A method for measuring the dynamic behaviour of the Rome–Florence high-speed track switch manoeuvring system

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A mathematical model of the dynamic loads induced at a switch machine and at the rails by the passing of a train was formulated and validated by physical tests. Attention was focused specifically on the switch machine itself and therefore the data gathered refers strictly to the attachment elements of the switch machine to the track and locally the track in the area of attachment.

The research is part of a series of projects that aim to establish the factors that must be considered during the design of a switch machine. An accurate mathematical model of the connection elements between the track and the switch machine is helpful, especially in this period of widespread adoption of high-speed trains and the new working conditions that this imposes, in order to ascertain the characteristics of the accelerations transmitted to the switch machine and, as a consequence, to design the connection system of the machine to the track, based on these criteria. The data acquired can also be used for the construction of a realistic test bench to further improve test procedures. The pursuit of this project is a fundamental step in understanding the factors involved in the design of a switch machine.

NOTATION

$a(f)$	acceleration (in the frequency domain)
$a_{2:1}$	a_2 component coherent with a_1
$a_{2.1}$	a_2 component non-coherent with a_1
$F(f)$	force (in the frequency domain)
$G_{xx}(f)$	auto power spectrum (autospectrum) of a signal
H_{xy}	frequency response function (FRF)
$x(f)$	system input (in the frequency domain)
$y(f)$	system output (in the frequency domain)
$\gamma_{xy}^2(f)$	ordinary coherence function between two signals

1 INTRODUCTION

The aim of this work is to determine the dynamic loads induced at the switch machine and at the rails of a train switch by passing trains. To dynamically represent the switch, appropriate measurements of acceleration and of frequency response were obtained at a set of characteristic points. By combining these measurements it was possible to estimate the loads induced at the rails and at the switch machine. The model was checked by comparing the estimated values with the corresponding measured values. This approach is valid when it is supposed that to know the presumable forces acting on the switch machine is sufficient to know the acceleration of the rails during the passage of the train and the frequency response of the chassis, the former being easily obtained by measurements on the line, the latter by laboratory tests.

The Rome–Florence High Speed Italian Railway line is used by ordinary trains and high-speed trains, that is with a maximum axle load of 20 t/axle, capable of reaching a maximum speed of 200 km/h (125 mile/h) and by lighter trains of 14 t/axle, with a maximum speed of 250 km/h (155 mile/h). It is expected that in the future trains of 20 t/axle will pass at 300 km/h (185 mile/h). To ensure the best working conditions and reliability of the mechanical parts, it is necessary to minimize the overstressing both of the rolling stock and of the permanent way. This is obtained, for example, by welding the rails, by appropriate inclination of the track in curves and by the use of ballast of specific characteristics.

The switches usually installed are commanded electromechanically with a tangency of between 0.04 and 0.12. Even if the Italian State Railways forsee the installation of new electrohydraulic switches for high-speed lines having a tangency of 0.022, it remains of fundamental inportance to establish the functioning characteristics of switches already in use and of those with a superior tangency that will be operated in a traditional manner.

The switches represent one of the most accentuated discontinuities that a train comes across and therefore the measuring of the dynamic loads at the switches assumes a fundamental importance, which also concerns the durability of the switch and train components and in particular the definition of the dynamic load on the parts that drive, control and anchor the switch machine and the point lock. During the design process it is extremely useful to be able to estimate these loads, once the forces of excitation are known, and to be able to introduce eventual modifications without having to measure the effects directly on the railway each time. The definition of experimental methods for the execution of appropriate dynamic tests is also important to set new safety, wear and stress resistance coefficients. Therefore a dynamic numerical model was created to represent the portion of the switch to which the switch machine is rigidly fastened through a metallic chassis.

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This procedure allows for both the optimization of the switch machine design with respect to dynamic behaviour and also representation of the train–track interaction at high speed.

2 THE DYNAMIC MODEL OF THE SWITCH

A railway switch consists of a series of steel parts (tracks, tongues, switch machine, lock and connecting rods) connected to each other in part through a rigid metallic chassis and in part through the sleepers. Everything is laid on a ballast.

To estimate the dynamic loads it is necessary to construct an experimental numerical model that utilizes the linear systems theory.

2.1 Basics of the linear systems theory

Single input–single output linear systems can be described in the frequency domain by the frequency response function H_{xy}, defined as

$$y(f) = H_{xy}(f)x(f) \tag{1}$$

where $y(f)$ represents the output of the system and $x(f)$ the input. For pairs of records extracted from two different processes it is possible to define many statistical functions indicating the connection between two signals. Assuming that the autospectral densities of both signals are not equal to zero and that their mean value is null, the coherence function $\gamma_{xy}^2(f)$ indicates the linear dependence of the two signals. Inversely $\{1 - \gamma_{xy}^2(f)\}$ is a measure of how much the output does not depend linearly on the input at frequency f. If the system is linear and there is no noise the coherence function is equal to 1. If the input and the output are completely independent the coherence function is zero. The coherence function is greater than zero and less than one in the case of:

(a) external noise being present in the measurements,
(b) the system relating the input and the output not being linear,
(c) the output being a result of other input signals besides $x(t)$.

For multiple input–multiple output systems it is important to note that if the ordinary coherence function between two inputs, 1 and 2, is such that $0 < \gamma_{12}^2(f) < 1$, it means that $x_2(t)$ depends only in part on $x_1(t)$. If there is a correlation, the effect of $x_1(t)$ on $x_2(t)$, indicated as $x_{2:1}(t)$, must be subtracted from $x_2(t)$ to give the conditioned signal (or residual signal) $x_{2.1}(t)$ that represents the part of $x_2(t)$ that is not dependent on $x_1(t)$. In equations, $x_2(t)$ consists of the sum of two non-correlating terms:

$$x_2(t) = x_{2:1}(t) + x_{2.1}(t) \tag{2}$$

Transforming equation (2) in the frequency domain the following equation is obtained in terms of autospectra:

$$\begin{aligned} G_{22}(f) &= G_{22:1}(f) + G_{22.1}(f) \\ &= \gamma_{12}^2(f)G_{22}(f) + \{1 - \gamma_{12}^2(f)\}G_{22}(f) \end{aligned} \tag{3}$$

in which the first term on the right-hand side is called the coherent output spectrum and the second is called the output noise spectrum. In practice the case of null ordinary coherence between 1 and 2 exists since $x_{2:1}(t)$ is linearly dependent on $x_1(t)$ and therefore can be removed. If $\gamma_{12}^2(f) = 1$ it means that there is a complete linear dependence between the inputs; in this case it is only one of the inputs [for example $x_1(t)$] that excites the system through a transmission route passing also by $x_2(t)$; $x_2(t)$ can then be cancelled, reducing the number of inputs.

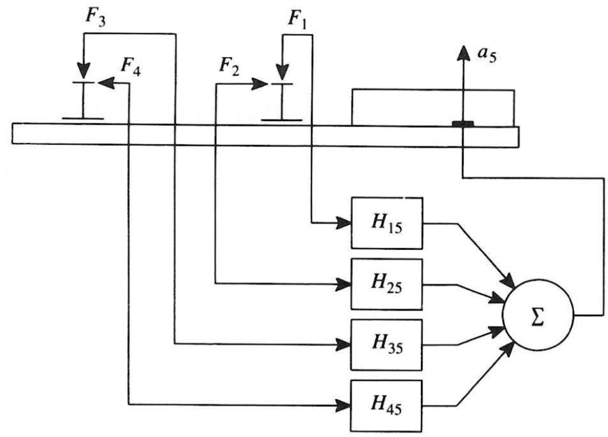

Fig. 1 Dynamic model of the switch

2.2 Estimation of dynamic loads on the switch machine based on the linear systems theory

As verified in test programmes, of which the results are shown in the following paragraphs, it is legitimate to take into consideration only the excitations recorded in a section near to the axis of the connecting rods. The physical system and the relative dynamic scheme are shown in Fig. 1.

The relations that tie the dynamic magnitudes are similar to equation (1) (assuming the acceleration measured on the switch machine to be the system output and assuming the forces caused by the trains on the track to be system inputs):

$$a_{i5}(f) = H_{i5}(f)F_i(f) \tag{4}$$

in the frequency domain where $i = 1, \ldots, 4$ represents the number of the input. The pairs of loads, vertical and horizontal, are correlated between themselves inasmuch as the axle produces a connection between the rails: it has been verified by experimental tests that the relative ordinary coherences are very high and so

$$\gamma_{13}^2(f) \approx 1 \quad \text{and} \quad \gamma_{24}^2(f) \approx 1 \tag{5}$$

As shown in the preceding paragraph the inputs 3 and 4 can be eliminated because they are linearly dependent on inputs 1 and 2. Then the system of Fig. 2 can be obtained, in which the only system inputs are F_1 and F_2.

Since a direct measurement of the forces generated by the trains on the track is not easy, these were estimated starting from the recorded accelerations on the track. For this purpose we used the definition of the frequency response function (FRF), equations (1) and (4). The FRF was determined with a contemporary measurement of force and of acceleration, as better explained in the following, and it was possible to estimate the forces after

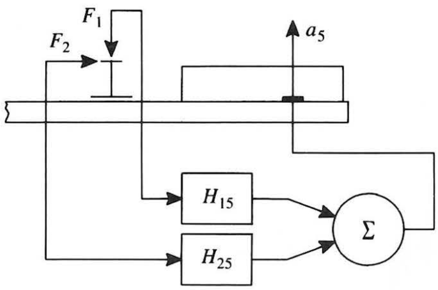

Fig. 2 Simplified model with two inputs

taking train acceleration measurements:

$$F_1(f) = \frac{a_1(f)}{H_{11}(f)} \tag{6}$$

The fundamental conditions for the execution of this operation are that the ordinary coherence of the actual measurements to obtain the FRF are near to one, thus showing a notable linearity of the system.

The acceleration signals were processed appropriately to determine the part of $a_2(t)$ that is non-coherent with $a_1(t)$ [see equation (2)] because the two inputs are partially correlated having $\gamma_{12}^2(f) > 0$. The vertical acceleration was chosen as reference at the point 1 because it is presumed to be the most directly responsible for a_5 and one eliminates the part of $a_2(t)$ coherent with $a_1(t)$.

It is possible at this point to assess the forces of excitation on the track, the accelerations and the force on the switching machine with the equations:

$$\begin{aligned} a_5 &= F_1(f)H_{15}(f) + F_{2.1}(f)H_{25}(f) \\ &= a_1(f)\frac{H_{15}(f)}{H_{11}(f)} + a_{2.1}(f)\frac{H_{25}(f)}{H_{22}(f)} \end{aligned} \tag{7}$$

$$F_5(f) = \frac{a_5(f)}{H_{55}(f)} \tag{8}$$

The complete sequence of the operations is synthesized in Fig. 3.

3 EXPERIMENTAL WORK

To establish a measuring technique and to acquire sufficient knowledge of the cause–effect relationships in the dynamic field, recordings were made on switches both on the line and in the laboratory following preliminary tests on the instrumentation. On the inside of the switch machine there are moving parts and safety critical components for which relatively high frequencies could be damaging; therefore all the analysis was carried out at up to 3200 Hz. Furthermore, as the following shows, this frequency range permits an optimum analysis of acceleration signals.

3.1 Preliminary tests on the line

The accelerations produced by the passing trains were firstly measured in correspondence with a switch with a tangent of 0.055. Four particularly significant measuring points were singled out at which accelerations were measured in a vertical and horizontal direction (normal to the track axis):

(a) switch machine,
(b) switch operating rod,
(c) switch blade,
(d) rail.

One of the more interesting results not shown here for reasons of brevity demonstrated that the position of the switch (correct path or diverging) was practically insignificant since trains having similar speed and composition produced vertical accelerations on the switch machine of the same order of magnitude. The speed of the trains was found to be important since trains that pass respectively at 60 and 110 km/h produced accelerations with a relative ratio of about 2 : 3. The transmission of the accelerations between distant points of the switch was practically null at these speeds. It can therefore be affirmed that in a given section of measurement the accelerations are measurable only at the passing over of the section by the train.

3.2 Preliminary laboratory tests

Measurements of transfer functions were also carried out between different points of a switch in the laboratories of the A. Siliani Plant at Florence. The tests were conducted supplying excitations through a 5 kg mass striking a load cell rigidly fixed to the track. The vertical and horizontal accelerations were measured through transducers fixed by means of a magnetic base. With this type of measurement it is possible to supply excitations of up to 50 kN.

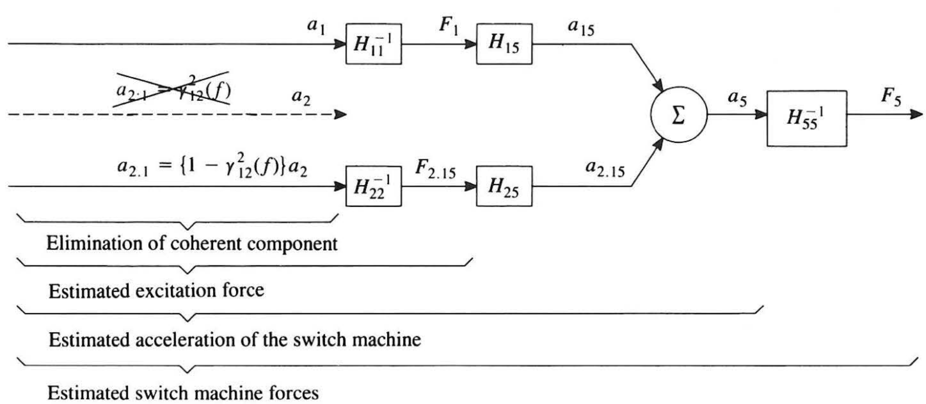

Fig. 3 Acceleration and force estimates

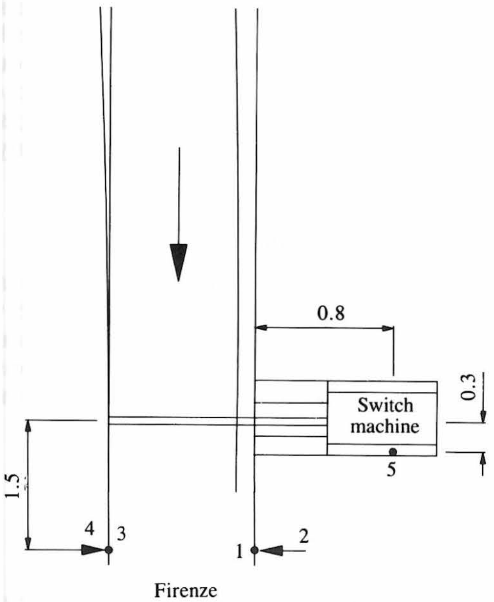

Fig. 4 Orvieto switch scheme

The excitation point was fixed while the response point was varied, moving the accelerometer to the various points considered.

It was observed that the difference between the mean FRF values measured at the load cell and a point 8 m away on the chassis was approximately 25 dB, which confirms the reduced transmission of the accelerations between distant points of the switch; also, the absence of pronounced peaks in the FRFs shows the presence of notable reductions due to the ballast damping.

3.3 Experimental tests conducted at Orvieto

The switch chosen, shown in Figs 4 and 5, is situated at 108.582 km along the Roma–Firenze up-line. The switch is normally taken in the correct path and from the rear of the turnout; it has a tangent of 0.034, can be taken at up to 100 km/h in deviation, at up to 250 km/h in the correct direction and is controlled by a P75 switching machine.

Measurements of acceleration levels were taken at points 1 to 5 (at the passing of the trains listed in Table 1) measurements of direct FRF (at points 1 to 4) and transfer FRF (from points 1 to 4 at point 5). Bruel and Kjær piezo-electric accelerometers and charge amplifiers were used to measure accelerations; an EMI six-channel analogue magnetic recorder was used for recording: an FFT Bruel and Kjær dual channel analyser was used to process measurement data; a Kistler load cell with its charge amplifier was used to measure the loads; a Hewlett-Packard model 217 computer with peripheral units was used for numeric data processing. The measurement chain was calibrated *in situ* with appropriate calibrators.

Accelerations caused by the trains in the five points of Fig. 4 were measured, conditioned and recorded on a six-track analogue magnetic recorder with a tape velocity sufficient to conduct analysis up to 3200 Hz. The accelerometers at points 1 and 2 were connected to a fishplate via threaded steel screws; the fishplate itself was clamped to the track. The accelerometers at points 3 and 4 were attached to the fishplate with plastic screws and mica washers to eliminate short circuits in the track circuitry caused by the simultaneous connection of the measurement points on the two rails. The accelerometer on the switch machine was connected through a magnetic base. The layout is shown in Fig. 6.

Fig. 5 View of Roma–Firenze up-line at PC Orvieto

The FRFs were measured by exciting the track with a specially instrumented hammer (Fig. 7) constructed for this purpose. Shock excitation was chosen in which the transducer was part of the hammer, enabling measurements on the line without having to interrupt the normal train traffic. The hammer was designed to be handy enough for prolonged use (striking mass of about

Table 1 Train characteristics

No.	Locomotive	Velocity km/h	Weight/axle tonnes
1	E447	205	20
2	E444	140	20
3	ETR450	245	14
4	E447	175	20
5	ETR220	180	16
6	E656	140	18
7	E447	160	20
8	ALe601	180	16
9	E447	195	20
10	E447	195	20

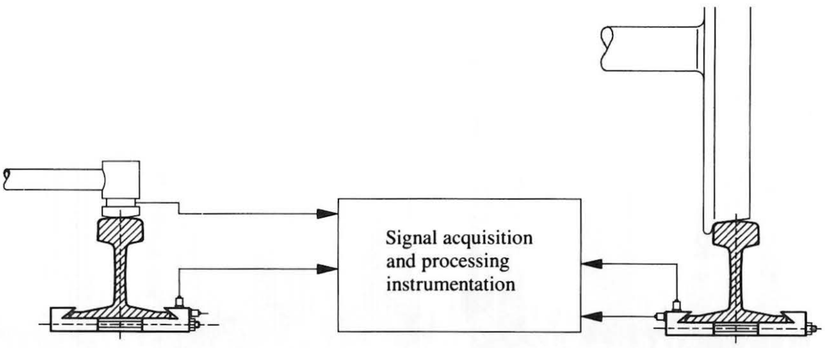

Fig. 6 Measurement scheme for FRF and train accelerations

4 kg) and relatively insensitive to the shearing caused by a force applied not perfectly orthogonal to the surface of impact. Its characteristics were checked by performing a reciprocal calibration with a Bruel and Kjær instrumented hammer. The measurement chain allows for the supplying of shock excitations of up to 50 kN.

The transfer functions are calculated by simultaneously measuring and transforming to the frequency domain force signals measured by the hammer at a point and of acceleration at the same point or at another point. Given that the excitation is deterministic and disregarding the noise in the measurements, the coherence indicates exclusively the repeatability of the measurement, but does not give indications of linearity. On the other hand, the suppliable forces are certainly

Fig. 7 Application of shock excitation on the track

lower than those produced by the trains and also the crest factor is comparable with that measured for the wagons. Therefore it is presumed that estimations of FRFs in the linear field were affected.

The accelerometer that measures the response was fixed at each point of measurement, as described in the previous paragraph. The acceleration and force signals, conditioned by charge amplifiers, were acquired in the 4–3200 Hz frequency band ($\Delta f = 4$ Hz) and processed by the analyser to determine FRFs and coherences.

4 PROCESSING OF THE DATA GATHERED AT ORVIETO

4.1 Processing of the acceleration signals

In Figs 8 and 9 the accelerations measured at point 1 of Fig. 4 following the passing of train 10 (E447 at 200 km/h) with different acquisition times are shown. It can be seen that these values are very high with an extremely abrupt fall-off which is completed in less than the elapsed time between the passing of the two axles. This is due to the damping of the ballast and as a consequence allowed for the testing of excitations caused by the passing of each single axle over the measurement section.

Of the acquired signals analysis it should be noted that, for very impulsive signals (those that present noticeable variations in a short time), the measured value depends greatly on the sample time that was chosen. The optimal value was determined as 0.122 ms, which corresponds to an end-of-scale frequency value of 3200 Hz.

The acquired signals were transformed into frequency by the analyser using the exponential window shown in Fig. 9 and were averaged on a number of significant axles not less than 7. The narrow-band autospectra obtained were stored on a disk and transformed via software into a $\frac{1}{3}$ octave band; the values obtained in the bands below 63 Hz are to be considered insignificant for the numerical methods adopted. The transformation into the $\frac{1}{3}$ octave band allows a smaller set of values to be obtained, energetically equivalent to a narrow-band spectrum, while simultaneously filtering the effect of isolated values. The estimates found are much faster and more stable from a numerical point of view bearing in mind the physical aspect of the problem.

Figure 10 shows the averaged autospectrum of the acceleration signals at point 1 for trains 3, 6 and 10. It

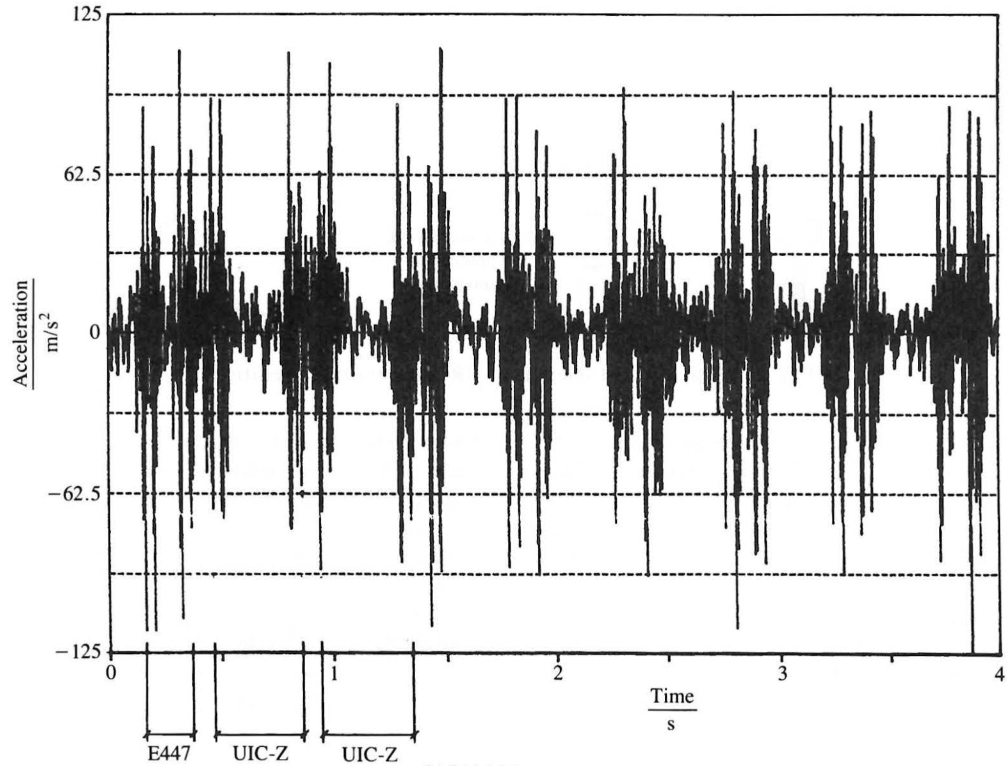

Fig. 8 Acceleration point 1 caused by train 10 ($T = 4$ s)

can be deduced from their analysis that:

1. At low frequencies (<250 Hz) the axle weight is very important in determining the levels of acceleration; however, these are practically independent of the speed of the train.
2. At higher frequencies (>250 Hz) the weight–speed combination assumes importance; train 10 has higher values (even though it does not have the maximum speed) because of its higher axle weight.

Train 10 therefore proves to be the most important in terms of energy levels and therefore all of the following analysis will refer to it. In Fig. 11 the acceleration spectra are shown for this train at points 1, 2 and 5.

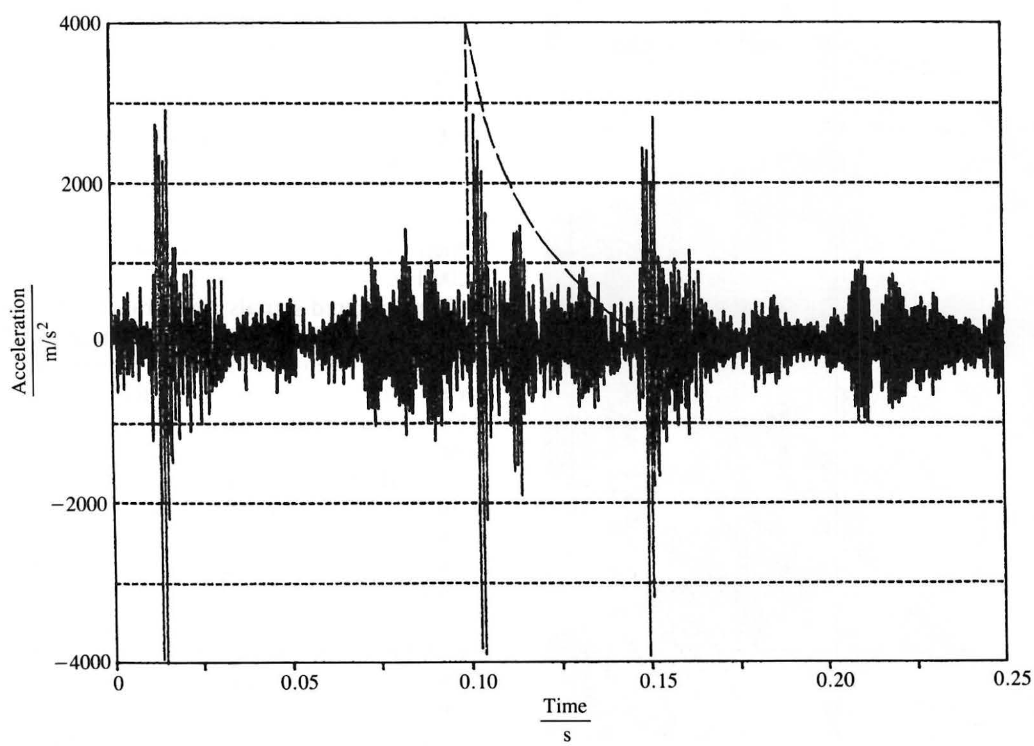

Fig. 9 Acceleration point 1 caused by train 10 ($T = 0.25$ s)

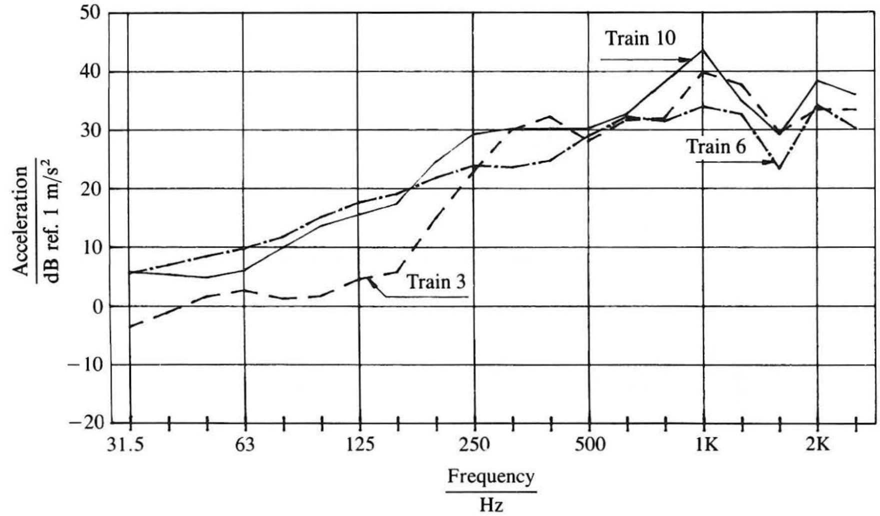

Fig. 10 Acceleration spectrum point 1 caused by trains 3, 6, 10

4.2 Processing of transfer functions

The high values of the coherence functions of the various responses therefore confirm a structural continuity between the switch machine and the track. It is consequently possible to carry out an estimate of the loads brought on by the train on the track and by the accelerations on the switch machine. As an example Figs 12 and 13 show the transfer function and the coherence in a narrow band for the measurement H_{11} (vertical on the track) obtained with the method described previously. All measured FRF were transformed in the $\frac{1}{3}$ octave band as previously described.

4.3 Estimate of the excitation forces and of the acceleration on the switching machine

Since $\gamma_{13}^2(f) \approx 1$ and $\gamma_{24}^2(f) \approx 1$, thereby eliminating the contributions of $F_3(t)$ and $F_4(t)$, it is possible to proceed according to the description in the previous paragraphs. In Figs 14 and 15 the coherence between the accelerations recorded at points 1 and 2 and the $\frac{1}{3}$ octave band autospectrum, of the part of $a_2(t)$ non-coherent with $a_1(t)$, are shown.

The acceleration and the load on the switch machine (point 5) were estimated, starting from the components produced by the single forces. Using equations (7) and (8), in Fig. 16 the comparisons between the acceleration spectrum and the estimated and measured force spectra are shown.

5 CONCLUSIONS

A numeric experimental model based on linear system signal analysis was proposed. This particularly simple model produces a representation of the dynamic train–track interaction at high speed and the behaviour of the portion of switch that is rigidly connected to the switch machine.

There is a good agreement between the form of the estimated acceleration spectrum and that of the measured acceleration, since the numerical error is limited to about 15 per cent. The proposed model is therefore reliable, even if two consecutive estimations have been used (Fig. 3).

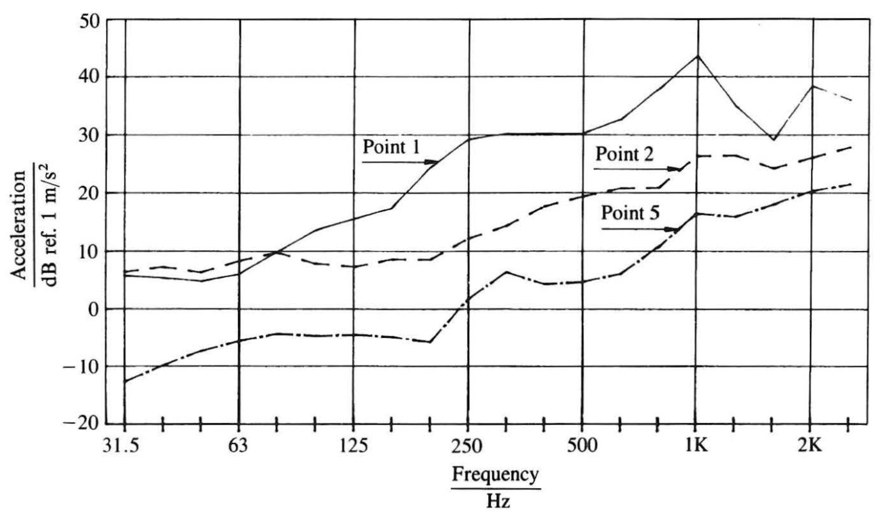

Fig. 11 Acceleration spectra points 1, 2 and 5 caused by train 10

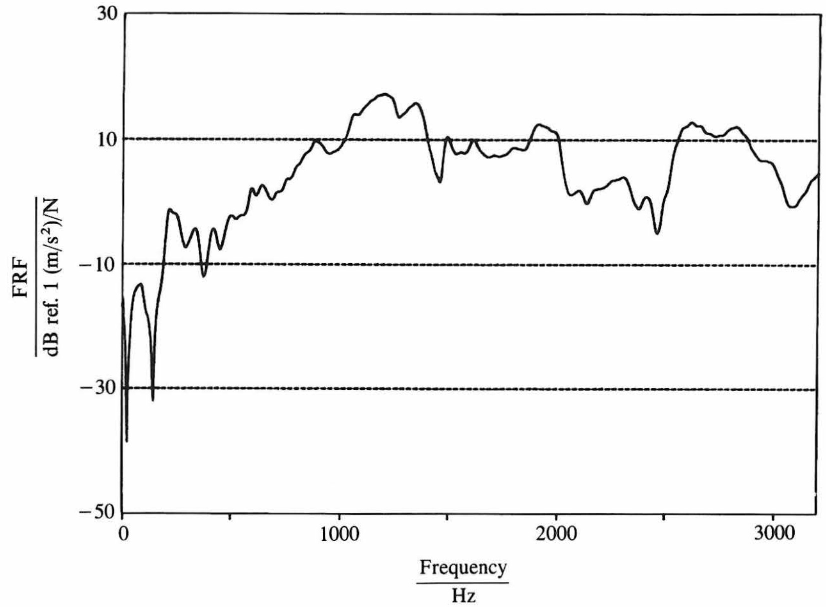

Fig. 12 Frequency response function $H_{11}(f)$

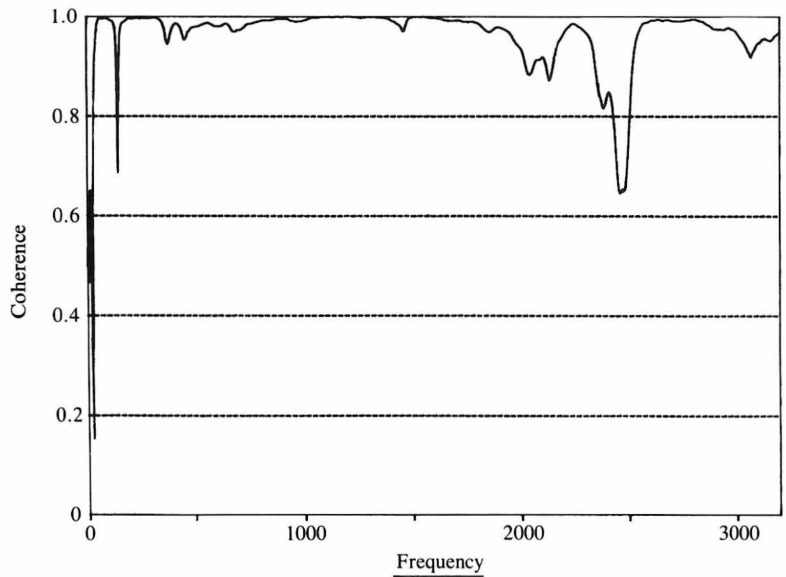

Fig. 13 Coherence of the frequency response function $H_{11}(f)$

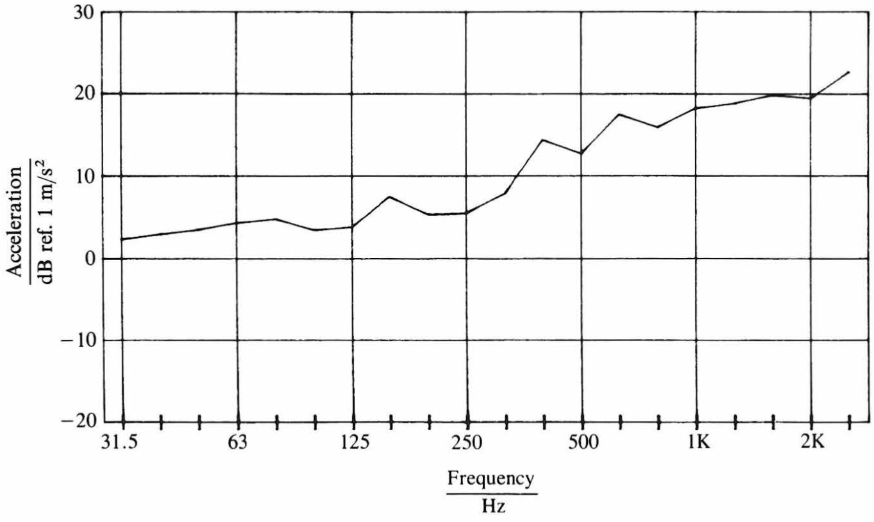

Fig. 14 Estimation of $a_{2.1}$

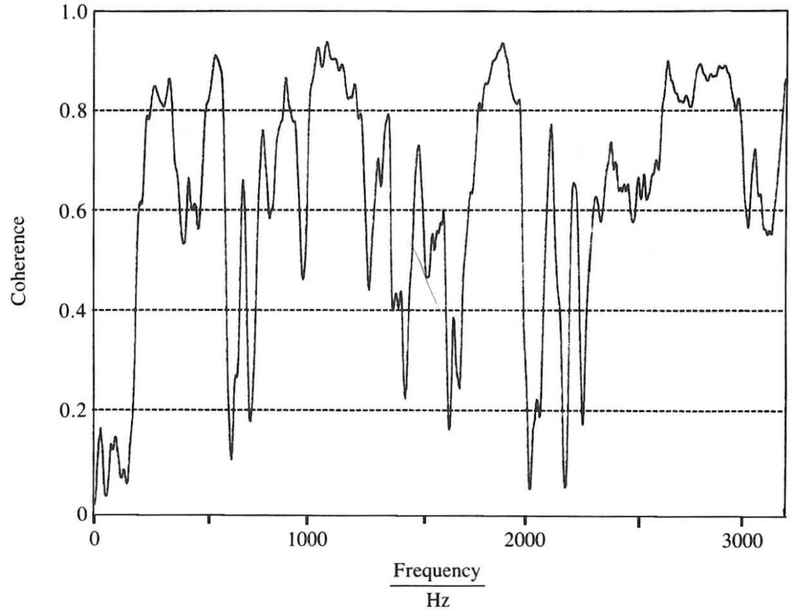

Fig. 15 Coherence between a_1 and a_2

The proposed method is still being developed; it will use numerical values for the excitations obtained by standard measurements carried out on standard and recently developed track equipped for the purpose. This eliminates the necessity to make an estimate using the relationship $F = a/H$ and therefore the resulting error is reduced. Further measurements will be taken utilizing multi-channel equipment with enough channels to match the number of detected signals. Therefore all the linear relations between the different signals (ordinary multiple and partial coherence) are taken into account.

To know the effects of the excitations on the internal parts of the switch machine, modal analysis on the switch machine will be executed; in addition, a series of modifications will be introduced to guarantee long life and functionality of the internal parts themselves.

The minimization of stress on the switch machine is possible by modifying the construction of the chassis and the consequent transfer functions. Taking transfer functions measured on the modified chassis and knowing the loads acting on the switch, the acceleration (or force) spectrum that acts on the switch machine can be estimated with the described method. If it is supposed that the dynamic behaviour of the carriage in transit on the modified switch does not vary substantially (constant excitation forces), a testing plant can be constructed to evaluate the validity of the introduced modifications capable of reproducing the conditions of use similar to that currently being studied. In this way it is possible to:

(a) simulate the actual working conditions without having to install the new switch machine on an existing switch and
(b) carry out dynamic and fatigue tests on the parts contained in the switch machine.

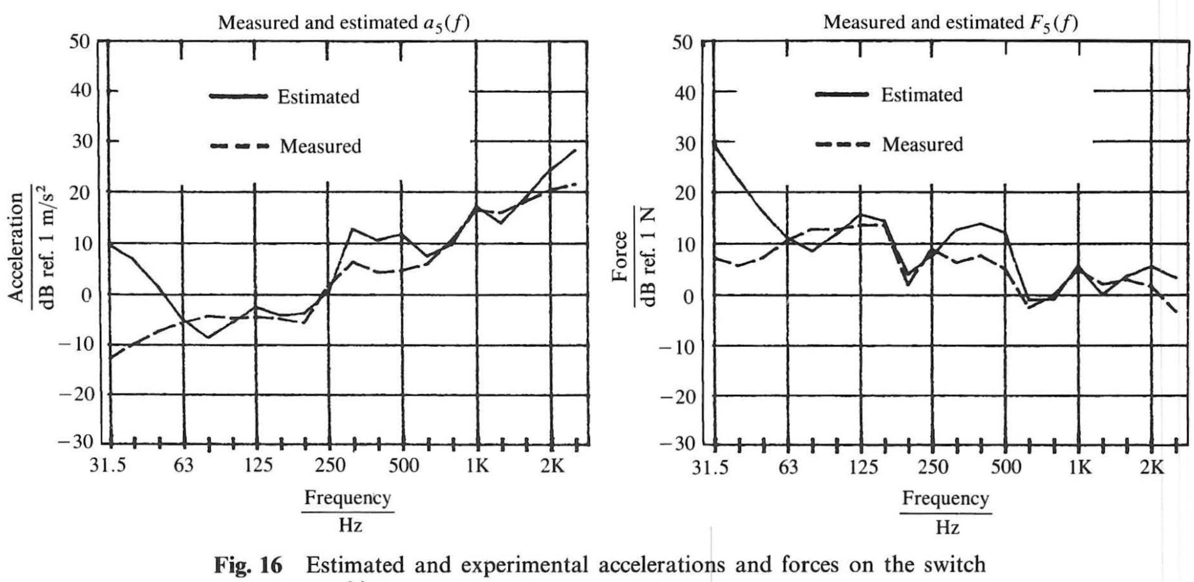

Fig. 16 Estimated and experimental accelerations and forces on the switch machine

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