DYNAMICAL MEASUREMENTS ON THE 1 : 2 SCALE MODEL OF THE BALLAST RAILWAY TRACK

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SUMMARY

Nowadays a lot of railway-vehicle system models were constructed to determine theoretically characteristics of the track (the deflection, the contact force, vibration propagation etc., see in [2, 3, and 4]). Of course these models have a lot of problems with uncertain input data (e.g.: damping and elastic coefficients), i.e. their results are uncertain too.
So it has decided to carry out an experimental research for better understanding the characteristics of the track system and its elements, and for determining the above mentioned data to use them later in a theoretical model.
The measurements were done by accelerometers on the geometrically 1: 2 proportional track-model. The signals were recorded analytically and after converting them into digital data they were evaluated by a software program. The Furrier spectra of the track and its element are available. The functions were compared with real experimental data to verify and to transform them into a real scale.

Keywords: Ballast track-model, Experimental measurements, Furrier transformation

1. INTRODUCTION

The measurements were done in the laboratory of the Railway Construction Department. Of course a real 1: 1 scale measurements proposes better possibilities to measure track parameters but as a reason of too little space there, it was built a 1: 2 proportional model.
Using accelerometers for determining track parameters are well-known method because after analyzing, and visualizing the acceleration spectra, they bear most of the above mentioned track parameters.
The main aims were to determine track element characteristics like damping, elasticity coefficients and resonance frequencies. Also how the acceleration is depending on varied track conditions such as static load.
The other point of the research was to calculate out a track-system equation, which gives an answer of the track for an applying input force but without modal testing, which was not available, it was not really capable. With such an equation it is possible to determine the applying force from the vehicle to the track from real measurements accelerations, which is still unknown for researchers but very important element of all the theoretical models.
But with the assumption that the system is linear at the first step, it was given the function of the track-model system.

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2. CONSTRUCTION OF THE MODEL

2.1 Description of the model

The conditions of the laboratory were not allowed to build a part of a real track as mentioned above so the measured model elements were the next:

- rail (7 kg/m) 2.5m long
- 8 concrete sleepers
- ballast (basalt) 0.15m thick under the sleeper
- gravel-sand 0.10m thick
- ballast mat 0.01m thick

The comparison of the real and the model track parameters are placed in tab. 1. It shows the differences between the two systems, which is one of the reasons for the deviation between real and model results.

Type	Rail	Sleeper	Pad (steel)	Fastening	Ballast	Frost protection layer
54-es	54-es	LM 14x111x360	Ribbed 14x111x360	GEO	30/65	Gravel-sand
Model	7-es	2K 80x120x1000	Plain 5x100x145	Rail-screw	12/35	Gravel-sand

Table 1. The track and the model elements

The model were surrounded by steel beams (fig. 1) and placed on the asphalt of the laboratory right under the pulsing machine. The box was 1.20x2.50x0.30m large and the above mentioned elements were put in after each other.

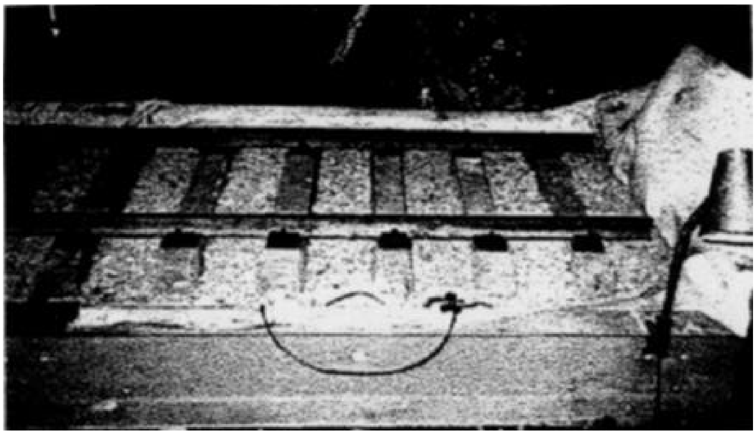


Figure 1. General picture of the track-model

2.2 Construction

First the ballast mat was placed into the box which was supposed to decrease the boundary effect of the surrounding beams.
After the gravel-sand was put which had the same mature such as a frost protection layer used in reality. It was compacted well with a vibrato machine.
On the surface of the sand were placed three accelerometers (1, 2, and 3, SETRA type) to measure ballast acceleration (fig. 2).
It was followed by ballast placing and compacting in 0,15m thickness.
The rails with the 8 sleepers were fitted together out of the box and later were lifted into the box.

Before filling out the sleepers gaps the forth accelerometer (also SETRA) was attached to the bottom of the loaded sleepers. After filling out the gaps and compacting them by heavy hammers another five (5, 6, 7, 8, 9) accelerometer base (steel pad, described later) were stiffed, three on the sleepers and two on the rail. The measured points are shown on fig. 2. The rest three accelerometers were BRUEL&KJAER.

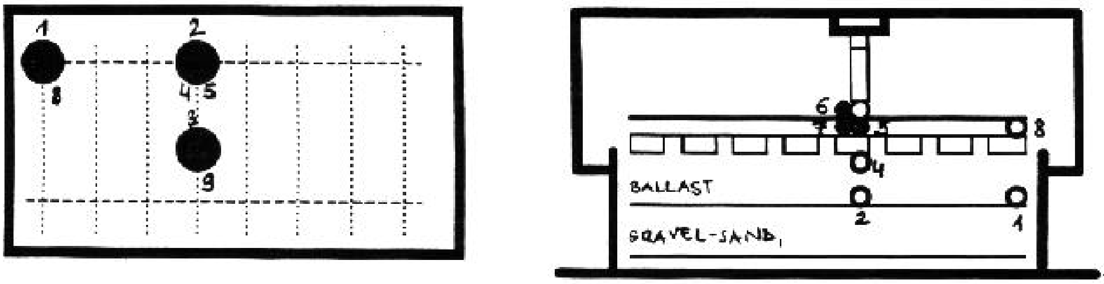


Figure 2. Measured points in the track-model

2.3 Transducers and measuring system

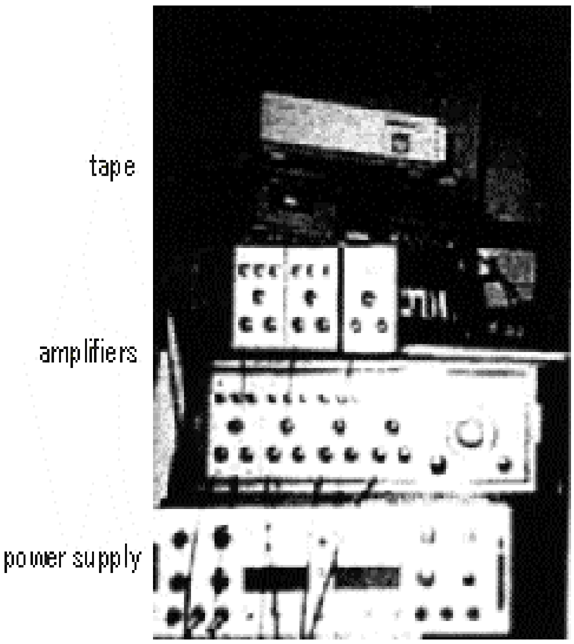


Figure 3. Measuring system

1-4. Accelerometers were SETRA and they were exited with a power supply at 10V. AC. Against of this each was connected to an amplifier like the rest three sensor to get sensible signals for the recorder. All of them were screwed to a steel pad 4x4x0.1cm, which stabilized and well fitted them to a measuring point. SETRA-s against ballast grains tortures was covered by a plastic helmet.

5-7. Sensors were BRUEL&KJAER and they were movable as the reason of their screw-head facilities between the 5 steel base.

From the amplifiers the signals were conduit to the analytic recorder (fig. 3).

2.4 Loading equipment

A 40 years old MTS pulsing machines with 250kN work-cylinder, which force characteristic curves are shown on the fig. 4, did the dynamical loading. It was decided to use cascading sinus sign because that was fairly well produced by the machine and the trains are causing periodic dynamic loads to the track too.

So the loading force was:

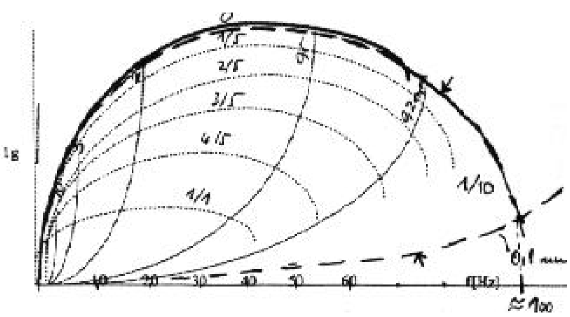
$$F_{(t)} = F_0 + F_{dyn} \sin(2\pi f\ t) \quad (1),$$

where F_0 is the static load which was varied during the measurement 20kN and 25kN

F_{dyn} was the dynamic force 2kN

f was the loading frequency from 5Hz to 115Hz stepping by 5Hz
To make a point contact between the force and the rail, a part of the rail were attached to the head of the MTS. The load applied above the middle sleeper's center (fig.5).

Figure 4. MTS loading characteristic



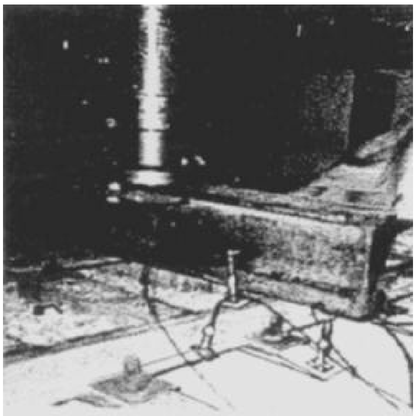


Figure 5. The model during loading with three accelerometers.

3. MEASUREMENTS

3.1 Calculation of the model properties.

After the geometric proportion 1: 2 the next changes appear:

- for the mass $M = M_m\ 2^3$
- static wheels load after the assumption that $y := 2y_m$ the static deflection calculated by the method Zimmermann (1. Annex) for F = 112.5kN (used by ORE) in the model F_0 = 21.598kN so F_0 = 20kN and 25kN were applied.
- for the frequency $f = f_m$ and for the time $t = t_m$
- for the speed $v = v_m\ 2$ and so on for the acceleration $a = a_m\ 2$

3.2 The measure

As mentioned before that the measurement were done step by step from each frequency and with different loads and so on different measuring points.
In the first case with 20kN static load the three movable sensors were after each other: top of the rail (6), bottom of the rail (7) and at the end of the loaded sleeper (5) (fig. 5). After load was increased to 25kN.
In the second case the accelerometers were replaced so the (5) was moved to the end of the last sleeper (8); and from the bottom of the rail (7) was moved to the middle of the loaded sleeper (9) (fig. 3). Here also 20 and 25kN force was applied.
By MTS was measured also the deflections of the track with static load and so on during the dynamic loading.

4. DATA EVALUATION

4.1 Evaluation.

The recorded signals were converted into digital data by a TEKTRONIX analyzer so it had become readable by a p.c. and eventually also with an TEKTRONIX software program the time functions and spectra were created.

The desired functions were the Furrier spectra of each measured point, which shows the vibration level of that point. But as a reason of the cascading force it was necessary to construct these functions manually because the software gave only one point of the whole acceleration spectrum for each frequency (fig. 6).
Transfer functions were created on the same way, which shows the damping effect between two measured points (fig. 8).

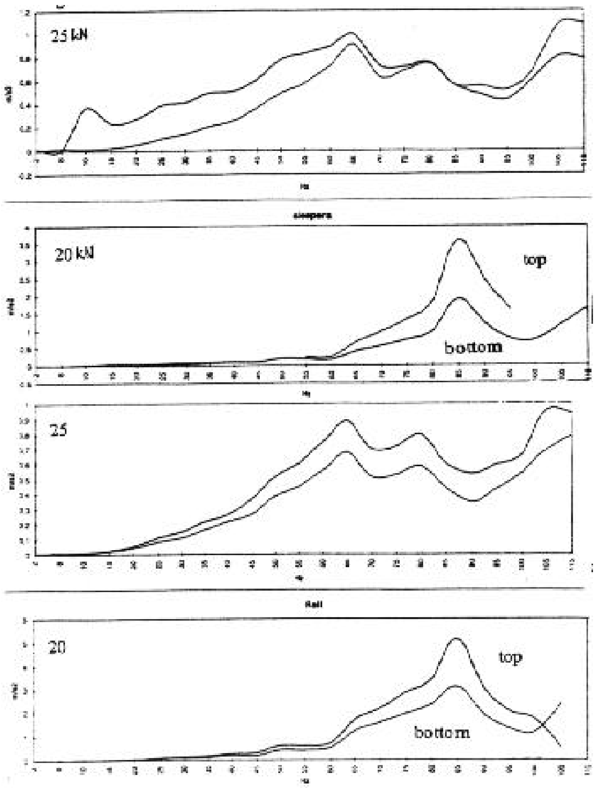
4.2 Remarks to the measurements

It is necessary to note that the model was not exact 1: 2 scale reduction of the real track as it is shown in tab.1 i.e. the fastening system was different because of the financial conditions.
Other notations for the loading machine that it was not able to produce exact sinus waves over 100Hz, especially (the reason for this case visible on the fig. 3).
For low frequencies the machine has made a lot of noise itself i.e. till 15Hz it was not possible to evaluate the transducers signals.
At 50Hz was not correct signals for the SETRA-s as a reason of that this is the resonance frequency of the currant network in Hungary.

5. RESULTS

The Furrier transform functions: They are visualized in the fig. 6. All the three have frequency peaks at 65Hz for 25kN and 85Hz for 20kN static load. The functions tell us that the loading force has a strong influence on the acceleration of track elements. Also it has a difference between measured signals on the top and on the bottom of the elements as we expected. The Japanese results (fig. 7) not very clear in the spectra under 150Hz so it is hard to compare with the model results.

Figure 6. Furrier transform functions of the elements (rail, sleeper)



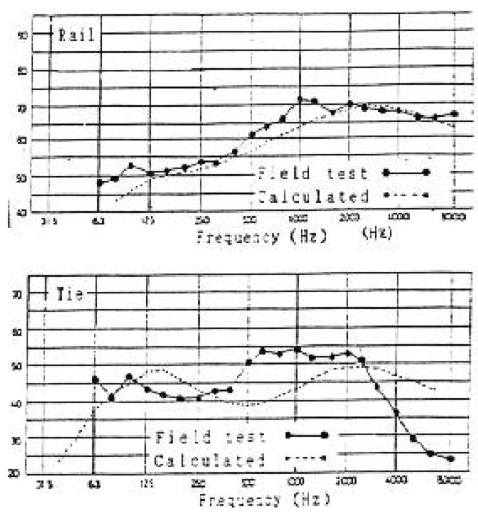
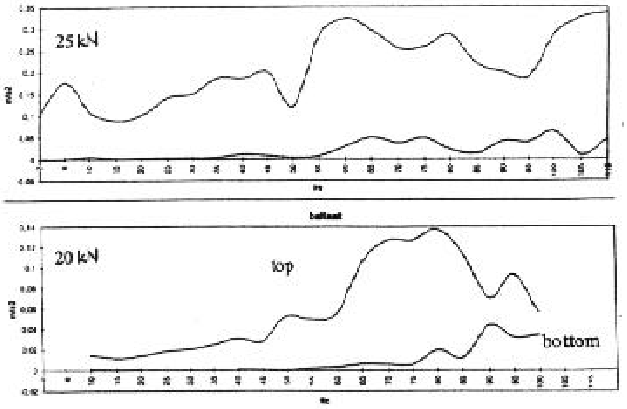


Figure 7. Japanese real measurement results [5]



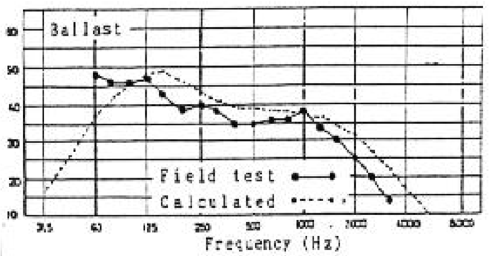


Figure 6. Furrier transform functions of the elements (ballast).

Figure 7. Japanese real measurement results [5]

The transfer functions of the elements (fig. 8): The rail does not show any resonance frequency such like the sleepers but its possible to estimate the resonance frequencies from the phase diagrams and this is for the rail between 600-700Hz, and for the sleepers it is more punctual between 145-155Hz. The damping effect between the top and the bottom of elements is about 5dB.

On the ballast transfer function are also not visible resonance as a reason of the low frequencies were too noisy to evaluate them, as it is written in the remarks.

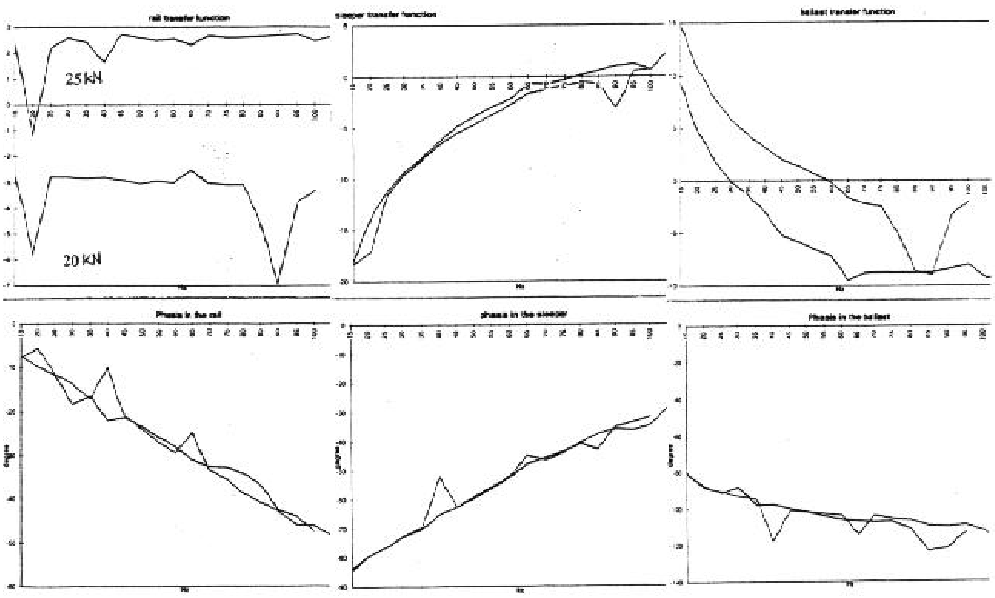


Figure 8. Transfer functions of track element and between elements.(damping functions)

Deflection of the track: It was $y_{average}$ = 1.7mm under the static load but during the dynamical loading it was increased to $y_{average}$ = 3.9mm. It shows similarities with other measurement which mention that the ballast begins to flow at a certain frequency, about 35Hz.

Track-model system equation: After producing the Furrier transform of the applied force it is possible to make the equation of the system by dividing the input force and the measured acceleration signal. So: $F_{(t)\ldots} \rightarrow \ldots \Gamma_{(\omega)}$

$$H_{(\omega)} = A_{(\omega)} / \Gamma_{(\omega)}$$

This function is shown on the fig. 9. With this system function it is possible to do estimations for the applying force only.

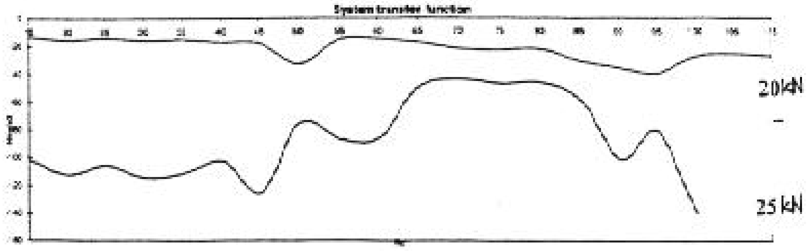


Figure 9. Track-model system transfer function

6. CONCLUSIONS

The track dynamical loading measurements were done by accelerometers in the narrow frequency range due to weak technical conditions which was not enough to determine precisely the resonance frequencies of the model elements. But estimating them and the system function were shown with the damping functions, too, which can be input data for further theoretical models.

In further investigations are necessary to extend the frequency range and apply a modal testing equipment to get exact results. From the presently available results can be calculated by integration PSD and ESD to show vibration energy of elements, which are important for track stability and vibration propagation investigations. Also, by dividing with ω deflection and velocity functions are computable.

7. ACKNOWLEDGEMENTS

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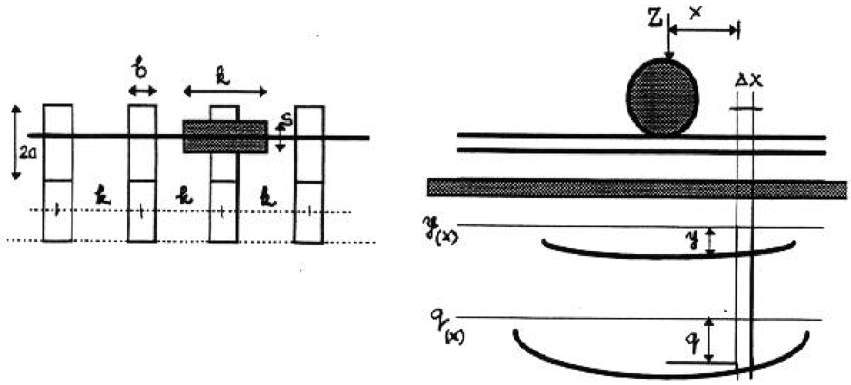
1. ANNEX

The theory of Zimmermann:

The theory calculates track strains based on the assumption that the infinite rail lay on the Winkler foundation where the sleepers replaced by a horizontal beam.



From the theory of Winkler:

$$p = y\,C,$$

where p - is a pressure applies to the foundation

y – is the settlement

C – is the elastic coefficient of the foundation

This gives for the case of the horizontal sleepers:

$$q = y\,C\,s$$

here q – strain applies to the foundation

s – is the length of the replacing beam: $s = 2\,a\,b / k$

To obtain the settlement the shear force calculated:

$$dZ / dx = y\,C\,s$$

and coupled with the equation of the elastic string:

$$M / EI = (-1 / \rho)(d^4y / dx^4)$$

From these two apply:

$$-y = EI / C\,s\,(d^4y / dx^4)$$

After introducing the constant L

$$L = \sqrt[4]{(4\,EI) / (C\,s)}$$

And with a new unknown ξ: $= x / L$, so the settlement differential equation obtains

$$-4\,y = d^4\,y / d\xi^4$$

We search for the solution in the form:

$$y = K\,(\sin\xi + \cos\xi) / e^{\xi} = K\,\eta$$

To determine K using estimation for the static case the equation is

$$Z / 2 = C\,s\,{}_0\!\int^{\infty} y\,d\xi = C\,s\,L\,K\,{}_0\!\int^{\infty} \eta\,d\xi$$

As the reason of ${}_0\!\int^{\infty} \eta\,d\xi$ is equal to one, so for K

$$K = Z / (2\,C\,s\,L)$$

And finally the settlement:

$$y = (Z / 2\,C\,s\,L)\,\eta$$

From this equation for the case that we search the deflection under the applying force, so $\eta = 1$ easily can calculate the static mass (Z) for the model.