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A SIMPLIFIED METHOD TO DESIGN SUSPENDED CABS FOR COUNTERBALANCE TRUCKS

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A "low-frequency" suspension system, placed between the driving cab and chassis of an existing fork lift truck was designed. The aim of this project was to develop a design procedure which is easy to implement and suitable for all types of fork lift trucks. It was also to show how the use of numerical simulation could be helpful to optimize the efficiency of such suspension systems. The cab specifications were: (1) to achieve a vertical vibration attenuation of at least 50% when this truck is tested under severe but realistic conditions, (2) to operate with no specific adjustment for drivers weighing between 60 and 100 kg, (3) to be efficient with a reasonable dynamic stroke (about 3 cm maximum). The suspended cab was modelled using ADAMS software. In the simplified method, the input acceleration signals (at the four fixing points of the cab) were not computed from a vehicle model (chassis and wheels) but directly measured under various driving conditions (passage of two or four wheels over an obstacle with a loaded or unloaded fork lift truck). This model allowed evaluation of the theoretical attenuation, obtained below the driver's seat along the three axes, in comparison with an infinitely rigid suspension. The attenuation ratio was calculated for several values of the characteristics of the suspension components (stiffness and damping). Similarly, for every design value tested, the design constraints were evaluated and at the end of this parametrical study, optimal suspension components were found. Finally, the suspended cab was built according to the results of the parametrical study and measurements subsequently confirmed that the attenuation of vertical accelerations was more than 50%.

1. INTRODUCTION

Counterbalance trucks are widely used in all fields of industry. They are considered as high-vibration vehicles for, unlike cars, they do not have suspension systems and thus transmit all the vibration due to ground irregularities to the operator [1, 2]. Drivers are only protected by the use of suspension seats which are efficient systems if well tuned to the dynamic characteristics of the vehicle. However, suspension seats only attenuate vertical vibration and require individual weight adjustments. An alternative solution is the use of suspended cabs, that is, "low-frequency" suspension systems incorporated between the driving cab and chassis. Such systems are expected to reduce the multi-axis vibration to which the driver is exposed and to operate with no specific weight adjustment. To date, fork lift trucks have not been equipped with low-frequency suspended cabs, because the reduction of vibration emission was not a marketing priority for the manufacturers. This paper reports a simulation-based procedure for the design of suspension components to isolate a cab from vibration. It is called "a simplified method" in reference to past developments from INRS [3], which were more laboratory methodologies than industrial procedures. The proposed procedure is easy to implement and is also reliable. Therefore, it

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can be used as a methodology for the design departments of counterbalance truck manufacturers.

This procedure was tested on a current model counterbalance truck with a load capacity of 2 tonnes. The characteristics of the suspension components were optimized by simulation of realistic driving conditions, taking into account both filtering efficiency and industrial constraints. The suspension was then constructed and mounted on the fork lift truck and the performance of this suspended cab was measured to validate the method.

Various criteria must be taken into account to obtain the optimal compromise between the envisaged performance of the suspension system and the manufacturing limitations:

- It should operate with no specific adjustment for drivers weighing between 60 and 100 kg. The variation in static deflection for these limits of driver weight must remain low (about 3 cm).
- It should operate with a reasonable dynamic stroke (about 3 cm maximum).
- It should reduce vertical vibration by at least 50% when this truck is tested under severe but realistic conditions.
- It should be, as far as possible, of a design that is easy to produce. The solutions retained must allow simple mechanical construction.

2. APPROACH EMPLOYED

2.1. MODELLING THE SUSPENDED CAB

2.1.1. *The cab*

The cab of the 2-tonne fork lift truck was modelled as faithfully as possible by means of the graphics module of the ADAMS software, using the dimensions contained in the manufacturing drawings or obtained by accurate metric mapping carried out directly on the cab (see Figure 1). The software simultaneously calculates the mass, the centre of gravity, and the moments of inertia around the axes intersecting this centre of gravity as well as their direction (Figure 1). For reasons of simplicity of technical development, the guides, springs, and dampers were positioned close to the four points fixing the cab to the chassis.

2.1.2. *The suspension components*

Through experience, the number of degrees of freedom of the suspension was limited to 3

- linear motion along the vertical axis (Z),
- rotation along the fore and aft axes (X), "roll",
- rotation along the lateral axis (Y), "pitch".

The three other degrees of freedom were locked to avoid unnecessary suspension complexity. Indeed, freeing the "yaw" mode (around the Z-vertical axis) serves no purpose. The two other linear modes ("linear motions" along the X-and Y-axis) were replaced at the driver's seat by rotational modes (roll and pitch), as the fixing points are located well below the centre of gravity of the cab and the driver's seat, which induces these two rotational modes. Locking of the yaw and X and Y linear motion modes was achieved on the model by means of four vertical sliding pillars located very close to the existing points for fixing the cab to the chassis. The guiding rings that slide on these axles are connected to the cab by

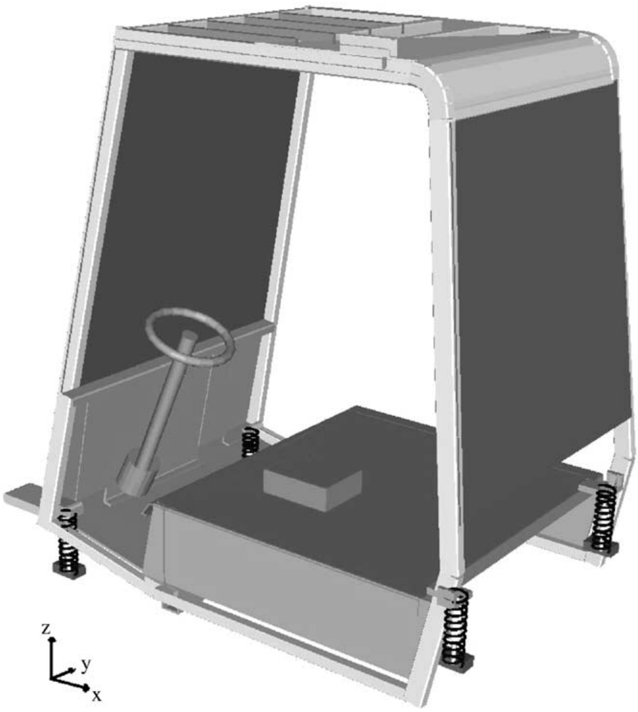

Figure 1. ADAMS model of the suspended cab.

elastomer couplings with varying degrees of flexibility to prevent hyper-stasis, thereby ensuring vertical guidance of the suspension while freeing roll and pitch.

2.1.3. *Input excitations*

Four vertical jacks were located at the four cab fixing points, representing the chassis of the fork lift truck, each being activated by a “motion generator”. These four activators were controlled by the acceleration signals measured on the real truck at the same locations while travelling over an obstacle. It was assumed that the accelerations existing on the chassis at the four cab fixing points were not modified by adding the “low-frequency” suspension system between the chassis and the cab. This basic assumption is acceptable, taking into account the ratio of the mass of the cab ($\sim$ 300 kg) to that of the chassis (3356 kg).

2.2. CRITERIA

As the natural frequency of the fork lift truck was around 6 Hz, the primary aim was to achieve a natural frequency of suspension oscillation mode of 2 Hz. It was also considered that a variation in static deflection of the suspension, of around 5 mm for a variation in driver weight of about 40 kg, would not necessitate a weight adjustment system. The free travel distance envisaged was 3 cm, but as the calculation was made with no end-stop buffers, this allowed calculation of the maximum dynamic stroke necessary during passage over the obstacle for the different experimental configurations.

3. PARAMETRICAL STUDY

A parametrical study was performed to investigate, by calculation, different configurations of stiffness, damping, mass, position of suspension components, etc., the aim being to extract that giving the best theoretical compromise between performance, comfort, and ease of production.

The theoretical performance of the different potential suspension designs was assessed by comparing the vertical (Z), lateral (Y) and fore and aft (X) acceleration values, calculated at a given point (below the driver's seat) of the suspended and non-suspended (rigid) cabs. The "rigid" cab was defined as the suspended cab equipped with four suspension springs, the stiffness of which was multiplied by a coefficient of 10 000.

3.1. INFLUENCE OF THE SPRING STIFFNESS

(a) *On the natural frequencies of the cab*: The three natural modes of suspension vary with the stiffness of the springs. Calculations were performed for seven different values of stiffness (5000, 7500, 10 000, 15 000, 20 000, 25 000 and 30 000 N/m). The results, corresponding to the roll, pitch and vertical modes, are shown in Figure 2.

(b) *On the variation in static deflection*: Figure 3 shows the variation in static deflection for different values of spring stiffness for a 40 kg variation in driver weight.

(c) *On the vibratory transmission ratios*: The transmission ratio is defined as the ratio of the r.m.s. values of the accelerations calculated on the suspended cab and on the rigid cab. These accelerations were calculated at a point located in the cab seat fixation plane vertical to the driver's centre of gravity. The integration was made over a 4s signal period.

As the vertical mode has the highest frequency of the three modes, it is along this axis that the reduction in vibration is least efficient. Figure 4 gives the different transmission ratios obtained for the unloaded fork lift truck crossing a 2×20 cm obstacle at 6 km/h. These testing conditions follow the paradigm of the European test code to measure vibration transmitted to the fork lift truck driver [4].

(d) *On the maximum dynamic stroke*: The curves of Figure 5 show, for the different spring stiffnesses at each fixing point, the maximum dynamic stroke of the suspension of the unloaded fork lift truck, with a driver weighing 80 kg, travelling forward over the obstacle at 6 km/h.

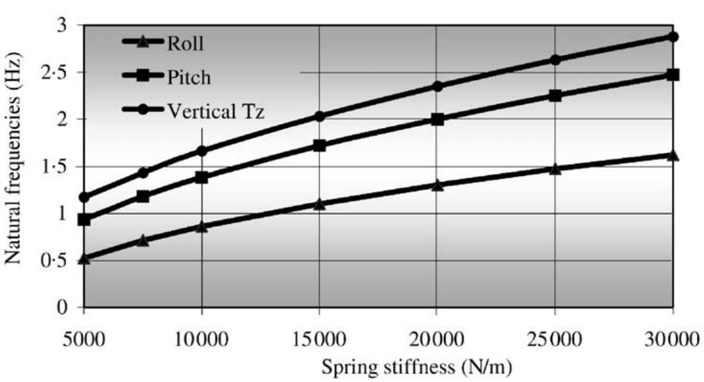

Figure 2. Changes in the three natural suspension modes against the stiffness of the springs.

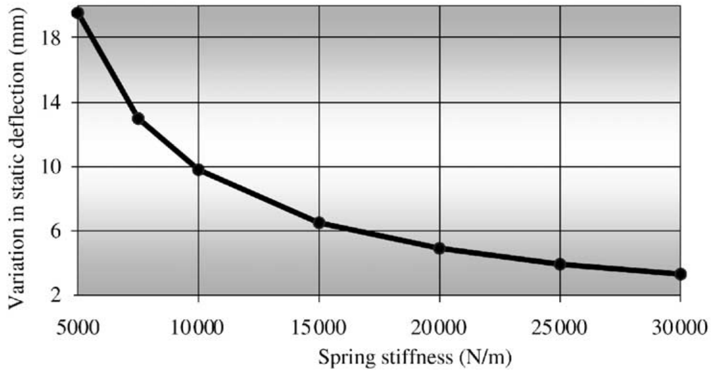

Figure 3. Variation in static deflection for a 40 kg weight difference against the stiffness of the springs.

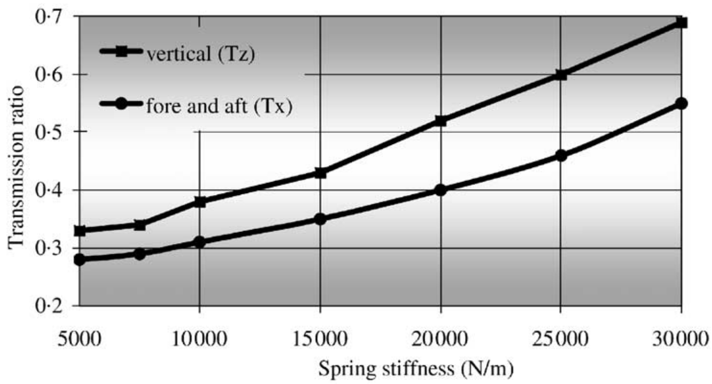

Figure 4. Vertical and fore and aft transmission ratios calculated for the different values of spring stiffness.

3.2. INFLUENCE OF CAB WEIGHT

It is known that the suspension's performance is linked to the ratio between the excitation frequency and the natural frequency of its modes, in particular, its vertical mode [5]. The closer this ratio is to 1, the greater the reduction of vibration by the suspension. For a given cab weight, this implies employing low-stiffness springs, which have the drawback of having a significant variation in static deflection for drivers of different weights. One solution for overcoming this drawback is to increase the stiffness of the springs and the cab mass by the same ratio, which retains low natural frequencies in order to achieve good filtering results. A calculation was performed with a mass, representative of a battery box of about 820 kg, incorporated under the cab and a spring stiffness of 65 000 N/m. Under these conditions, the same natural vertical frequency (2·35 Hz) was obtained as with the suspended cab alone with four springs with a stiffness of 20 000 N/m and therefore, in theory, identical performance along the vertical axis was achieved. However, the variation in static deflection for a 40 kg variation in driver weight is only 1·5 mm instead of 5 mm.

Table 1 lists the different values of the three vibratory modes, the variation in static deflection and the transmission ratios for a variation in mass of 40 kg calculated for two

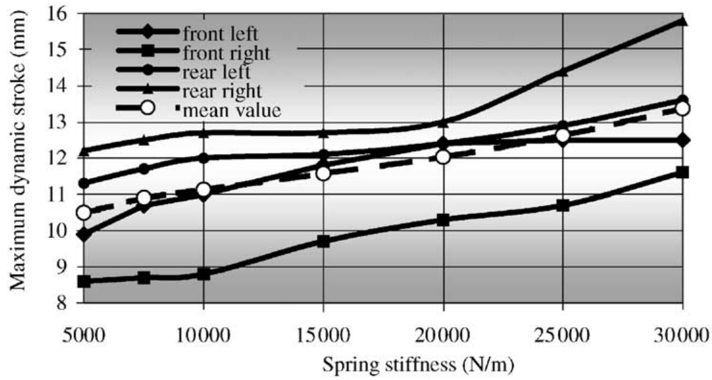

Figure 5. Maximum dynamic stroke calculated at each fixing point and for the different values of spring stiffness.

different experimental suspension configurations:

- suspension of the cab alone, with a spring stiffness of 20 000 N/m and
- suspension of the cab with the addition of a mass (~ 800 kg) and springs with a stiffness of 65 000 N/m, allowing the same natural oscillation mode frequency (2·35 Hz) to be obtained.

The results show that the vertical mode remained unchanged. On the other hand, the first two natural frequencies increased because of the uncoupling effect resulting from the lowering of the centre of gravity.

3.3. DISCUSSION

It was observed that the maximum stroke of the suspension when travelling over the obstacle increases with the stiffness of the springs. This is due to the fact that as the stiffness increases, so do the natural frequencies of the modes, particularly those corresponding to pitching, and approach the excitation frequency of the chassis. The resonance of the suspension is reached, where the movement of the cab tends to be out of phase with that of the chassis. The resulting relative displacement increases proportionally to the quality factor of the suspension ($Q = \omega_0/2\pi\,\Delta f$, where ω_0 is the natural frequency and Δf is the -3 dB bandwidth). There is, therefore, an advantage in employing the lowest spring stiffness possible to obtain:

- optimum vibratory performance and
- lowest dynamic stroke,

but with the drawback of having:

- a considerable variation in static positioning, depending on the weight of the driver.

It would appear that a satisfactory compromise may lie in choosing, for the cab suspension, four springs with a stiffness of around 20 000 N/m. Indeed, this configuration would hopefully reduce vibration when the fork lift truck is travelling over an obstacle, by about 50% along the vertical axis and around 60% along the fore and aft axes. At the same time, it would limit the variation in deflection in the average position of this suspension to $\pm$ 2·5 mm between two drivers of 60 and 100 kg.

TABLE 1

Values of the roll, pitch and vertical modes of two experimental cab suspension configurations, variation of static deflection for a 40 kg variation in driver weight and transmission ratios corresponding to two different weights

Experimental configuration	Only cab suspended $K = 20\,000$ N/m	Cab + 800 kg suspended $K = 65\,000$ N/m
Roll mode Θ_X (Hz)	1·30	2·21
Pitch mode Θ_Y (Hz)	2·00	3·42
Vertical mode T_Z (Hz)	2·35	2.35
Variation in static deflection Δ_Z (mm)	4·9	1·5
Vertical transmission ratio for a 6 km/h run over a 2×20 cm obstacle	0·515	0·515

These different results were obtained by retaining a reduced damping coefficient close to 0·4, which gives a damping of 1000 N s/m per damper for the suspended cab alone.

One solution, incorporating the batteries into the weight of the cab, allows a reduction of the higher vibration to be obtained along the vertical axis, whilst achieving a smaller variation in average height for drivers weighing between 60 and 100 kg. However, this solution was not retained for two main reasons: (1) it would increase manufacturing difficulties due to the higher forces to be taken into account in the definition of the guides and (2) it would cause potential problems of handling and stability when cornering on account of the disconnection of a significant mass from the chassis.

After examination of the different results obtained by simulation, the choice of spring characteristics was fixed at 20 000 N/m to develop the cab suspension.

The effects of damping on the dynamic properties of suspension systems are well known; increasing damping reduces the dynamic stroke of the springs and thus prevents end-stop impacts. On the other hand, this affects the filtering performance. The damping values were not optimized because the manufacture of suitable dampers is not as easy as for springs. We took into account reasonable values, i.e., common damping values for suspension seat dampers.

4. TECHNICAL DEVELOPMENT

The cab was isolated from the chassis by springs supported by inserts fitted to the front on the left and right bumpers and directly on the rear counterweight. The stiffness of these compression springs was 20 000 N/m. They comprised 10 spirals, had a free length of 145 mm, an internal diameter of 41 mm, and a wire diameter of 6·5 mm. Vertical guides were fitted rigidly to the shock absorbers, on which smooth bearings slide. These were linked to the cab by flexible elastomer elements with sufficient transverse stiffness to absorb guide axle alignment faults and to allow rotational roll and pitch movements while locking low-frequency transverse and yaw movements. The springs can be pre-loaded during installation by adjusting the nuts and screw intended for this purpose. Current model dampers for suspension seats were used. For such dampers the damping value was estimated at 3200 N s/m. Details of the assembly developed for the front suspension can be seen in Figure 6; the rear suspension operates on the same principle and is shown in Figure 7.

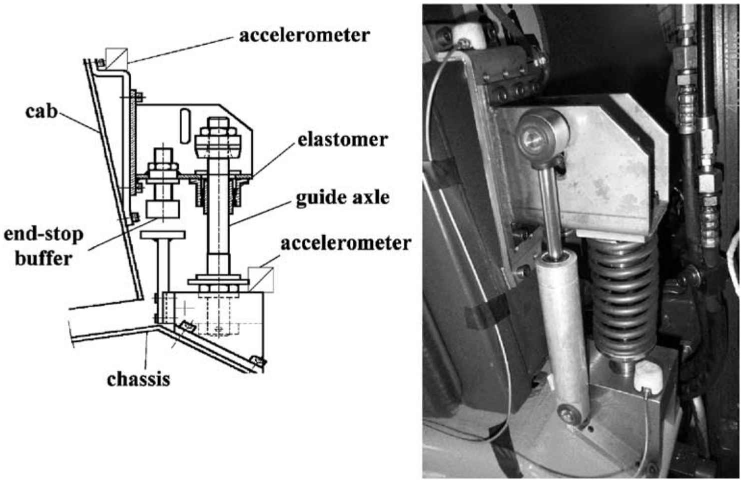

Figure 6. Diagram and view of the front suspension components.

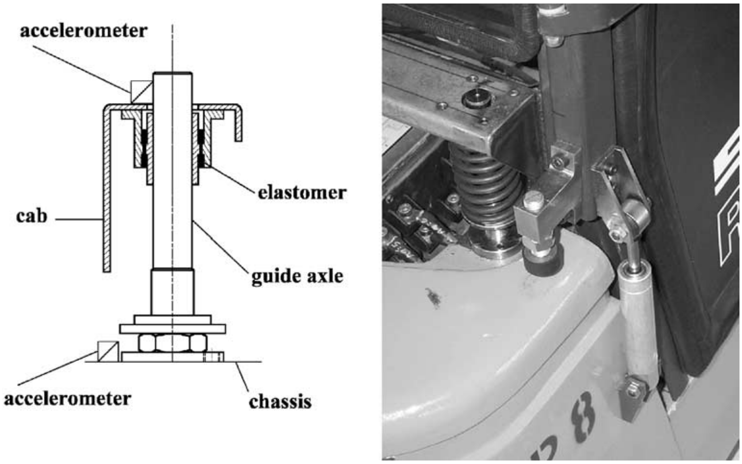

Figure 7. Diagram and view of the rear suspension components.

5. TESTS AND COMPARISON BETWEEN MEASUREMENTS AND SIMULATION

5.1. TEST PROGRAMME

The suspension defined above was installed on the 2-tonne fork lift truck and vibration measurements were taken during the forward and reverse passages of the truck over a 2×20 cm obstacle (a view of the obstacle is given in Figure 8) at two constant speeds, one slow (about 3 km/h) the other faster (about double). The aim of these measurements was to verify whether the model used to predict the performance of the suspension is reliable and that the basic assumptions were acceptable.

The results obtained with the suspended cab were compared to the results with the rigid cab. The rigid configuration was realized by fastening the suspended cab to the chassis at the original fixing points with wedges forced between the cab and the chassis.

Figure 8. View of the suspended cab.

Pairs of accelerometers were placed at the front right (see Figure 6) and rear left, one on the chassis and the other on the cab. A fifth sensor was installed at the base of the driver's seat along the vertical axis as close as possible behind the seat, to evaluate the attenuation of vibration provided by the suspension.

The vibrations measured on the chassis were entered into the model to activate the corresponding "motion generators"; it was assumed that the left and right accelerations were identical since the front or rear wheels travel over the obstacle at the same time. The results obtained on the cab by calculation were compared to those of the corresponding measurements to verify the validity of the model.

5.2. RESULTS

Figures 9(a) and 9(b) show the time histories of the accelerations measured and calculated on the cab, front right (Figure 9(a)) and rear left (Figure 9(b)), with the truck travelling forward over the 2×20 cm obstacle on four wheels at about 6 km/h. A close agreement can be observed between the measurements and calculations at the same points.

To check the filtering efficiency of the suspension, the vertical acceleration was measured under the driver's seat for a forward and a reverse run over the 2×20 cm obstacle at 6 km/h. For each of these runs, the acceleration measured was compared to the acceleration calculated at the same place for a rigid cab. This calculation was performed with ADAMS by entering the acceleration time histories measured at the four fixing points on the chassis and the precise location of the driver's seat. Table 2 shows the resulting transmission ratios for the forward and reverse runs. It can be noted that the initial goals were achieved; the

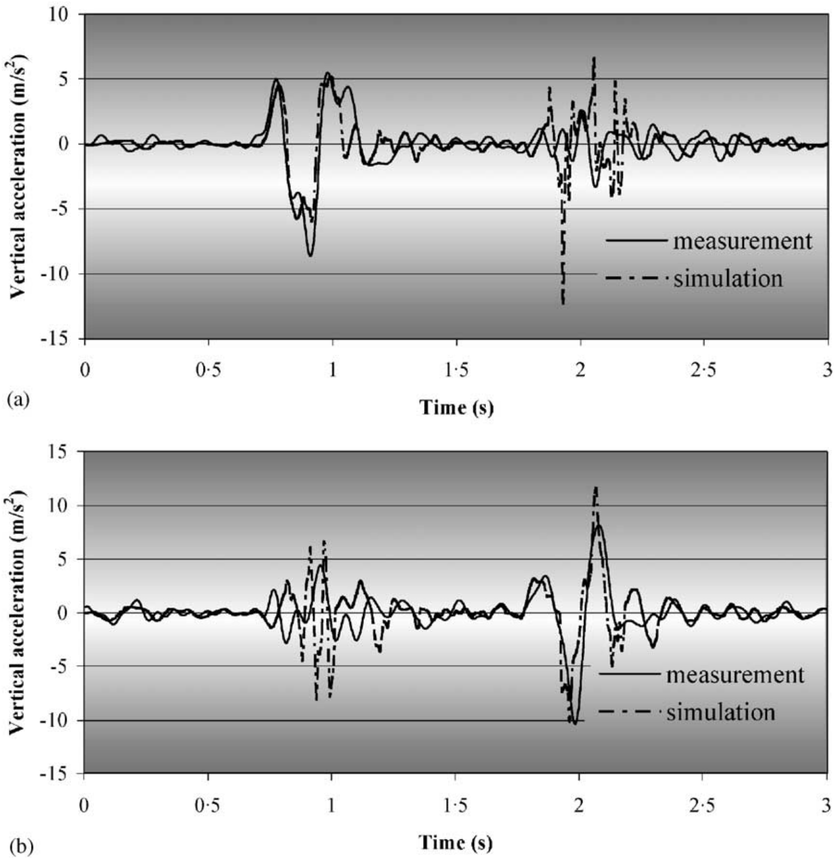

Figure 9. Comparison between predicted and measured vertical acceleration, when running over a 2×20 cm obstacle at 6 km/h: (a) front left side; (b) rear left side.

vertical acceleration at the driver's place on the suspended cab was approximately half that of the rigid cab.

6. CONCLUSIONS

The aim of this study was to test a simulation-based procedure to develop suspended cabs for fork lift trucks. In terms of filtering performance, the objective was to provide an attenuation of about 0·5 of the vibration produced by the truck travelling over a defined obstacle (2×20 cm) at about 6 km/h, assuming that such testing conditions are representative of typical excitations under real conditions of use.

The method used consisted of:

- on the basis of the design drawings, modelling the cab as faithfully as possible in the graphics module of a modelling software program to calculate its mass, its inertia, and the position of its centre of gravity.
- measuring the vibration levels at the fixing points of the cab rigidly fixed to the chassis with the four wheels of the truck travelling over a defined obstacle (2×20 cm), and introducing them into the model. It was assumed that the levels measured on the chassis

TABLE 2

r.m.s. values measured on the rear left chassis and on the cab at the base of the driver's seat, and corresponding transmission ratio (speed = 6 km/h; 2 × 20 cm obstacle)

Direction	RL chassis measurement (m/s^2)	Driver measurement (m/s^2)	Transmission ratio
Forward	3·26	1·75	0·53
Reverse	3·03	1·78	0·58

of the truck are only slightly modified by uncoupling the dynamic behaviour of the cab and chassis during installation of the suspension system.

- using the numerical model to optimize the characteristics of the suspension components.

The method was validated by manufacturing and installing the "optimal" suspension system between the truck and the cab. It was ascertained that the time histories measured on the cab were in agreement with those predicted by the model. It was also ascertained that the desired attenuation of about 0·5 was reached.

It was shown that, with little effort and low cost, very efficient suspended cabs could be designed. Better performances than those presented in this paper could be expected by testing other combinations of parameters such as the location of the fixing points or mass addition.

Nevertheless, this method is well adapted to manufacturers' design departments because they already have simulation tools.

In addition, in many cases the installation of a suspension does not necessitate specific arrangements, because the cab is fixed to the chassis with rubber mountings. These need only be replaced with "low-frequency" components.

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