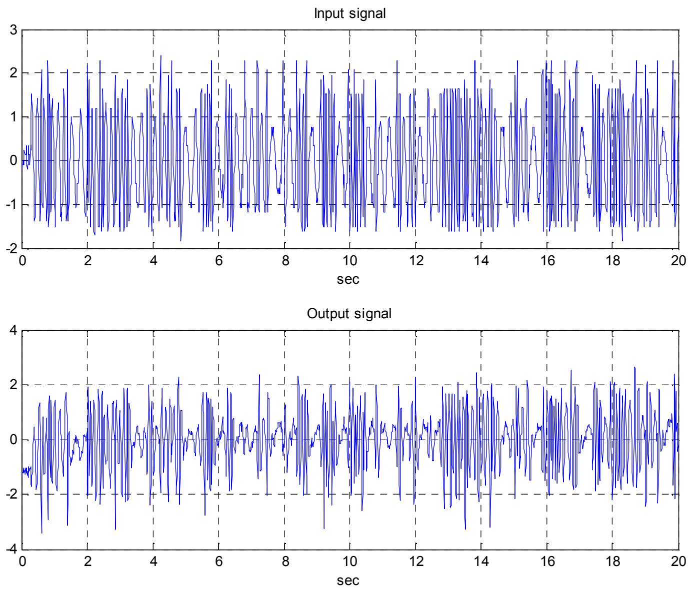

Figure 4: Input & output signal

This transfer function contains for itself enough independent equation that determine the mechanical parameters if we know the mass of the whole system. The

Figure 4 shows a displacement function of pulsator and wheel under a measurement. We can correct the model estimation if we condition the measured signals with some transformation steps. In this way we normalise the collected values and correct the possible offset errors.

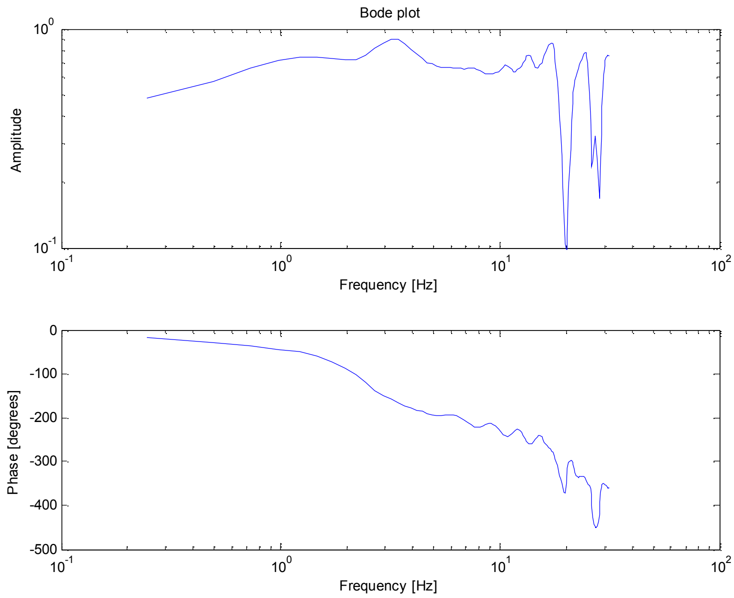

Figure 5: Frequency functions

In the Figure 5 we examine the frequency functions of measured system. We can see the dominant frequencies where the gain of the system is high.

6. STRUCTURE ESTIMATION OF TRANSFER FUNCTION

The mechanical description give us estimation for the road-wheel transfer function's structure: ARMA(4,3).

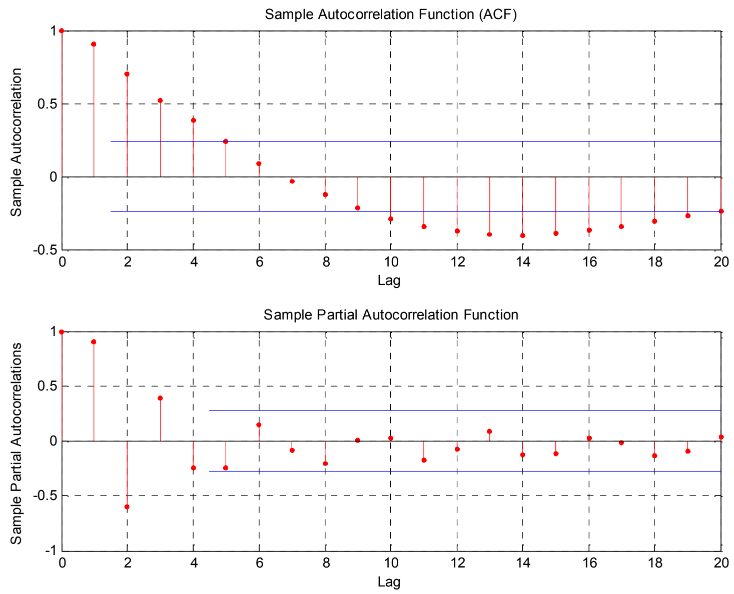

Figure 6: Correlation functions

This is true for the real model which is in its states in continuous-time. If we estimate the structure from measured data in the first step we can create the transfer function explained in discrete-time state. The structure of discrete function doesn't follow absolute the structure of continuous one. Because of this we have to estimate the structure with the help of other tool. The solution for this task is given by the auto-correlation and partial auto-correlation functions. These functions are shown in Figure 6. The auto-correlation function shows the order of the moving average part of the model. We see the values which calm down in the infinity, so the order of this part is zero. From the partial auto-correlation function we can read out the rank of autoregressive part. We can see four significant lag. On the basis of this we estimate our discrete model to an AR(4) model. This will be the base of the transfer function parameter estimation. Using the System Identification Toolbox we get the following results:

$$G_{u0}(z) = \frac{0.13585}{z^4 + 1.6574z^3 + 1.4799z^2 + 0.99914z + 0.29702}$$

Converted into continuous-time state:

$$G_{u0}(s) = \frac{0.567s^3 + 22.6081s^2 + 404.1454s + 2889.3704}{s^4 + 12.1397s^3 + 248.1937s^2 + 1563.409s + 2558.906}$$

7. COMPARISON

We put the measured and the modelled outputs got by exciting the model with the same input function into the same figure. This model prediction and its error is shown in Figure 7.

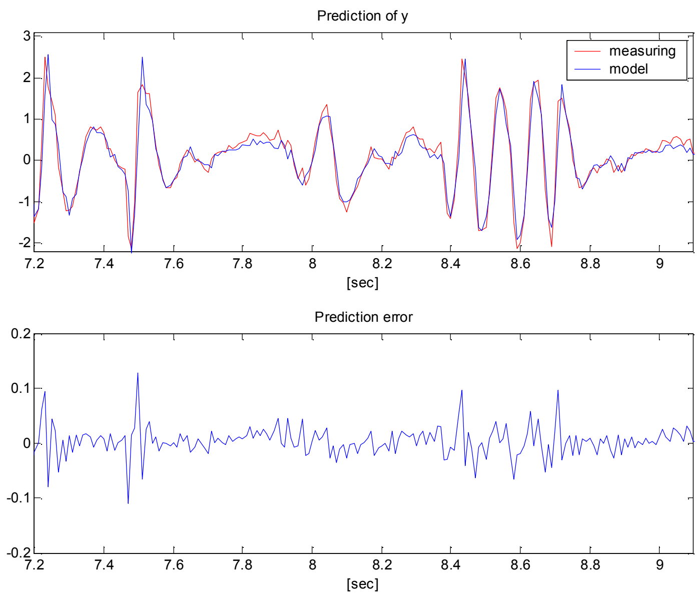

Figure 7: Outputs and prediction error

We can do better comparison with the frequency functions shown in Figure 8.

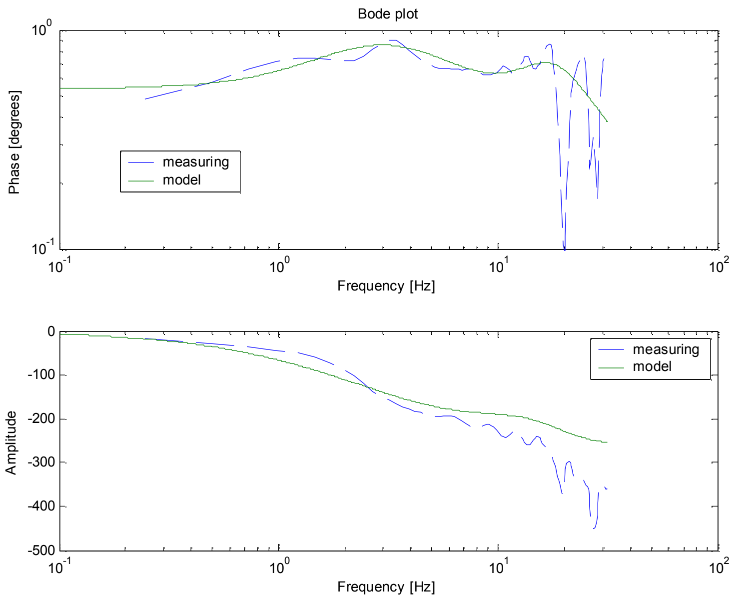

Figure 8: Bode plot of measuring & model

We can see that the model approach suitably the measured data. So we can accept the estimated parameters. The differences in at the high frequencies came from the measuring system, not from the quarter-car motion. Naturally we can refine the transfer function parameters with additional measurements.

8. CALCULATION OF MODEL PARAMETERS

The simultaneous equations came from the mechanical description determine exactly the elements' parameters of the quarter-car. We can express these parameters and in this way we get the following relations:

$$A = \frac{(b_1 - b_0)\cdot(a_3 - b_3)\cdot b_3}{a_2 - b_2}$$

$$B = \frac{(b_2 - b_0)\cdot b_3}{A}$$

$$C = a_3 - B - b_3$$

$$m_u = \frac{M}{B + C}$$

$$m_s = M - m_u$$

$$k_u = m_u \cdot b_3$$

$$k_s = \frac{m_u \cdot m_s \cdot (a_3 - b_3)}{m_u + m_s}$$

$$c_u = \frac{b_2 \cdot m_u - k_u \cdot k_s}{m_s}$$

$$c_s = \frac{b_0 \cdot m_s \cdot m_u}{c_u}$$

We used three additional variables (A, B, C) to simplify the calculations. If we substitute the estimated transfer function parameters:

$$
\begin{aligned}
m_u &= 23.6796 \quad \mathrm{kg} \\
m_s &= 256.3204 \quad \mathrm{kg} \\
k_u &= 13.4264 \quad \frac{\mathrm{N \cdot s}}{\mathrm{m}} \\
k_s &= 5462.0491 \quad \frac{\mathrm{N \cdot s}}{\mathrm{m}} \\
c_u &= 33072.5 \quad \frac{\mathrm{N}}{\mathrm{m}} \\
c_s &= 5212.2092 \quad \frac{\mathrm{N}}{\mathrm{m}}
\end{aligned}
$$

9. SIMULATION

With the got parameters we can design a simulation model. There are a lot of tool to solve the motion equations came from the mechanical description. Also simulation procedure exists based on numerical methods. This calculation has a few advantages. With the Matlab SimMechanics Toolbox we can draw profit from these advantages. Its essence is that we don't have to describe our system with the analytical relations. There are defined elements like body, joint, etc. which can be connected to each-other on the basis of the geometric terms. With these simple elements the building of the 2 DoF linear vibration is very easy work. The Figure 9 shows the model:

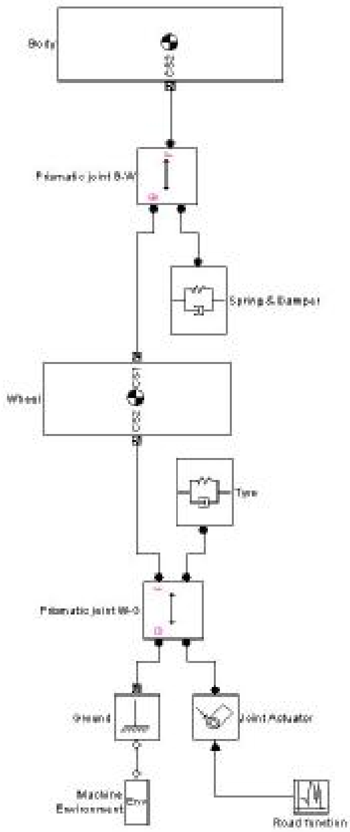

Figure 9: The Simmechanics model

We used the measured data acquired in the procedure of identification for the validation of the simulation. Thus the displacement of the wheel is formed according to the values Figure 10 in the cases of model and measurement.

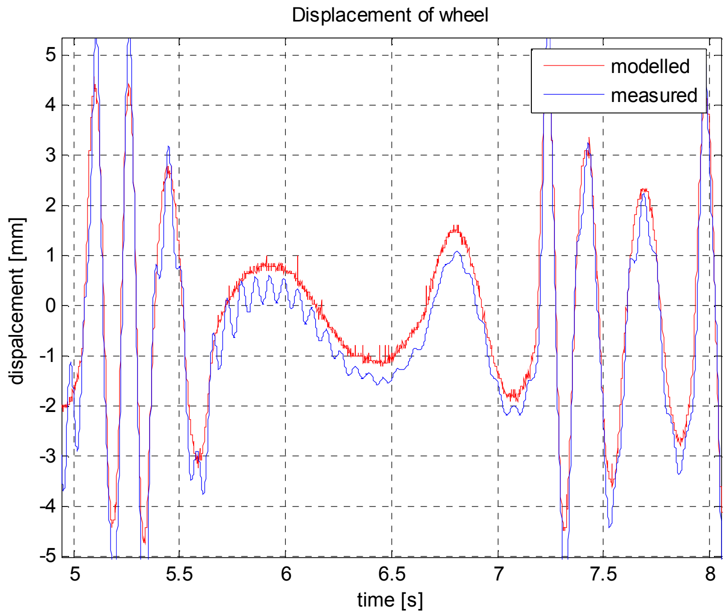

Figure 10: Displacement of wheel

We can see that from the viewpoint the tendency the model follows satisfactorily the real displacements of the quarter-car. But we can observe that the model gives less eased displacements. We have to look for the cause of this effect in the model parameters. If we execute the identification steps in many cases we could select the more convenient parameters by the help of the model. For the better comparison we created the frequency functions too from the displacements shown in Figure 11.

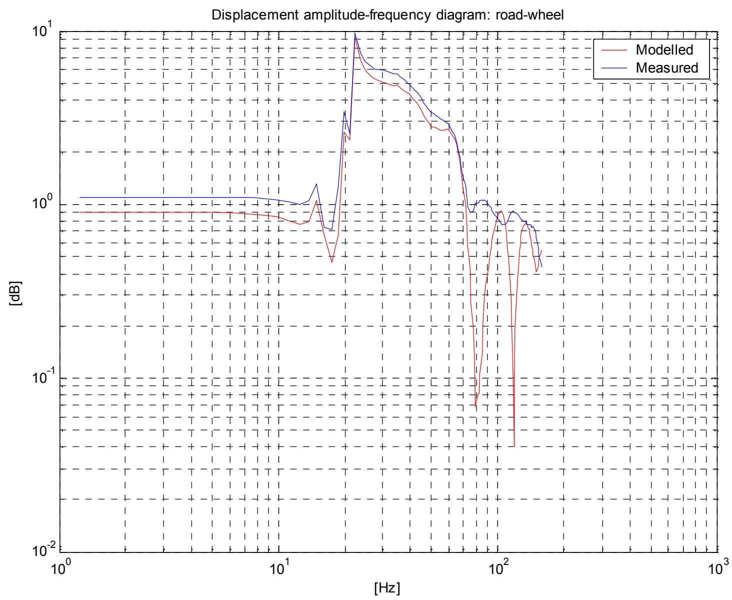

Figure 11: Road-wheel displacement frequency diagram

According to the figure the dominant transfer factors appear at the same frequencies with the same values. We can neglect the differences around 100 Hz because this interval is out from the functional vibration range of the quarter-car's suspension. We can take the same establishing from the viewpoint of the accelerations too. The transfer diagram of accelerations is shown in Figure 12.

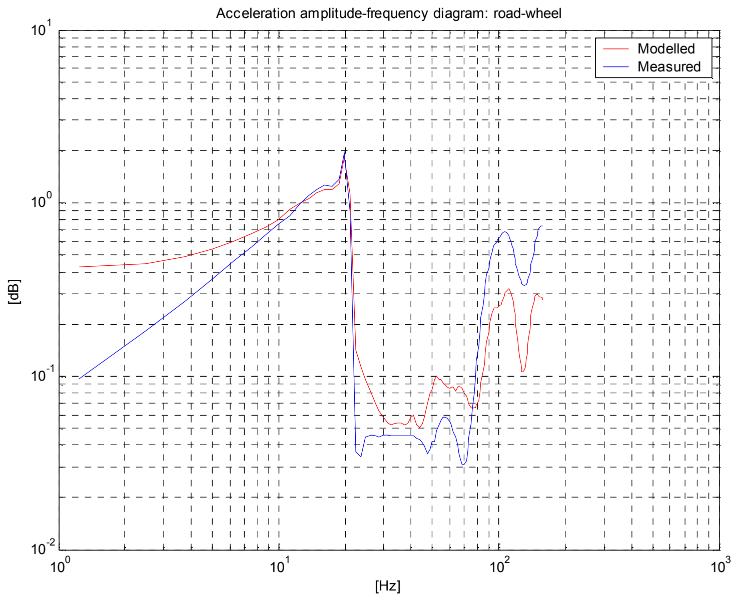

Figure 12: Road-wheel acceleration frequency diagram

10. CONCLUSIONS

According to this description we have shown how we can execute the modelling steps of a system in these conditions. It means that we wanted to identify the element parameters of system which are analysed on the basis of its behaviour in the system. Though we neglected a few effects in the procedure of the model building, we have got yet totally satisfactory results. The substance of this method is exactly this advantage: For the simulation it defines model parameters which can reproduce the undescribed, neglected effects too.

11. REFERENCES

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