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# Design of a suspension of a training sulky

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**Abstract**

A design study based on a modelling approach was used to optimise the characteristics of a new concept of suspension for training sulkies. The numerical model of sulky including the suspension and, in a second stage, a mechanical model of driver, allowed the definition of technical specifications. They were used to manufacture a prototype of suspension mounted on a current production model of sulky. The prototype of suspension was then tested in the lab and in a racecourse in real conditions of use.

The resulting vibration exposure was assessed from vibration measurements. It was slightly lower than the limit value enacted in the European Vibration directive (1.15 m s$^2$), but drastically reduced in comparison with the exposure measured on the original sulky (2.56 m s$^2$). The prototype and moreover the design procedure is currently in a transfer process towards sulky manufacturers.


## 1. Introduction

The vibration exposure of harness racing trainers has been assessed according to standard AFNOR NF E 90-401 [1] and with 15 current production sulkies. It showed values of vertical acceleration (from 1.4 to 3.7 m/s$^2$ [2]) at the driving place higher than the limit value enacted in the European vibration directive [3].

Racing sulkies are suspensionless because of weight and size constraints. But the races are not long enough to cause risks in comparison with the 4 h of daily training usually required to prepare competitions.

Numerous types of the so-called suspensions of training sulkies are available on the market. Most of them consist of seats articulated on their front size and supported by spring/damper systems on their rear border, see Fig. 1(a). It is easy to understand that the vibration is fully transmitted to the driver through the pivot joint. Moreover, the springs are too stiff to attenuate the vibration. More basic suspension systems such as suspension blades see Fig. 1(b), torsion springs, see Fig. 1(c), or more sophisticated systems like articulated cradles, see Fig. 1(d), did not show any effect in terms of vibration attenuation. All the tested systems amplified the vibration instead of filtering it.

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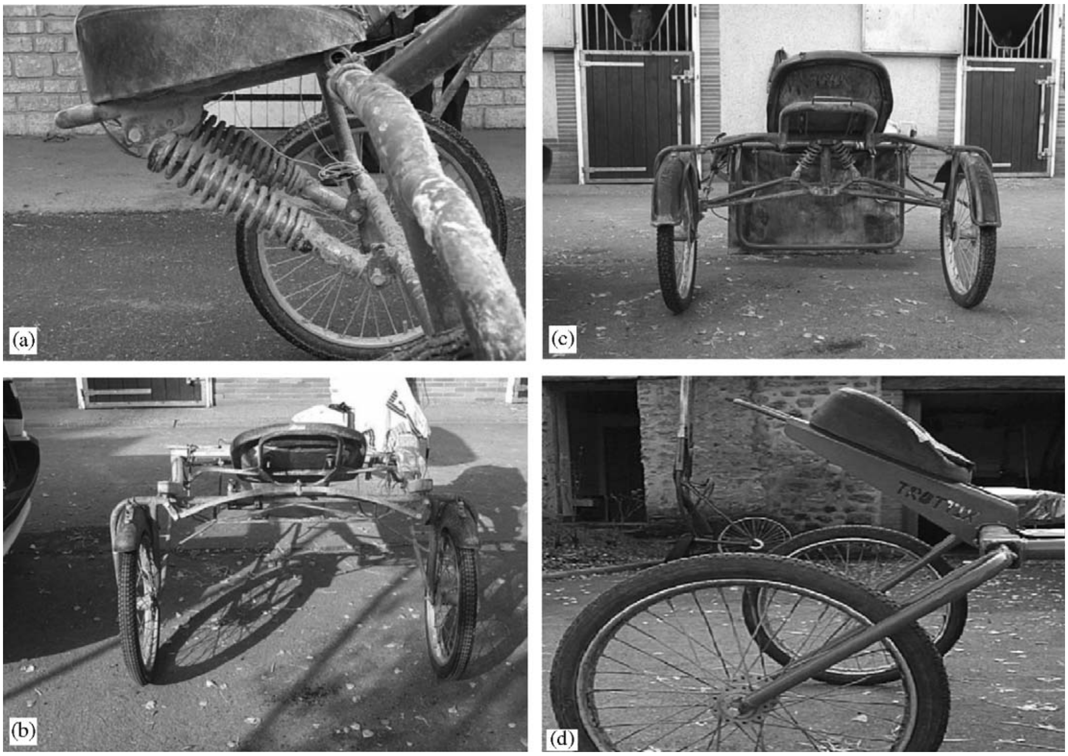


Fig. 1. Typical suspension systems for training sulkies currently on the market: (a) articulated seats with spring/dampers; (b) suspension blades; (c) articulated cradle; and (d) torsion springs.

In France, the population daily concerned with this situation is estimated at about 3000 workers. Harness racing trainers have a real passion for their work and for horses. So they can endure a lot of constraints and they are not inclined to protect themselves against their hard working conditions. The French organisation for occupational health in the agricultural world (Mutualité Sociale Agricole) is conducting a wide campaign among trainers and sulky manufacturers to increase the awareness of the occupational risks. INRS participates in that national project with a feasibility study whose objective is to demonstrate that technical solutions exist to lower the vibration exposure of trainers towards acceptable values. The aim was to develop a method, based on modelling, and to use this method to design an optimised suspension of sulky. Then the prototype of suspension was validated with field tests in realistic conditions of use.

## 2. Modelling of a training sulky and its virtual suspension

Prior to the design study, a current production model of training sulky had been chosen as a case study. The model was known as widely used by trainers and its suspension was composed of an articulated seat supported by spring/damper systems (see Fig. 1(a)) such as motorbike suspensions. The tubular design of the sulky made it possible to further modifications.

The objective was to show that with some design improvements the original sulky could efficiently attenuate the vibration.

The position of the centre of gravity and the inertia of the original sulky have been determined with the help of a CAD model. The masses of all the components that could be dismantled had been previously verified by weight measurements. This data was used to implement a numerical model with the mechanical multibody software Solid Dynamics$^{\mathrm{TM}}$.

In addition to the model of the original sulky, a suspension based on the concept of articulated cradle was inserted, see Fig. 2. The whole model was composed of 6 rigid bodies: the tubular chassis including shafts supposed rigid, the wheels which were connected to the chassis with pivot joints, the cradle which could pivot

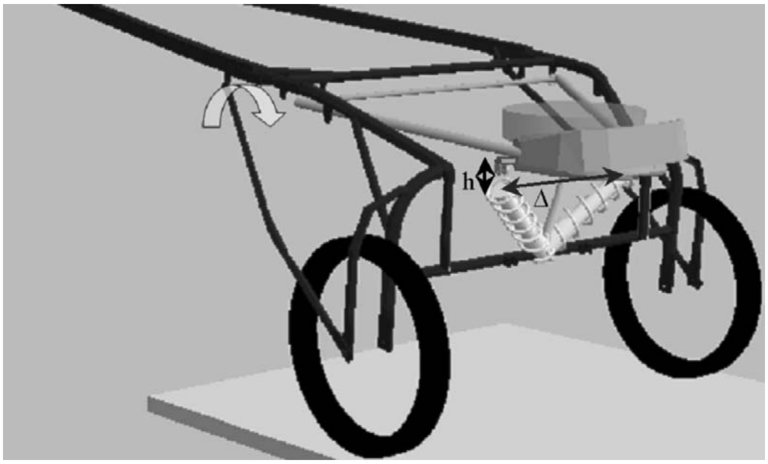


Fig. 2. Numerical model of the original sulky equipped with an articulated and suspended cradle.

around the chassis by means of a pivot joint and two rigid parts simulating the excitation input. The first part further named horse body simulated the input motion due to the horse drive and the second part named track body modelled the interaction between the wheels and the racing track. The horse body was connected to the chassis by means of a spherical joint and a horizontal translational joint (to prevent any overconstraint) and the track body was connected to the wheels by means of cylinder/plan contact elements.

Previous field-test measurements had been performed with the original sulky on a racing track [4]. During a first series of tests, the acceleration was measured in 3 directions at the shaft ends on the horse's flank. A second series was realized with an automobile towing the sulky instead of a horse. In that way the vibration measured at the wheel axle (in the vertical direction) was only caused by the unevenness of the track and not by the horse drive. The acceleration was then digitised and input as time history in the model. The acceleration measured at the shaft ends was input to the 3 space components of the horse body and the acceleration measured on the wheel axle was input to the track body. Fig. 3 illustrates the PSD of the acceleration input used for modelling.

The acceleration measured at the shaft ends is a narrow band signal which predominant frequency is located at 4 Hz. It corresponds to the trot pace. The acceleration measured on the wheel axle is a random signal which bandwidth is limited in the range (7–20 Hz). It depends on the state of the surface of the track.

The components of the new suspension design have been kept as the spring/damper systems of the original sulky. Their mechanical properties have been measured in the laboratory from cyclic tests performed with an electro-hydraulic actuator. They behave quite linearly, so the stiffness (16,500 N/m) and the damping coefficient (100 N/m/s) were considered as constant and fitted to the measurements. In our experience in suspension design [5], the damping coefficient was thought to be insufficient to prevent the suspension from topping or bottoming. So a linear damper was inserted in-between the two spring/damper systems. Its damping coefficient was also determined by cyclic tests (600 N/m/s).

The two spring/damper systems were mounted between a reinforcement bar of the chassis and the articulated cradle. They were inclined in the transverse direction (V-shaped) in order to lower the global stiffness of the suspension (see Fig. 2). The design process consists of optimising the angle of these components for each given value of the driver's weight. The travel distance of the spring/damper systems was 6 cm.

Table 1 gives the RMS values of the acceleration input signals used in the sulky model and illustrated in Fig. 3, and the RMS values of the acceleration computed at the driver's seatplace supposing the suspension blocked, i.e. infinite stiffness of the spring/dampers (output). These latter acceleration values must be considered as representative of the exposure in vertical and fore and aft direction without any suspension.

*Remark*: The assumption of no deformation of the sulky was verified by preliminary measurements and modelling: a finite element model of the whole sulky was built up, including the wooden shafts, the original steel frame and the additional cradle loaded with a 100 kg point mass (the suspension spring/damper were modelled as rigid bodies to compute other natural frequencies than the suspension frequency which, in the

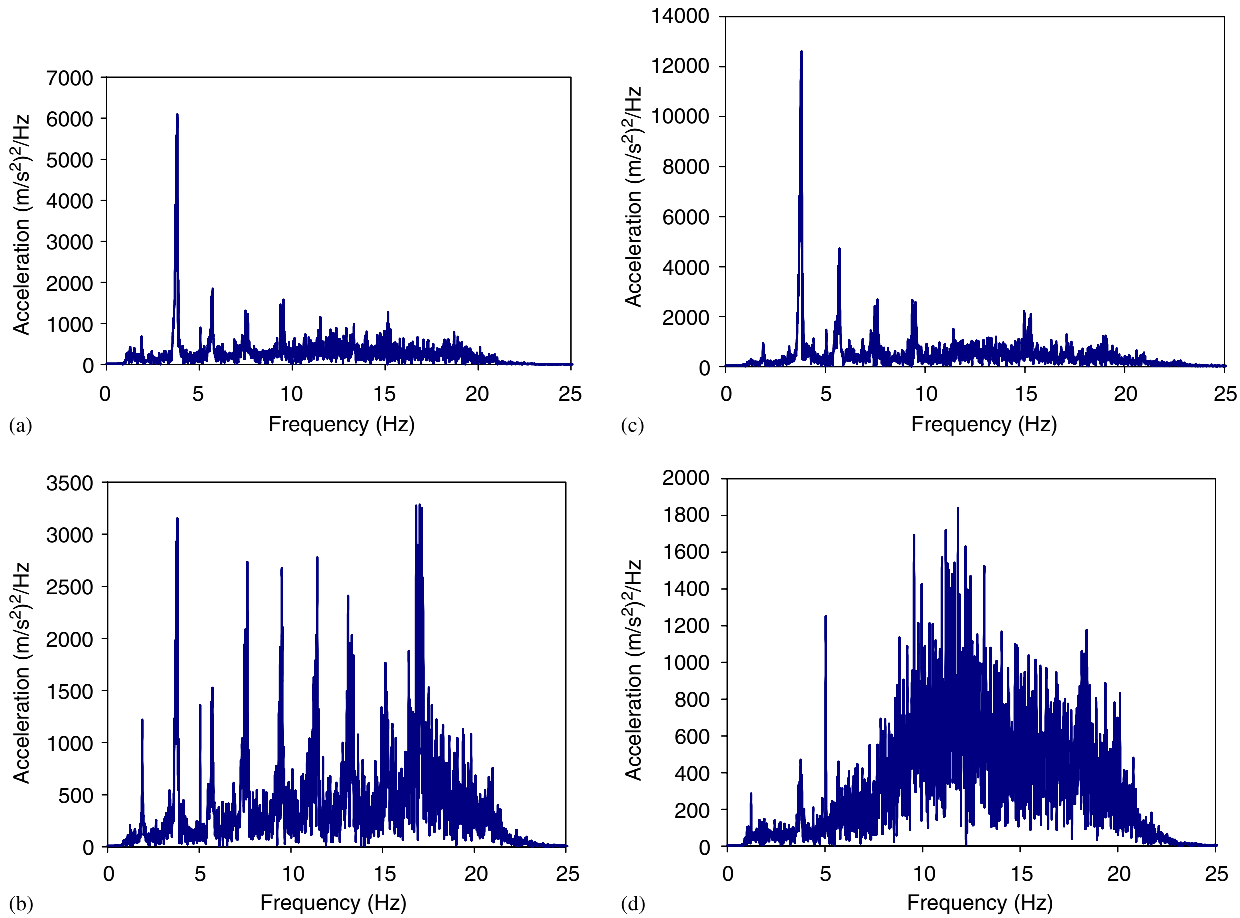


Fig. 3. Power spectral density of the acceleration signals input in the model of sulky: (a) shaft ends—fore and aft; (b) shaft ends—transverse; (c) shaft ends—vertical; and (d) wheel axle—vertical.

Table 1
Acceleration values of input excitation signals and computed output acceleration signals at the driver's seat with completely rigid suspension

| Acceleration | Unweighted RMS value ($m/s^2$) | Weighted RMS value ($m/s^2$) | ISO 2631-1 frequency weighting filter |
|---|---|---|---|
| Input | | | |
| Shaft ends—fore and aft | 6.66 | 6.17 | $W_d$ |
| Shaft ends—transverse | 8.14 | 6.96 | $W_d$ |
| Shaft ends—vertical | 12.90 | 12.10 | $W_k$ |
| Wheel axle vertical | 6.14 | 5.42 | $W_k$ |
| Output (computed with suspension blocked) | | | |
| Seat place—vertical | 6.86 | 6.11 | $W_k$ |
| Seat place—fore and aft | 4.63 | 4.21 | $W_d$ |

experience in suspension design [5–8] should not exceed 2.5 Hz). One of the two shafts was tested with a shock hammer to measure its first natural frequency under hinged/free conditions (48 Hz). This test was used to fit the FEM model of the shaft and adjust the value of the Young modulus (14,000 Mpa). The FEM of the whole sulky was then used to compute the first natural frequency with the following boundary conditions: spherical

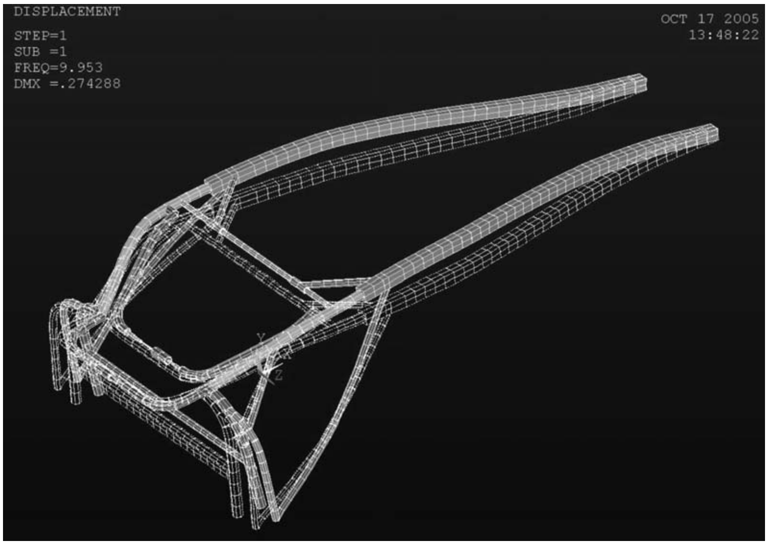


Fig. 4. Deformed shape of the 1st mode of the FEM model of sulky (computed natural frequency: 9.95 Hz).

joints at shaft ends and fixed vertical displacement at the wheel axles. The first natural frequency was computed 9.95 Hz (see Fig. 4) and corresponds to the bending of the shafts and the resulting rotation of the frame. In comparison with the targeted suspension frequency (below 2.5 Hz), the natural frequency of the driver response (below 5 Hz, see next sections) and the frequency content of the horse excitation (around 4 Hz), it was then consistent to assume a rigid behaviour of the structure. With regard to the track excitation input which frequency content encompasses the computed natural frequency of the sulky, it is clear that a rigid body model would not be sufficient to predict possible deformation of the shafts and rotation of the frame due to the unevenness of the track. Nevertheless, the effect of the deformation of the shaft was implicitly taken into account when measuring the acceleration on the wheels.

## 3. Parametrical study

The whole model composed of the original sulky and the proposed concept of articulated cradle supported by V-shaped spring/damper systems was then used to perform a parametrical study and to optimise the design parameters of the suspension.

The design parameters of the suspension are:

- the distance (named $\mathbf{\Delta}$ in Fig. 2) between the upper joints of the two spring/damper systems; and
- the vertical setting of the seat (distance between the upper joints and the seat base, named $\mathbf{h}$ in Fig. 2).

The angle of the spring/damper systems is controlled by the parameter $\Delta$. As a result, the more inclined the V-shaped spring/damper systems, the more flexible the suspension. For a given load placed on the seat, there is an optimal value of $\Delta$ which is a compromise between the filtering performance of the suspension and the dynamic stroke. The latter must remain compatible with the travel distance.

The design parameter $\mathbf{h}$ has been introduced to compensate the static deflexion due to the gravity and the geometrical deflexion due to the angle of the spring/damper systems.

Calculations have been carried out successively with four different loads: 52, 60, 75 and 100 kg. The load was considered as an inert mass and modelled as a steel cylinder (see Fig. 2). It was adjusted by fitting the height of the cylinder.

All the calculations have been carried out with $\mathbf{h}$ constant. The optimal vertical setting was determined from the results of the parametrical study (see below).

With every load, series of calculations were performed varying the parameter $\Delta$ from 10 to 50 cm by steps of 5 cm. For every single configuration, four criteria were calculated:

- *The gain of the suspension*: The gain is defined as the ratio between the RMS value of the weighted vertical acceleration (according to weighing $w_k$ prescribed in the standard ISO 2631-1 [9]) calculated on the seat and on the chassis (the locations of the two observing points being superposed in the centre of the seat surface when the sulky was at rest). The gain is the inverse of the S.E.A.T. factor which characterizes the ability of a suspension to attenuate the vibration:

$$\mathrm{Gain} = \frac{\sqrt{\frac{1}{t}\int_0^t \left(a_{w_k\ \mathrm{frame}}\right)^2 \mathrm{d}t}}{\sqrt{\frac{1}{t}\int_0^t \left(a_{w_k\ \mathrm{seat}}\right)^2 \mathrm{d}t}}. \tag{1}$$

  The same definition could be used to quantify the attenuation of the suspension in the horizontal direction. Nevertheless, because the horizontal vibration was lower than in the vertical direction (see Table 1) and above all, because the drivers require some force feedback from the horse when holding the reins it was decided not to optimise the horizontal attenuation of vibration.
- The maximum and the minimum of the dynamic stroke. These outputs are used to verify that neither topping nor bottoming occurred:

$$\begin{aligned} \max\ \mathrm{dyn\ stroke} &= \underset{t}{\mathrm{MAX}}(d(t)),\\ \min\ \mathrm{dyn\ stroke} &= \underset{t}{\mathrm{MIN}}(d(t)), \end{aligned} \tag{2}$$

  where $d(t)$ represents the relative displacement of the suspension.
- The static deflexion that takes into account both the load deflexion due to the gravity and the geometrical deflexion due to the angle of the spring/damper systems.

## 4. Determination of the optimal settings

For each load category, the optimal distance setting between the upper spring/damper joints was firstly determined as corresponding to the maximal gain (cf. Fig. 5(a)): for a 100 kg load, the optimal setting was 35 cm, for 75 kg, 40 cm, for 60 kg, 42 cm and for 52 kg, 45 cm. The design values were then chosen as slightly below the optimal values to ensure safety margins before possible mechanical instabilities caused by too large inclinations of the spring/dampers: 30 cm for a 100 kg load, 30 cm for 75 kg, 40 cm for 60 kg and 42 cm for 52 kg.

It was then verified that for every load category and its corresponding distance setting, the dynamic stroke of the suspension remained compatible with the travel distance, i.e., the maximum of the dynamic stroke was lower than the bottom end-stop distance (see Fig. 5(c)) and the minimum of the dynamic stroke was higher than the top end-stop distance (see Fig. 5(d)). So it was made sure that no shocks had occurred during the simulated runs.

For every load category the vertical setting was chosen as the corresponding calculated static deflexion. For instance, with a 75 kg load and with respect to the distance setting of 35 cm, the static deflexion is 0.4 cm (see Fig. 5(b)). This means that inserting a wedge 0.4 cm thick between the seat and the spring/damper upper joints would keep the seat horizontal when loaded with 75 kg. In the same way, the vertical setting was 1.85 cm with 60 kg and 2.15 cm with 52 kg.

The optimal values of **h** and $\Delta$ were used to define the kinematical law of the suspension adjustment. The principle consisted of changing the distance between the spring/damper upper joints by means of a horizontal endless screw. The joints were guided in a bow-shaped slit so that the seat was raised when the spring/damper systems were inclined (see Fig. 6).

The computed gain was between 5 and 7 (see Fig. 5(a)), regardless of the load applied (but assuming the optimal distance settings previously defined) i.e. the vibration at the driver's workplace was calculated as 5–7 times lower than on the sulky chassis. In the same way the natural frequency remained around 2 Hz.

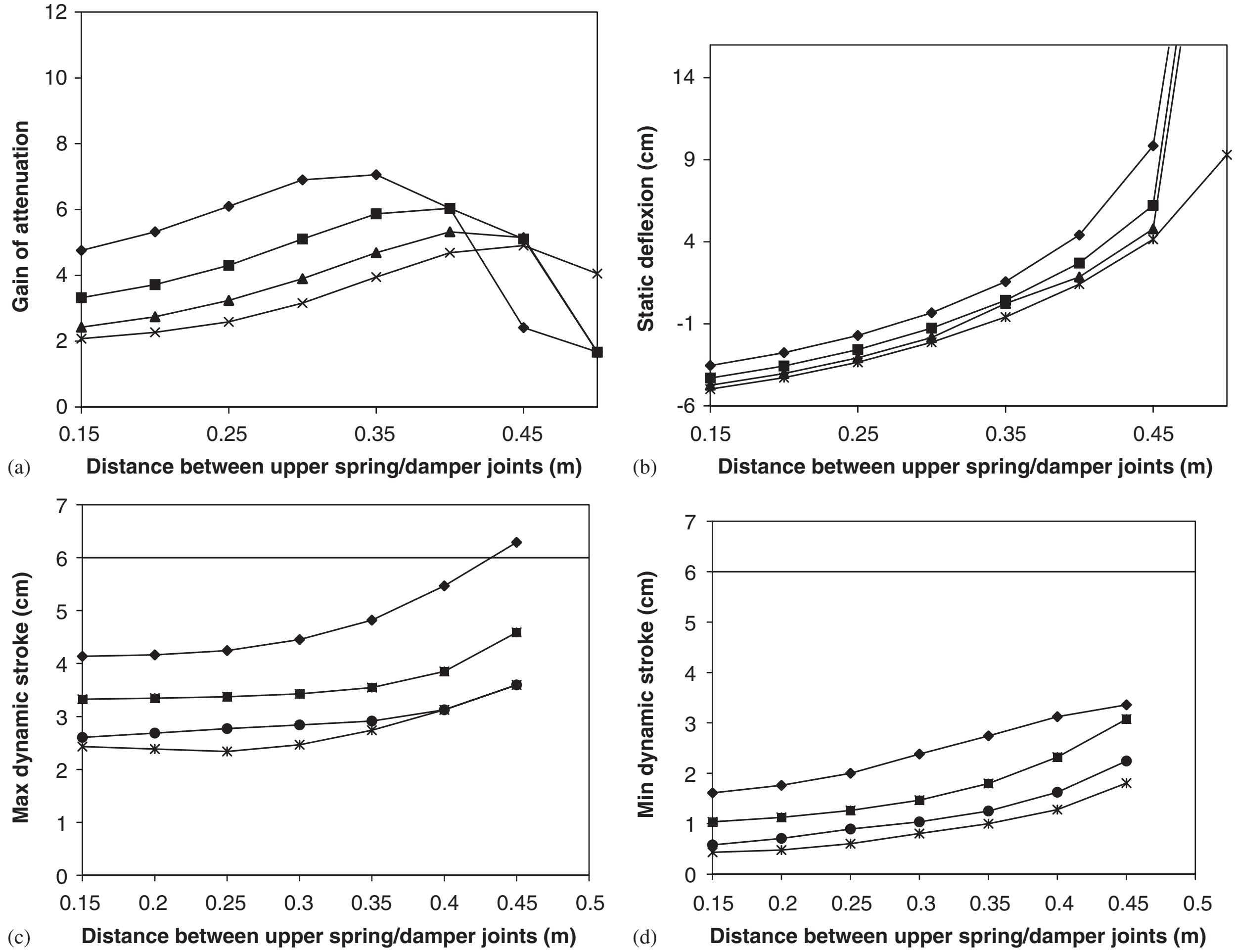


Fig. 5. Calculated performance of the suspension of training sulky: ◆, 100 kg; ■, 75 kg; ▲, 60 kg; and ✱, 52 kg. (a) Gain of attenuation; (b) static deflexion; (c) maximum of the dynamic stroke; and (d) minimum of the dynamic stroke.

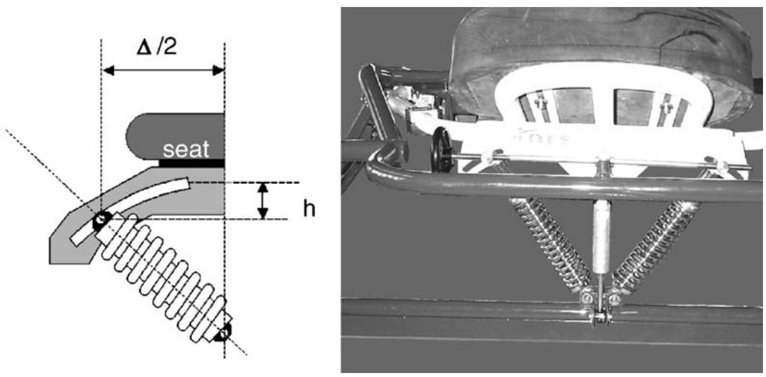


Fig. 6. Principle of the suspension adjustment.

Nevertheless, the suspension performance was predicted with an inert mass instead of a real human subject. It is well-known that the flexibility of the human body can severely impair the filtering performance because of mechanical coupling phenomena.

Then the next step of the design study was to asses the performance of the suspension taking into account the dynamic response of the driver.

## 5. Modelling of the driver response

The deformations of the human body and in particular its resonance modes can affect the dynamic response of the suspension. The combined movement results from the coupling between the structure (sulky + suspension) and the human subject (driver), so the predicted gain of attenuation may differ from the gain calculated with an inert mass.

The driver was modelled as a numerical dummy and inserted into the sulky model in order to assess the real filtering performance of the suspension. The objective was to confirm there was still an interest in manufacturing such a prototype.

The numerical dummy was developed as a multirigid body, each part of the model representing a body segment. The model was anthropomorphic to make it easier the mass distribution but it should not be considered as a biomechanical model. It comprised 9 independent rigid bodies articulated with ideal joints.

Fig. 7 shows a mechanical diagram of the whole dummy. The mobility of the lower limbs was taken into account with the joints at ankles (spherical joints), at knees (pivot joints) and at hips (spherical joints). The upper limb mobility was partially modelled. Only the thorax could move up and down with a translational joint and the lumber part was connected to the pelvis with spherical joints. The arms could be adjusted to match given postures (hands in lap, outstretched arms, etc.) but the joints at shoulders and elbows were kept infinitely rigid during calculations. So their contribution to the dynamic response of the subject was only in terms of mass and inertia.

It is clear that the motion of the spine and the pelvis could not be simulated in a very accurate and realistic way. This would have required more degrees of freedom. Nevertheless the number and the type of joints were sufficient to get a good description of the contributing mode shapes.

The model was connected to the seat by means of a contact element with a horizontal stiffness to simulate the friction between the buttock and the seat surface (see Fig. 7).

The mass distribution of each body segment was fitted to anthropometric data available in Refs. [10,11].

A routine was programmed in the Solid Dynamics™ environment to build and size the numerical dummy in an automatic way. The weight adjustment was realized within a scaling procedure. For example, a 80 kg

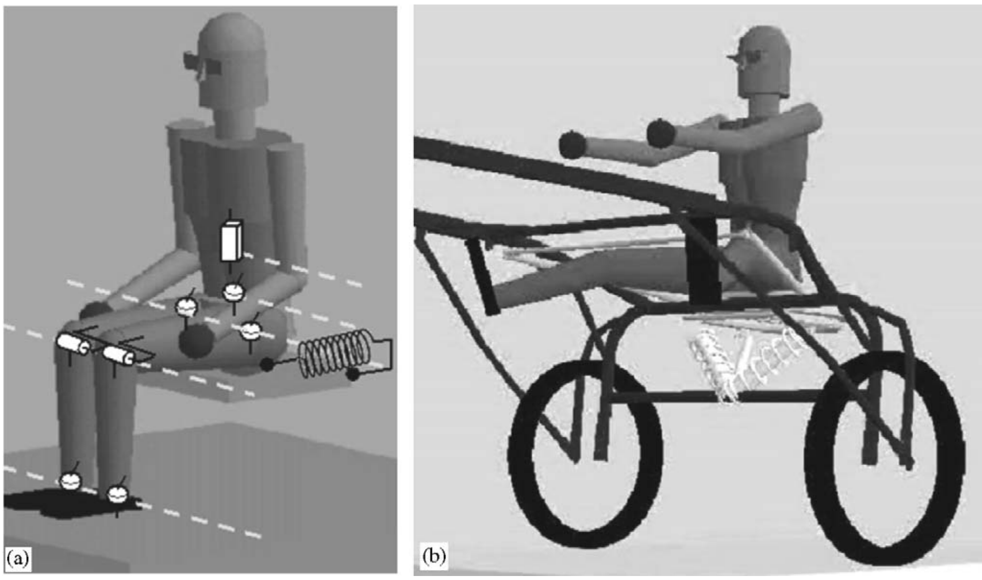


Fig. 7. Numerical dummy: (a) mechanical diagram; and (b) numerical dummy inserted in the sulky model.

model was sized to a 100 kg dummy by multiplying all the dimensional parameters defining the geometries of the different parts by a factor of 1.25 (lengths, widths, heights, radii, etc.). The density of each body segment was kept constant, so that the total weight was proportional to the dimensional parameters. (pelvis density $= 1000\,\text{kg/m}^3$, upper and lower limb density $= 900\,\text{kg/m}^3$, torso density $= 1000\,\text{kg/m}^3$, and head and neck density $= 960\,\text{kg/m}^3$).

The dynamic behaviour of the numerical dummy in the seating position was then obtained by fitting its vertical apparent mass (ratio between the acceleration at the surface of the rigid seat and the transmitted force in the horizontal direction) to the curves given in the standard ISO 5982 [12]. The joint properties (stiffness and damping) were adjusted to obtain a good correlation between the calculated apparent mass and the ISO 5982 mean value, corresponding approximately to a subject of 75 kg (see Fig. 8(a)). The properties of the thorax joint played a predominant role in that fitting procedure because the main resonance of a seated subject, located at around 4 Hz is associated with the bending of the spine (S-shaped mode shape [13]). As a first order model, the mode shape computed with the numerical dummy corresponded to an up and down motion of the torso which was allowed by the translational joint. The apparent mass of the subjects of 58 and 98 kg were obtained by multiplying the joint properties with the scaling factor (weight ratio).

Because no data was available in international standards with respect to the fore and aft direction, the apparent mass of the seated dummy was fitted to lab measurements performed with a human subject of 75 kg [14]. The measurements were carried out with a subject seating on a rigid seat supported by a shaking

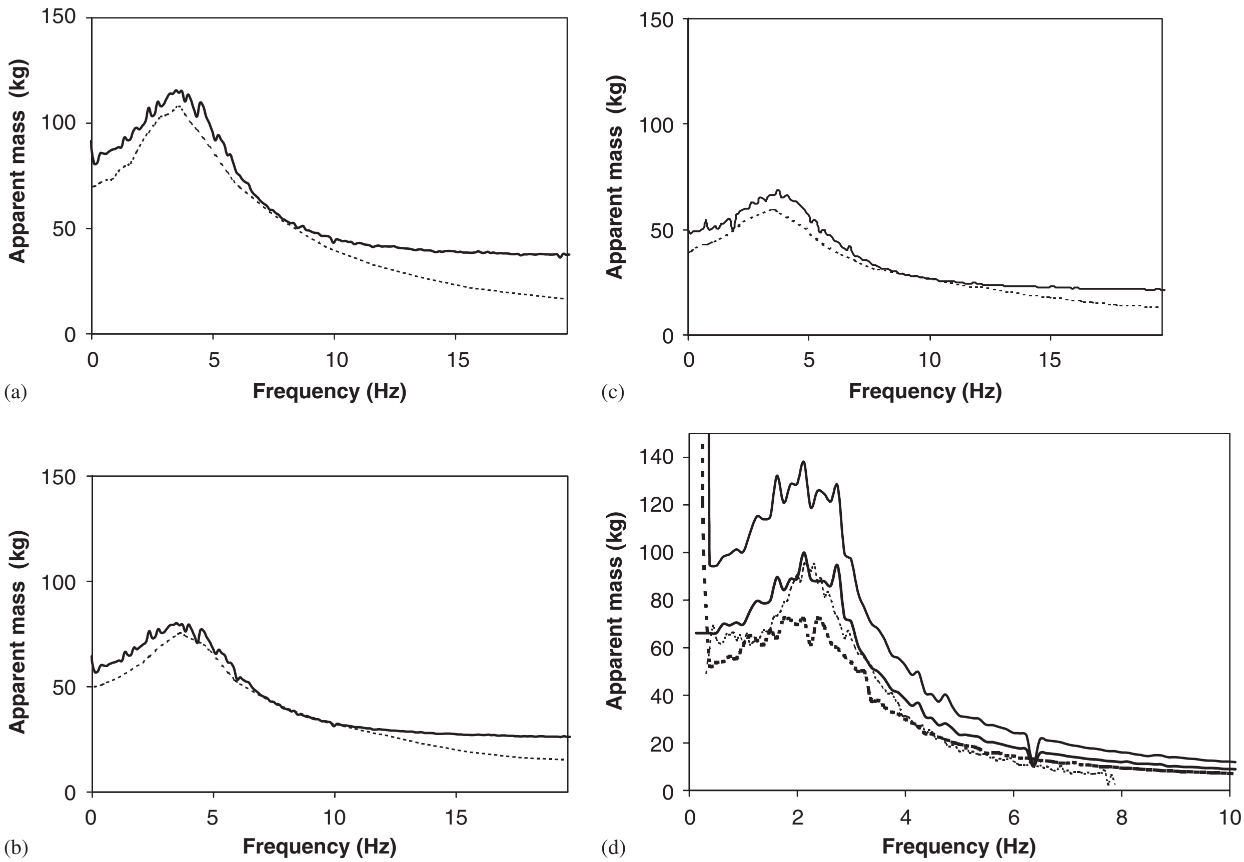


Fig. 8. Fitting of the apparent mass of the numerical dummy: (a) vertical, - numerical dummy 100 kg, - - ISO 5982 98 kg; (b) vertical, - - numerical dummy 75 kg, - - ISO 5982 75 kg; (c) vertical, - numerical dummy 60 kg, - - ISO 5982 58 kg; and (d) fore and aft, - numerical dummy 100 kg, ▬ numerical dummy 75 kg, - - measurement 75 kg, ▬ ▬ numerical dummy 60 kg.

platform. The experiments showed 2 predominant resonances: the first resonance around 0.7 Hz corresponded to a rotation of the hip. The second resonance around 2.25 Hz corresponded to the relative displacement (translational) between the thighs and the seat surface. The joint properties were adjusted to fit the measured apparent mass (see Fig. 8(b)). The computed deformed shapes at 0.7 and 2.25 Hz were identified as qualitatively similar to the measured mode shapes. Fig. 8(b) shows the apparent mass calculated for two other weight categories, 60 and 100 kg after scaling of the dummy.

The numerical dummy was then included in the model of sulky (see Fig. 7). The contact between the buttock and the seat was modelled with the same contact element and the tangential stiffness as previously explained. The feet were connected to the chassis by means of spherical joints.

Then the calculations carried out to optimise the suspension parameters were repeated with the numerical dummy instead of the inert mass. The computation performed with a 60 kg dummy leaded to a gain of 2.21. With a 100 kg dummy, the gain was 2.65.

So, taking into account the flexibility of the human body affected seriously the performance of the suspension. Nevertheless the estimated gain was considered sufficient to launch the production of a prototype.

## 6. Manufacture and validation of the suspension prototype

The suspension was manufactured according to the specifications resulting from the parametrical study. Fig. 6 shows a detailed view of the suspension adjustment system.

Prior to field tests, the performance of the suspension was tested in the laboratory by means of a shaking platform. The objective was to simulate the acceleration input and to measure the gain of attenuation. In real conditions of use, the training sulky is submitted to a double excitation: the horse drive and the wheel/track contact. The shaking platform was limited to $2^\circ$ of freedom (vertical and horizontal) and did not allow to generate a double uncorrelated acceleration simultaneously at the shaft ends and at the wheel axle. So the acceleration has been input successively in two series of tests.

In the first series of tests, the wheel axle was resting on the shaking platform (see Fig. 9(a)). A random acceleration, whose PSD was close to the input signal used in the design study, was imposed onto the platform. The shafts were connected to a fixed point by means of a pivot joint.

Firstly, the tests were performed with an inert mass of 50 kg and then with two different human subjects (light subject: 65 kg; heavy subject: 80 kg). Different levels of input acceleration were tested.

The measurements confirmed the predicted gain, i.e., between 5 and 6 with an inert load and around 2 with both the heavy and the light subject.

In the second series of tests, the wheels were in contact with the ground. The shafts were connected to a vertical post by means of pivot joint. The post was fixed to the shaking platform. A narrow band random signal with a predominant frequency about 4 Hz was input to the platform. In the same way, the tests were performed with an inert mass and with two different human subjects. Unfortunately, the amplitude of the vibration was too high and the inert mass (composed of a bag filled with steel balls) became unstable. Therefore, the tests carried out with the mass have not been post-processed. Nevertheless, the tests carried out with subjects confirmed the predicted gain, i.e. an attenuation of about 2. All the results are presented in Table 2.

The prototype of suspension was then validated in the field, in real conditions of use.

A testing session was programmed in the racecourse named Grosbois near Paris. Four racing horses were requested and a professional trainer was in charge of the tests.

Previously, the sulky had been equipped to measure the vertical acceleration in the centre of the seat surface and the vertical acceleration under the seat, on the chassis. At the same time, the relative displacement of the suspension was measured. All the measurements were recorded on magnetic tapes and digitised in the lab. The gain was post-processed from the two acceleration channels. The prototype was equipped in parallel with two other accelerometers and an integrated acquisition system to compute in real time the vibration exposure at the driver's workplace.

Each session of tests began with two warm-up laps and were followed by two or three quick laps (from 1 min 08 s to 1 min 44 s a lap, distance: 1100 m, see Fig. 10).

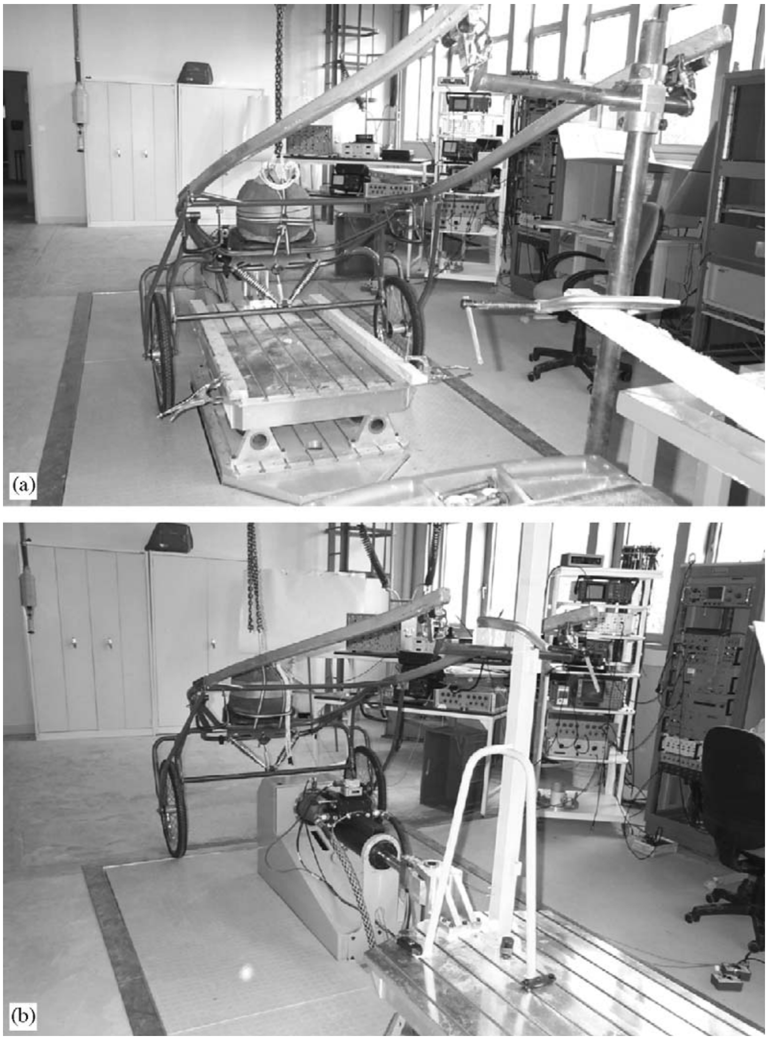


Fig. 9. Validation of the suspension in the laboratory: (a) excitation at the wheel axle; and (b) excitation at the shaft ends.

Table 2
Performance of the suspension of training sulky tested in the laboratory

| Test conditions | Gain of vibration attenuation Z direction (weighing, $W_k$) |
|---|---|
| Excitation at wheel axle—inert mass (50 kg)—Random signal RMS 3.5 m/s$^2$ | 5.92 |
| Excitation at wheel axle—inert mass (50 kg)—Random signal RMS 5 m/s$^2$ | 6.44 |
| Excitation at wheel axle—heavy subject—random signal RMS 5 m/s$^2$ | 2.24 |
| Excitation at wheel axle—light subject—random signal RMS 2.8 m/s$^2$ | 1.88 |
| Excitation at shaft ends—heavy subject—random signal RMS 1.7 m/s$^2$ | 2.37 |
| Excitation at shaft ends—light subject—random signal RMS 1.39 m/s$^2$ | 1.78 |

The gain was measured between 2.94 and 3.85 depending of the type of session and the horse tested.

The vibration exposure was assessed at 1.04 m s$^2$ for one complete training session and 1.18 m s$^2$ for another one (see Table 3). The exposition is then quite the limit value of 1.15 m s$^2$ enacted in the European vibration directive [3]. As comparison, the vibration exposition measured with the same training sulky equipped with its original suspension was 2.56 m s$^2$ [2].

Fig. 10. Field test of the suspension of training sulky.

Table 3
Vibration exposure measured with the prototype of sulky suspension

| Horse tested (Name) | Type of session | $X_s$ Lin (m/s$^2$) | $Y_s$ Lin (m/s$^2$) | $Z_s$ Lin (m/s$^2$) | $X_s$ aw (m/s$^2$) | $Y_s$ aw (m/s$^2$) | $Z_s$ aw (m/s$^2$) | Aeq (m/s$^2$) | Aeq max (m/s$^2$) | azp Lin (m/s$^2$) | awzp (m/s$^2$) | S.E.A.T. $Y_s$ (aw)/awzp, % | A(8) (m/s$^2$) | Time (mn:s) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Horky Bonheur | | | | | | | | | | | | | 1.04 | |
| | Warm-up | 2.14 | 3.27 | 1.72 | 0.89 | 0.56 | 1.35 | 2.01 | 1.35 | 8.41 | 5.25 | 26 | | 1:51 |
| | Training | 2.54 | 4.03 | 2.02 | 1.07 | 0.80 | 1.57 | 2.46 | 1.57 | 10.35 | 6.10 | 26 | | 1:08 |
| Léo de Flo | | | | | | | | | | | | | 1.18 | |
| | Warm-up | 1.50 | 2.37 | 1.68 | 0.66 | 0.48 | 1.38 | 1.80 | 1.38 | 5.31 | 4.32 | 32 | | 7:31 |
| | Training | 2.79 | 5.01 | 2.29 | 1.00 | 0.82 | 1.76 | 2.54 | 1.76 | 9.44 | 6.17 | 29 | | 1:37 |
| | Training | 2.92 | 5.07 | 2.57 | 1.11 | 0.93 | 2.04 | 2.89 | 2.04 | 9.23 | 6.03 | 34 | | 1:44 |

A(8) factor was assessed with a daily duration of 4 h (2 h warm-up and 2 h training).

## 7. Conclusion

The design study, based on a modelling approach, allowed to draw the specifications of a suspension of training sulky in an optimisation process. The production of a prototype was followed by series of validation tests which confirmed the predicted filtering performance.

Despite a drastic attenuation of the vibration at the driver's workplace, in comparison with current production models, the exposure measured on the prototype is close to the limit value enacted in the European vibration directive.

Now the prototype is being circulated among training centres to get feedback from the daily concerned people.

In parallel, ergonomics studies are in progress. One of their topics is the improvement of the driving posture. In particular the design of the footrests remains perfectible. The proposed prototype included innovative footrests that enable the legs to rest at right angles. It is commonly admitted that the posture as well as the vibration are two contributing co-factors to low back pain.

## Acknowledgments

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