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Vibration And Ride Comfort Analysis of Four-Wheel Vehicle System Using Lagrangian Dynamics

By

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Abstract

The efficiency or performance of a roadway vehicle is determined based on its specific performance indices. The ride quality index or ride comfort index is one of the measures of those performance indices. This measure is evaluated based on different standards set by indexing organizations e.g., Sperling ride index, ISO 2631 ride comfort specifications etc. In the present paper, a light-duty four-wheel vehicle system is modelled using Lagrangian method and is subjected to bump and pothole inputs. The vibrational response of the vehicle is evaluated and judged based on Sperling ride comfort specifications.

Keywords: Sperling ride index, three-wheel vehicle, ride comfort, Lagrangian method, bump input

1. Introduction

A road or railway vehicle model may be formulated using the Newtons method which is a vector method of modelling the system as it mainly deals with the evaluation of forces using free body equilibrium. The other method is the scalar method which involves the evaluation of position vectors, Eulerian orientations of the bodies, linear and angular velocity vectors, translational and rotational kinetic energies, potential energy, spring potential energies, Rayleigh's dissipation energy and the generalized forces is called Lagrangian method [1-6]. The scalar terms are much easier to deal with as compared with the vector terms; therefore, the complex systems having many degrees of freedom are usually formulated using Lagrangian method [7-12].

A road or railway vehicle is subjected to several types of vibrational inputs as it traverses the road or track. These vibrational inputs may not be only from the road surface or the track surface; the unbalanced forces within the vehicle, and wind drag forces also contribute to vibrational inputs to the vehicle carbody or sprung mass [13-18]. However, the road surface or the track surface irregularities may be considered as the major source of inputs to the vehicle causing discomfort issues. The road surface or the track surface irregularities may be modelled in two ways i.e. either probabilistic or deterministic [19-25]. The probabilistic inputs are the random irregularities which in their simplest form are usually modelled as stationary and ergodic which represents that the statistical feature of the function remains unchanged with the time [26-31]. The deterministic inputs are cusp, bump, jog, plateau, though, sinusoid and damped sinusoid. The analytical form of these inputs is expressed using two parameters, amplitude and duration-related parameters [32-37].

He and McPhee [38] used a multidisciplinary optimization approach to a mechatronic vehicle built with active suspension. In this work, both random and deterministic inputs have been applied independently to the system and its response was evaluated. Kim et al [39] developed a road sensing system which senses both random and deterministic road surface inputs and applied them to 7 degrees of freedom full vehicle system built with active suspension. Van der Sande et al [40] applied stochastic and deterministic road inputs to the quarter car model of road vehicle model built with active suspension and determined useful results through experimentation and simulation. The random road excitation to half car vehicle model with magnetorheological damper to evaluate the multi-objective optimization problem was considered by Prabhakar et al [41].

This study performs the ride and sensitivity analysis of a coupled vertical-lateral 13 DoF independent suspension light four-wheel vehicle full car model formulated using the Lagrangian method. A complex multi-DoF vehicle system may be effectively analysed using Lagrangian method [42-48]. The simulation in this paper has been done in the frequency domain with MATLAB. The data from the field measurements are obtained in the time domain which is transformed into the frequency domain using Fast Fourier Transformation (FFT). Results of simulation and field measurements are compared together to validate the present model. The vehicle is moved on a straight path with a uniform speed of 60 km/hr. Critical parameters influencing ride comfort have been investigated based on ISO [49] and ISO 2631 (1997) criteria [50] and a sensitivity analysis of ride comfort is presented which suggests a proper blend of the parameters for the optimum ride. The ISO/ ISO 2631-1 annexure provides the most universally accepted comfort specifications for rail and road vehicles [51-52],

2. Literature Review

Gopala et al., 2022 [1] used a previewing management to investigates a "quarter-car" design of a four-wheel automobile having a "nonlinear" springy moving at a steady rate on a randomized roadway condition. The "Bouc-Wen" approach assumes persistence to account for the "spring's nonlinear" behaviour. The empirical interpolation method was used to simplify this previously "nonlinear" automobile design. The spectrum reduction technique is used to calculate the "root mean square" of the controlling forces, absorber movement, and roadway retaining features. "Monte Carlo" computation is used to verify the SDM's corresponding linearized design findings, which reveal that anticipatory management improves automobile performance.

Sharma et al., 2022 [2] presented a interdisciplinary analytical approach for determining the fatigued lifespan of the automobile shell construction of a railways locomotive and buggies subjected to randomized dynamical stresses. Initially, the dynamical force records of the vehicle-body architecture were analysed using a composite wear assessment approach, which included Multi-Body Component computation and the "Finite Element Method" (FEM). Dynamical strain is determined by applying longitudinally trains kinematics to the stress. Secondly, vehicle-body structural endurance lifespan and fatigued degradation were estimated using a complex degradation accumulating approach in wear assessment. Considerable consistency is found between the simulated outcomes from the analytical model and the investigational findings. After the FEM has been created, the mode may be calculated. The modal behaviour is employed as stimulation to generate the required kinetic load at every junction. Destruction to the vehicle's structure was calculated using a hybrid of the "NMCCMF" degradation accumulation approach and kinetic load. Therefore, author look at how the longitudinal stress affects the fatigue damage to the automobile structure. Fatigue degradation to the vehicle-body is exacerbated by the longitudinal stress.

Gopala et al., 2022 [5] examined a prototype of a quarter-scale automobile equipped with a skyhook absorber, travelling over a bumpy road, having 2 "Degrees of Freedom" (DOF). Appropriate functioning of the skyhook absorber is determined by its controlling settings, specifically its spring constancy and damping ratio. By comparing the load of controlling of LQR under the stochastic optimum controller to that of skyhook absorber, the optimized settings of the shock absorber may be determined. A measurement score, calculated as the weight cumulative of the "mean square acceleration", roadway traction, damper length, and controlling load, is used to fine-tune the skyhook absorber suspension's settings and raise the car's functionality to that of a dynamic damper setup with predictive controlling.

Sharma et al., 2021 [6] analysed the parameters of an "Indian Railway Rajdhani" (LHB) coach. The Lagrangian approach is used to create an appropriate computational technique with degree of freedom of 40. Eleven weight framework including passenger chair aid, position, carriage chassis, bolsters, bogie structure, and wheel axles are considered in the computational framework of a "rail vehicle". On a tangential course, the car is modeled moving at a velocity of 100 kilometres per hour. Testing information is gathered from the "research designs and standards organisation" (RDSO), Lucknow, and compared to the computation findings for verification. Statistical assessment is used to calculate an approximation of the influence that various rail-vehicle characteristics have on the riding experience.

Palli et al., 2021 [10] studied the dynamical responsiveness of a component underneath the operation of certain broad time-dependent stresses may be determined by a method known as time dependent dynamical assessment. The reloading period range is long enough that the inertial or absorption impacts are taken into account. The current study utilises finite element analysis to undertake a period - record assisment of a conventional locomotives carriage under Indian railway circumstances. After modelling a "bump in the track's surface" in the shape of an "ellipsoid" and assuming that an automobile travelling at that speed would cross the bump in 0.144 seconds, the authors mapped the time-dependent alteration in movement at several crucial regions on frameworks of trucks and cars subjected to the same loading settings. Because of the uneven weight arrangement, the forward and back of the locomotives wagon and carriage chassis are more sensitive to wheel resonance than the middle.

3. System Modeling

A coupled vertical-lateral 13 degrees of freedom light-duty roadway passenger vehicle model is shown in Figure 1. The present model is formulated using Lagrangian method; assigning 5 DoF to the sprung mass i.e., bounce, lateral, roll, pitch and yaw, 2 DoF each to the four unsprung masses i.e. vertical and lateral considering independent suspension. The vehicle parameters and their values are shown in Table 1. The mathematical model is developed with assumptions.

- The vehicle is moving at a uniform speed and the degree of freedom in the longitudinal direction is ignored.
- Wind drag forces are steady.

- Rigid mass is assumed, and vehicle structure does not distort during motion.
- The vehicle possesses longitudinal symmetry.
- Linear vehicle suspension is assumed.
- The tyre has linear suspension properties.

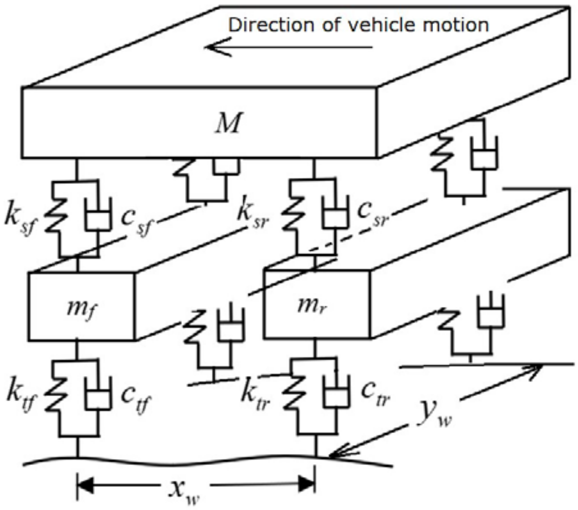

Figure 1: *Vehicle model*

Table 1. *Parameter value of the four-wheel vehicle [29].*

Parameter	Value	Parameter	Value
m	581 kg	k_{tf}	400 kN/m
m_f	28.5 kg	c_{tf}	700 N-s/m
m_r	48.5 kg	k_{sr}	55 kN/m
I_M^x	725 kg-m^2	c_{sr}	6300 N-s/m
I_M^y	1400 kg-m^2	k_{tr}	400 kN/m
I_M^z	1200 kg-m^2	c_{tr}	700 N-s/m
k_{sf}	50 kN/m	x_w	2.36 m
c_{sf}	6000 N-s/m	y_w	1.29 m

3.1. Equations of motion

The EoM of the formulated model is derived with the Lagrange’s approach and they may be expressed as:

$$\frac{d}{dt}\left[\frac{\partial L}{\partial \dot{q}_i}\right] - \frac{\partial L}{\partial q_i} + \frac{\partial E_P}{\partial q_i} + \frac{\partial E_D}{\partial \dot{q}_i} = Q_i \qquad (1)$$

3.2. Description of road surface irregularities

The present analysis utilises the PSD of road surface irregularities by Honda et al. [53]. The random road surface input f(x) may be expressed by function:

$$f(x) = \sum_{k=1}^{N} \delta \cos(2\pi\omega_i x + \varphi) \qquad (2)$$

Meanwhile δ is the maximum displacement, ω, the frequency in the interval ω_{min} to ω_{max} in which the PSD function is φ

The parameters δ and ω are determined using following expressions:

$$\delta^2 = 4S(\omega_i)\Delta\omega \quad (3)$$

$$\omega_i = \omega_{min} + (K - 0.5)\Delta\omega \quad (4)$$

Where, i = 1, 2, 3……. N

$$\Delta\omega = (\omega_{max} - \omega_{min}) / N \quad (5)$$

PSD of vertical irregularities obtained by Fast Fourier Transformation of time-domain data. Power regression analysis is utilised to determine the best-fit curves for the considered road surface profiles for the evaluation of the values of C_s and N to represent the PSD as:

$$S_r(\eta) = C_s \eta^{-N} \quad (6)$$

Meanwhile $S_r(\eta)$ is PSD of road surface input, C_s is an empirical constant, η is spatial frequency in cycles/m, N is slope at which amplitude decreases with frequency and ISO 2631-1 [55] specifies the value of overall vertical weighted acceleration to be expressed as:

$$a_{Wz} = [\sum W_k a_z)^2]^{1/2} \quad (7)$$

and the value of overall lateral weighted acceleration to be expressed as:

$$a_{Wy} = [\sum W_d a_y)^2]^{1/2} \quad (8)$$

$W_{k,d}$ are Frequency weightings for vertical and lateral motion respectively and $a_{z,y}$ are sprung mass centre vertical and lateral RMS acceleration.

The vibration total value of weighted RMS acceleration as per annexure ISO 2631-1 [55] is expressed as:

$$a_v = [k_y^2 a_{Wy}^2 + k_z^2 a_{Wz}^2]^{1/2} \quad (9)$$

where, k_y and k_z are the constants with values as 1.4 and 1 respectively.

4. Results

Meanwhile $S(\omega_i)$ is the PSD function (in m^3/cycle), and ω_{min} and ω_{max} are the cut-off frequency range, respectively. Time-domain data of vertical and lateral irregularities obtained using surface profilometer is shown in Figure 2.

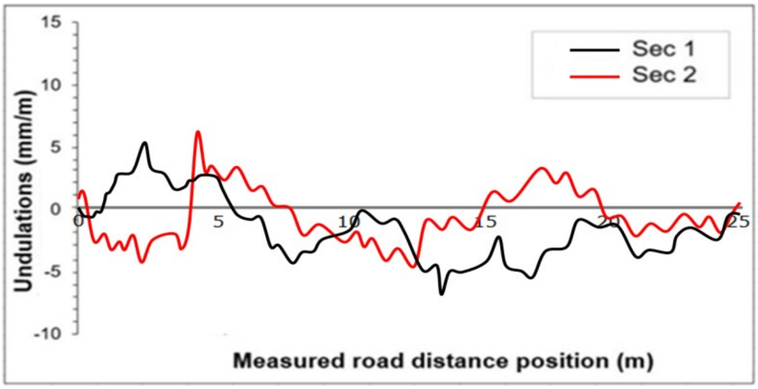

Figure 2. *Time-domain data of vertical irregularities.*

4.1. Sprung mass displacement

The sprung mass displacement in the time-domain is shown in Figure 3 when the vehicle is traversing the random road surface. The peak displacement of 0.13 m is observed. In the present analysis the random irregularities are considered as stationary and ergodic.

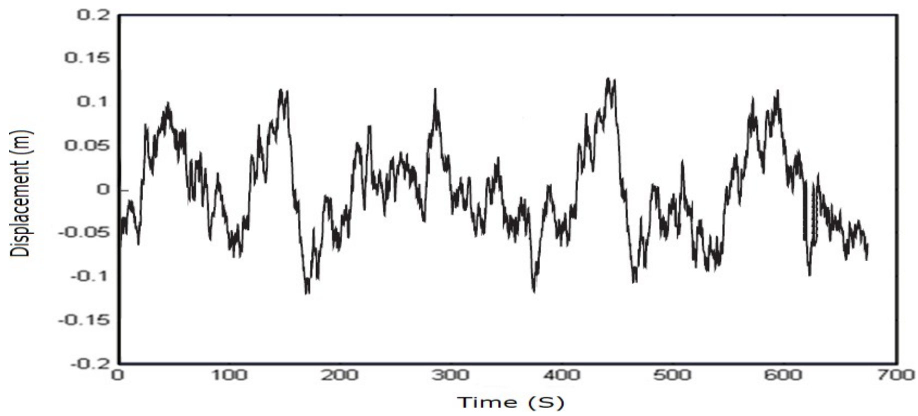

Figure 3: *Sprung mass displacement in time domain subjected to random inputs*

4.2. Power Spectral Density acceleration

Vertical Power Spectral Density acceleration of sprung mass center from simulation and experimental testing are shown in Figure 4 and Figure 5. Similarly lateral Power Spectral Density acceleration of sprung mass center from simulation and experimental testing are shown in Figure 6 and Figure 7, The simulated and experimental test results can be compared well and the model is justified.

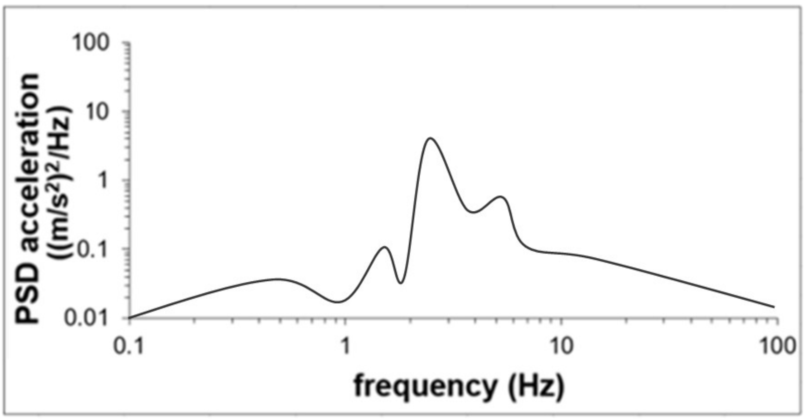

Figure 4: *Vertical MSAR of sprung mass (simulation).*

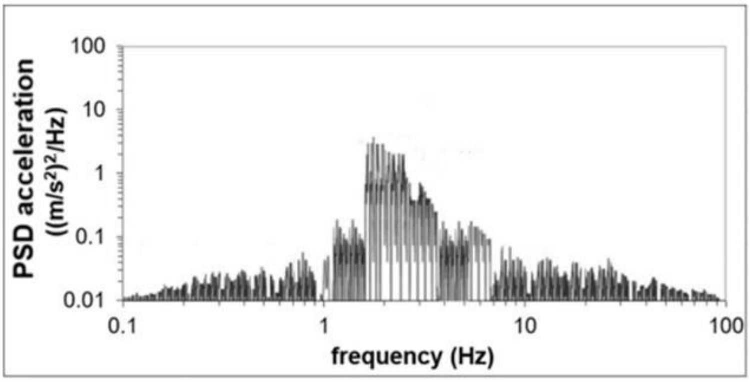

Figure 5: *Vertical MSAR of sprung mass (field measurements).*

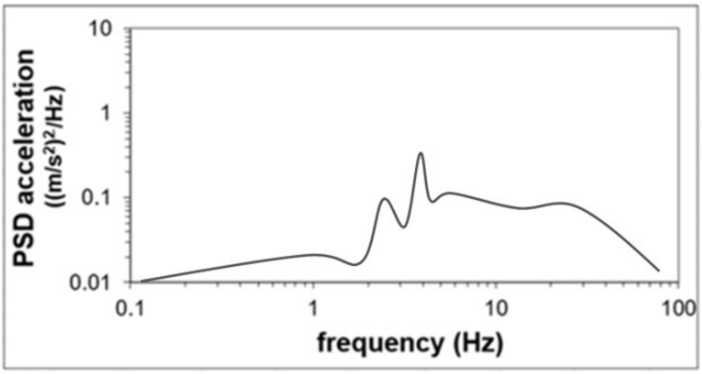

Figure 6: *Lateral MSAR of sprung mass (simulation).*

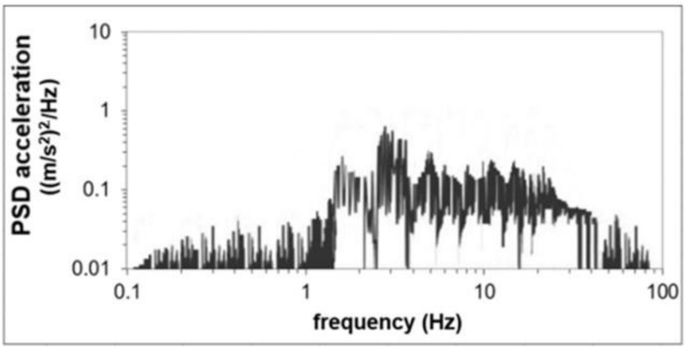

Figure 7. *Lateral MSAR of sprung mass (field measurements).*

4.3. Ride comfort

The ride quality and the ride comfort of a light passenger four wheel vehicle are important concerns for the occupants nowadays. The ride quality is defined as the ability of a vehicle suspension to maintain the vehicle motion well inside the range of human comfort which depends on displacement, acceleration, jerk and other aspects i.e. noise, humidity and temperature. To determine the vehicle ride quality vehicle is investigated based on the abovementioned factors. For determining the ride comfort the vehicle is judged based on the influence of mechanical vibrations on the occupants. The ride quality and ride quality is predicted using different evaluation methods worldwide. The ISO specification [54, 55] of the same is the most popular evaluation method universally accepted by all nations. The principal frequency weighting values described in ISO 2631-1 [55] are multiplied in RMSAR of the sprung mass c.g. and the weighted Root Mean Square (RMS) acceleration values are plotted for frequency (Figures 8 and 9). Weighted vertical RMSAR (Figure 8) describes that the response of sprung mass c.g. remains inside the ISO comfort boundary excluding the frequency range from 1.4 to 7.2 Hz for 8 hrs comfort. Vertical ride lies in discomfort for frequency range from 1.6 to 6.6 Hz for 4 hrs comfort and from 1.7 to 6.5 Hz for 2.5 hrs comfort. The vertical ride is well inside the comfort range for 1 hr comfort except for frequency range from 1.9 to 3.3 Hz. Weighted lateral RMSAR (Figure 9) indicates that the response of sprung mass c.g. remains inside the ISO comfort criteria except for frequency at nearly 3 to 3.5 Hz for 4 hrs comfort and from 2.8 to 3.7 Hz for 8 hrs comfort. For 1 hr and 2.5 hrs comfort lateral ride remains within the boundary.

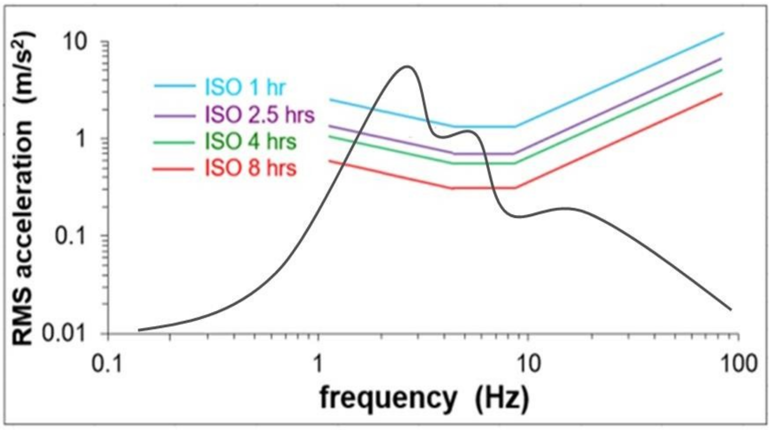

Figure 8. *Weighted vertical RMSAR for sprung mass c.g.*

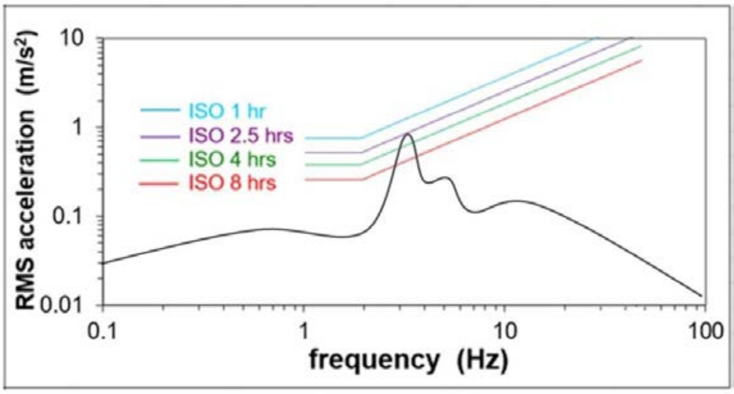

Figure 9. *Weighted lateral RMSAR for sprung mass c.g.*

The overall weighted vertical RMS acceleration & overall weighted lateral RMS acceleration for the frequency range from 1 to 80 Hz are evaluated as 2.27 m/s^2 & 0.58 m/s^2 respectively. The total value of weighted RMS acceleration is evaluated as 2.32 m/s^2 . This value is the level of 'very uncomfortable' vehicle ride comfort index according to the ISO 2631-1 annexure.

Conclusions

From the present ride analysis, it is found that weighted vertical RMSAR of sprung mass c.g. remains inside the ISO-2631 comfort guidelines apart from frequency range from 1.4 to 7.2 Hz for 8 hrs comfort, 1.6 to 6.6 Hz for 4 hrs comfort, from 1.7 to 6.5 Hz for 2.5 hrs comfort and from 1.9 to 3.3 Hz for 1 hr comfort. Weighted lateral RMSAR of sprung mass c.g. remains inside the ISO comfort guidelines apart from the frequency at nearly 3 to 3.5 Hz for 4 hrs comfort and from 2.8 to 3.7 Hz for 8 hrs comfort. For 1 hr and 2.5 hrs comfort lateral ride remains within the comfort boundary. The present formulated model is also useful for the investigation of vehicle stability using eigenvalue analysis. The damping is present in the system the eigenvalues would be complex conjugate and the system is unstable provided the real part of the eigenvalue of a rigid body in a particular mode remains positive. The eigenvalue analysis also provides information about the damped and natural frequencies of the considered rigid bodies in different modes, decay rate and the damping present in the system. A human biodynamic subject model may be integrated with this vehicle model to analyse the seat to head transmissibility characteristics and induced acceleration analysis in human segments.

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