

# TO STUDY SOME ASPECTS ON THE VIBRATION SYSTEM OF THE 8 SEAT MINIBUS PRODUCING AND ASSEMBLING IN VIET NAM AND THE INFLUENCE OF SOME NONLINEAR PARAMETERS ON IT.

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ABSTRACT - Now a day in Vietnam there are many auto-companies, not only the JVs company and the domestic company producing and assembling automobile. Every year over 80,000 vehicles are made and sold out. The 8 seat minibus is one.

The group of young engineers of Hanoi University of Technology is now working for the state research work KC05.DA13 on designing the 8 seat minibus.

In this technical paper the authors would like to present some results of the calculation and experiments of the vibration system for the above minibus and the influence of some nonlinear parameters on it by using Matlab\_Simulink 6.5.

1. INTRODUCTION - Vibration is one of important criteria to assess the quality of automobiles in designing and testing processes. Many methods have been set up to evaluate the quality of automobile's vibration system, among which, digital simulation is extremely effective for not only producing correct and reliable results in a short period of time but also reducing the high cost of verification experiments. With specialized functions such as programming, processing numbers and graphs for simulation, analyzing dynamic systems, etc, Matlab\_Simulink is a highly practical software for resolving technical and vibration problems. For that reason, apart from practical experiments, we chose to use this software to carry out simulations with the purpose of studying and verifying the vibration mechanism for 8 seat minibuses manufactured at Saigon Automobile Factory (SAGACO) - Vietnam.

## 2. MAKING VIBRATION MODEL

### 2.1. Some particularity of 8 seat minibuses

Minibus vehicle is an 8 seated with full load of 1555 kg and independent suspension systems designed of Mc. Pherson type with coil spring and hydraulic damper. There are some specifications of this vehicle and vibration system:

Specifications	Value	Outfit
Overall weight, unloaded	975	kg
Dimensions: length/width/high	3635/1475/1895	mm
Treads (Front/rear): Bt/Bs	1,28/1,29	m
Distance, centroid to rear axle: b	1060	mm
Distance, centroid to front axle: a	1290	mm

### 2.2. Suppositions when making model

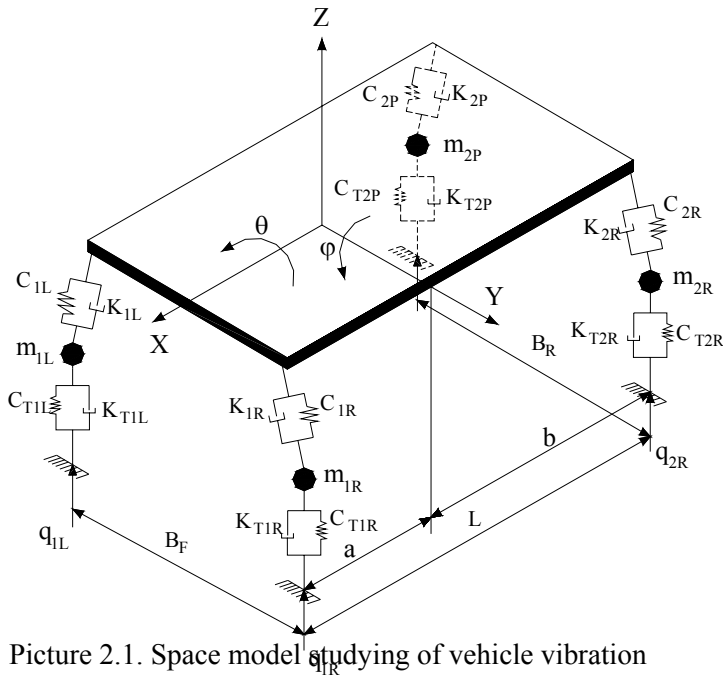
- Dynamic system illustrates vibration of 2WD vehicle in the form of space model.
- Vehicle's body (sprung mass) is supposed to be vehicle's vertical axis-symmetric
- Sprung mass is considered as a flat sheet with mass M placed at center T. Movement of the sprung mass is a combination of 3 movements: linear displacement on erection axis Z crossing center T, respective with generalized co-ordinate Z, pitching around horizontal axis Y (axis crossing center T and

perpendicular to vertical surface), respective with generalized co-ordinate  $\varphi$  and rolling around vertical axis X (axis crossing center T and perpendicular to horizontal surface of vehicle's body), respective with generalized co-ordinate  $\theta$ . Inertia torques of sprung mass with regard to horizontal and vertical axis crossing its center are respectively  $J_y$  and  $J_x$ .

- Unsprung mass respectively at front axle  $m_1$  and rear axle  $m_2$  is linked to road through spring and damper, characterized by stiffness  $C_T$  and damping  $K_T$  of tire. Front (rear) axle is divided into 2 independent masses  $m_{1L}, m_{1R}$  ( $m_{2L}, m_{2R}$ ) which linear displacement in erection axis Z, respective with generalized co-ordinate  $\zeta_{1L}, \zeta_{1R}$  ( $\zeta_{2L}, \zeta_{2R}$ ).

- Vehicle moves equally.

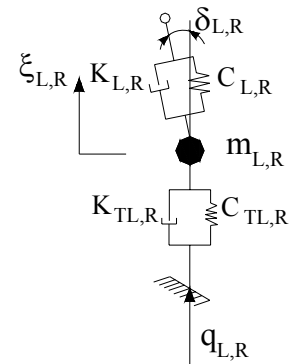
### 2.3. Vibration model



Picture 2.1. Space model of studying of vehicle vibration

Notations of model:

- Axis Z: erection axis crossing center of vehicle's body
- Axis X: vertically crosses center of vehicle's center and lies on the vertical surface of vehicle
- Axis Y: horizontal axis crossing center of vehicle's body and lies on the horizontal surface of vehicle
- Z: vertical displacement of center of vehicle's body
- $\varphi$ : displacement angle (pitching) of vehicle's body around horizontal axis Y
- $\theta$ : displacement angle (rolling) of vehicle's body around vertical axis X
- $\zeta_{1L,R}$ : vertical displacement of center of front unsprung mass (L: left; R: right)
- $\zeta_{2L,R}$ : vertical displacement of center of rear unsprung mass (L: left; R: right)
- $C_{1L,R}$ : stiffness of front suspension (L: left; R: right)
- $C_{2L,R}$ : stiffness of rear suspension (L: left; R: right)
- $K_{1L,R}$ : damping coefficient of front suspension (L: left; R: right)
- $K_{2L,R}$ : damping coefficient of rear suspension (L: left; R: right)
- $C_{T1L,R}$ : stiffness of front tires (L: left; R: right)
- $C_{T2L,R}$ : stiffness of rear tires (L: left; R: right)
- $K_{T1L,R}$ : damping coefficient of front tires (L: left; R: right)
- $K_{T2L,R}$ : damping coefficient of rear tires (L: left; R: right)
- L: long base of vehicle
- a, b: co-ordinate center of vehicle
- $B_F, B_R$ : distance of two wheels in the same axle (F: front; R: rear)



Picture 2.2. Vibration model of unsprung mass

- $Z_{1L,R}, Z_{2L,R}$ : contacting points of suspension system with front and rear vehicle's body (L: left; R: right)
- $\xi_{1L,R}, \xi_{2L,R}$ : contacting points of front and rear unsprung mass with suspension system and tires (L: left; R: right)
- $\delta_{L,R}$ : Damper mounting angle (L: left; R: right)

#### 2.4. Establishing differential set of equations describing fluctuation

The minibus's vibration model includes 6 objects: body, left and right unsprung front masses, left and right unsprung rear masses, and road surface. So we have 7 differential equations corresponding to 7 generalized coordinates referred to above.

Differential set of equations can be made based on the use of equation Lagrange Type II or D'Alambe principle. In this research we use the second method: D'Alambe principle.

According to D'Alambe principle:  $\vec{F} + \vec{F}_{qt} = 0$

In which:  $\vec{F}$  is a combination of external force placed on body;  $\vec{F}_{qt}$  is inertia force.

$$\begin{aligned}
& -m_{1L}\ddot{\xi}_{1L} + [F_{C1L} + F_{K1L}] + [F_{CT1L} + F_{KT1L}] = 0 \\
& -m_{1R}\ddot{\xi}_{1R} + [F_{C1R} + F_{K1R}] + [F_{CT1R} + F_{KT1R}] = 0 \\
& -m_{2L}\ddot{\xi}_{2L} + [F_{C2L} + F_{K2L}] + [F_{CT2L} + F_{KT2L}] = 0 \\
& -m_{2R}\ddot{\xi}_{2R} + [F_{C2R} + F_{K2R}] + [F_{CT2R} + F_{KT2R}] = 0 \\
& -M\ddot{Z} - [F_{C1L} + F_{K1L}] - [F_{C1R} + F_{K1R}] - [F_{C2L} + F_{K2L}] - [F_{C2R} + F_{K2R}] = 0 \\
& -J_y\ddot{\varphi} + a[F_{C1L} + F_{K1L}] + a[F_{C1R} + F_{K1R}] - b[F_{C2L} + F_{K2L}] - b[F_{C2R} + F_{K2R}] = 0 \\
& -J_x\ddot{\theta} + \frac{B_F}{2}[F_{C1L} + F_{K1L}] - \frac{B_F}{2}[F_{C1R} + F_{K1R}] + \frac{B_R}{2}[F_{C2L} + F_{K2L}] - \frac{B_R}{2}[F_{C2R} + F_{K2R}] = 0
\end{aligned} \tag{2.1}$$

\* Transmission force from the left and right suspended front systems

$$F_{C1L} = \frac{1}{\cos \delta_{1L}} C_{1L} (Z - a\varphi - \frac{B_F}{2}\theta - \xi_{1L}) \quad F_{K1L} = \frac{1}{\cos \delta_{1L}} K_{1L} (\dot{Z} - a\dot{\varphi} - \frac{B_F}{2}\dot{\theta} - \dot{\xi}_{1L}) \tag{2.2}$$

$$F_{C1R} = \frac{1}{\cos \delta_{1R}} C_{1R} (Z - a\varphi + \frac{B_F}{2}\theta - \xi_{1R}) \quad F_{K1R} = \frac{1}{\cos \delta_{1R}} K_{1R} (\dot{Z} - a\dot{\varphi} + \frac{B_F}{2}\dot{\theta} - \dot{\xi}_{1R})$$

\* Transmission force from the left and right suspended rear systems

$$F_{C2L} = \frac{1}{\cos \delta_{2L}} C_{2L} (Z + b\varphi - \frac{B_R}{2}\theta - \xi_{2L}) \quad F_{K2L} = \frac{1}{\cos \delta_{2L}} K_{2L} (\dot{Z} + b\dot{\varphi} - \frac{B_R}{2}\dot{\theta} - \dot{\xi}_{2L}) \tag{2.3}$$

$$F_{C2R} = \frac{1}{\cos \delta_{2R}} C_{2R} (Z + b\varphi + \frac{B_R}{2}\theta - \xi_{2R}) \quad F_{K2R} = \frac{1}{\cos \delta_{2R}} K_{2R} (\dot{Z} + b\dot{\varphi} + \frac{B_R}{2}\dot{\theta} - \dot{\xi}_{2R})$$

\* Elastic force, wheel damper:  $F_{CT}, F_{KT}$

#### 2.5. Non-linear factors in the model

##### 2.5.1. Non-linear physical factors

To be presented in this part are several non-linear physical factors such as damper resistance and wheel separation coefficients, they are strong non-linear factors which affect directly the automobile vibration quality.

##### 2.5.1.1. Damper resistance coefficient

Damper resistance coefficient is not a linear but non-linear factor; its value depends on the damper's vibration speed comprising 4 different phases: strong compression, light compression, strong bounce, and light bounce. The resistance coefficients in strong compression and bounce phases, moreover, change continuously (depending on the open rate of strong compression and strong bounce valves).

$$K = \begin{cases} K_{tm} & \text{if } v \leq -0,3 \\ K_{tn} & \text{if } -0.3 < v \leq 0 \\ K_{nn} & \text{if } 0 < v \leq 0,3 \\ K_{nm} & \text{if } v \geq 0,3 \end{cases} \quad (2.4)$$

$K_{tm}$ ,  $K_{nm}$ ,  $K_{nn}$ ,  $K_{tn}$ : Resistance coefficient in strong bounce phase, strong compression phase, light compression phase and light bounce phase

### 2.5.1.2. Wheel separation

Wheel separation accident happens when a wheel goes up in its compression stroke. It is characterized by the gradual reduction of reaction force that the ground exerted on the wheel. Besides, wheel separation causes reaction force to be cancelled.

Therefore, wheel separation is a physical phenomenon that affects much the force transmission and direction guide ability of automobiles. Wheel separation is represented via the reaction force applied by the ground on the wheel as equation (2.5).

$$F_T = \begin{cases} C_T(\xi - q) + K_T(\dot{\xi} - \dot{q}) & \text{if } \xi - q - f_i < 0 \\ 0 & \text{if } \xi - q - f_i \geq 0 \end{cases}$$

### 2.5.2. Non-linear geometric factors

When the machine runs in rough roads or applies brake, etc, the load applied on the suspension system changes, differing the distance between the body and the axles. This causes the vertical declination angle  $\delta$  of the suspension system to change. Such changes of  $\delta$  according to the load is one of non-linear geometric factors of the suspension system.

If  $\Delta H$  is the relative change between the suspended and unsuspended weights,  $\delta$  is the damper's inclination angle, from picture 2.3, it's possible to build a geometric relationship between  $\Delta H$  and  $\delta$ .

$$\delta = \arctang \left( \frac{L_d \cos \left( \arcsin \left( \frac{OC_1 - \Delta H}{L_d} \right) - OO_1 \right)}{OO_2 + OC_1 - \Delta H} \right) \quad (2.6)$$

In which:

$L_d$ : Length, horizontal arm

$OC_1$ ,  $OO_1$ ,  $OO_2$ : lengths deduced from Mc. Pherson kinetics study diagram

$Z_i$ : generalized coordinate of suspended weight

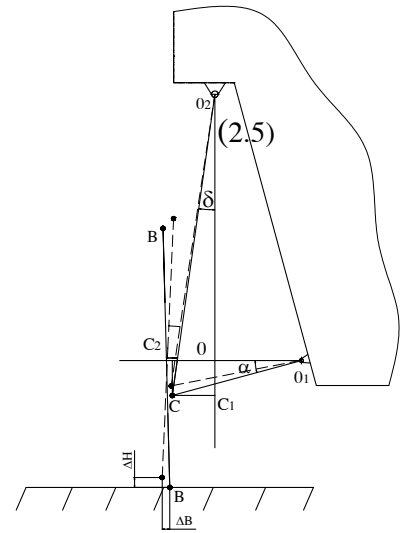
$\xi_i$ : generalized coordinate of unsuspended weight

$$\Delta H = Z_i - \xi_i$$

## 3. SIMULATING VEHICLE VIBRATION USING MATLAB SIMULINK 6.5

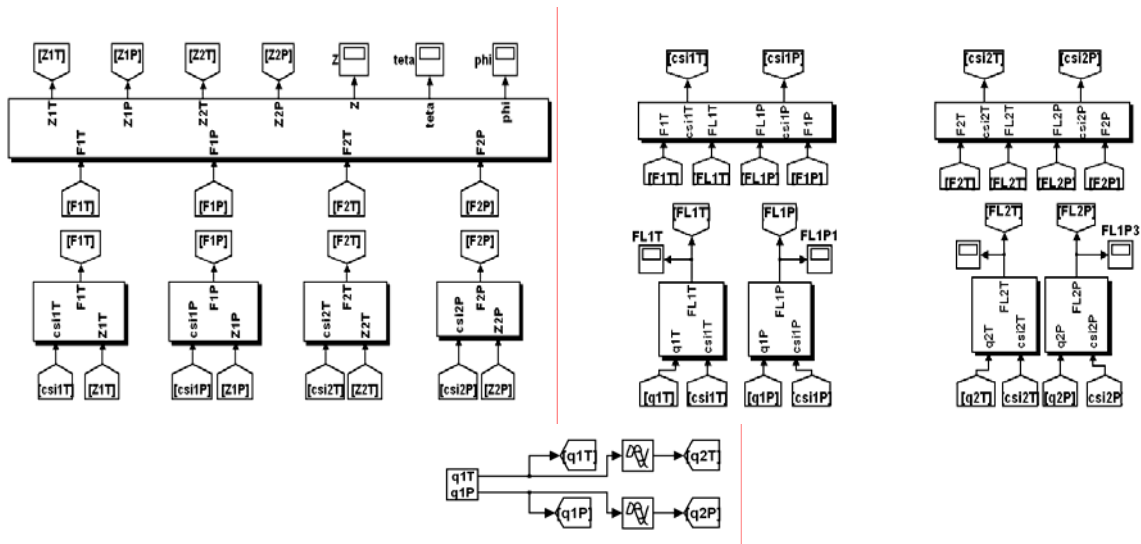
Vibration simulating of two-axle buses based on the set of differential equations representing vibration of the mechanical system and the utilities library available in Simulink of Matlab

Simulation model includes blocks representing machine's body, axles, suspension system, and road surface corresponding to 7 generalized coordinates in set of equations (2.1), along with non-linear factors



Picture 2.3. Geometric diagram of the suspension system

are set into separate modules. This allows processes of studying, surveying or changing parameters to be more independent and precise.

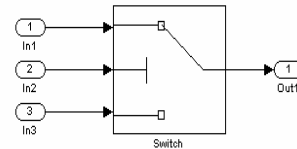


Picture 3.1. Master diagram of 8-seat minibus vibration simulation

### 3.1. Non-linear factors simulation

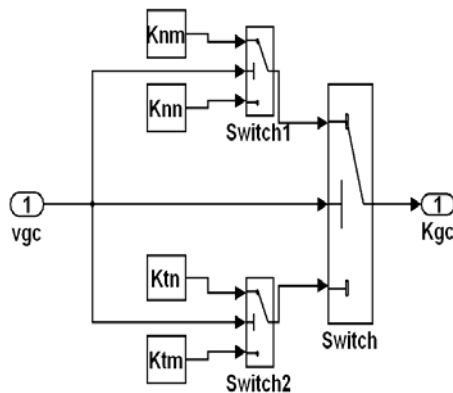
#### a. Damper resistance coefficient

There is a Switch function block in Matlab\_Simulink for simulating some non-linear factors. Such function block acts as a switch that allows parameters (including 3 input and 1 output signals) to be changed regarding status. This will transmit Signal 1 or 3 to output signal in comparison with input signal 2 (controlling signal).

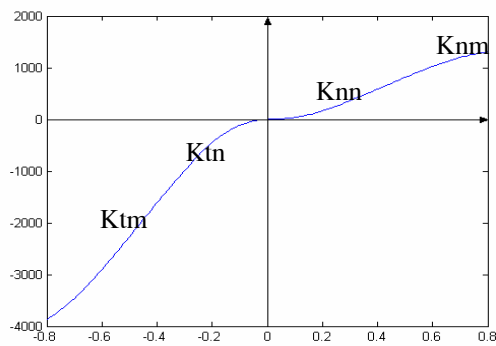


Picture 3.2. Switch function Block

Therefore, we can completely use “Switch” to simulate the nonlinear factor of vibration system. (See picture 3.2)



Picture 3.3: Subsystem for K processing



Picture 3.4: Features of damper resistance coefficient

Switch 1: Control values of damper resistance coefficient in compression strokes

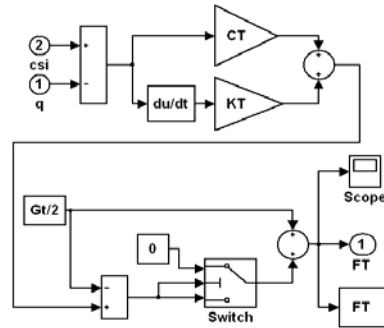
Switch 2: Control values of damper resistance coefficient in bounce strokes.

Switch 3: Control values of damper resistance coefficient on the whole working process

b. Wheel separation

In the process of studying wheel separation as well as using Switch block to deduce the force that the ground exerted on axles via wheels.

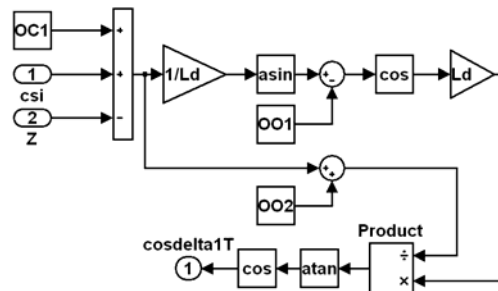
- Value of threshold signal is 0
- When the reaction force exerted on tires is cancelled, wheel separation happens.
- Otherwise, wheel separation does not happen.



Picture 3.5. Subsystem for wheel separation processing

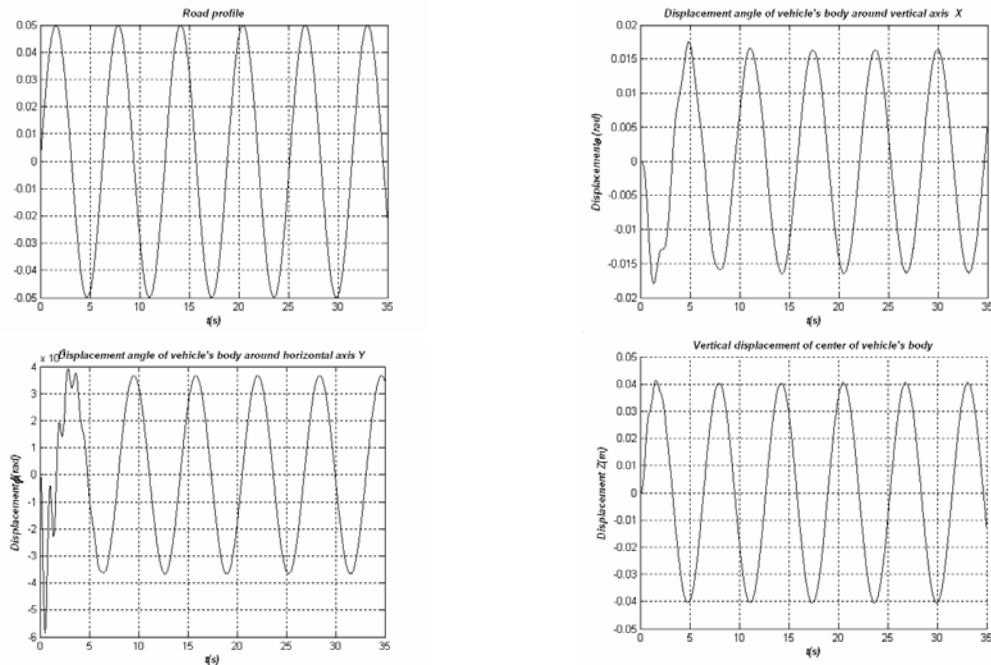
c. Non-linear geometric

From equation (2.6) which shows the relationship between  $\delta$  and  $\Delta H$  it's possible to use basic blocks of Matlab\_Simulink so as to build 4 modules deciding corresponding  $\delta$  angles for each wheel.



Picture 3.6. Module showing  $\delta$  and  $\Delta H$  relationship

3.2. Several results of simulation study



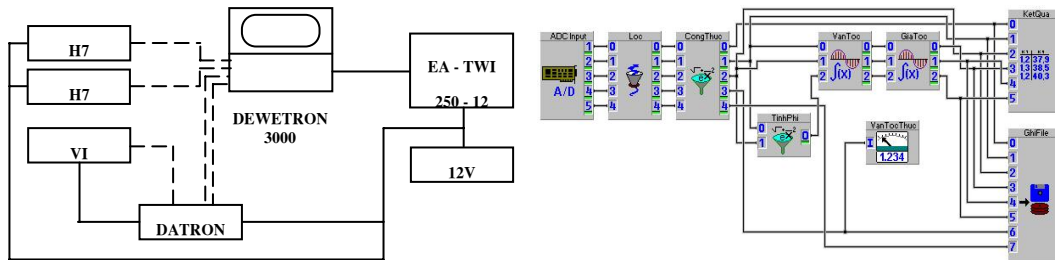
Picture 3.7. Road profile and displacement of vehicle's body around axis X, Y, Z.

## 4. EXPERIMENT ON VIBRATION OF 8 SEAT MINIBUS

### 4.1. Equipments

Equipments include:

- One Dewetron 3000 computer and two software DLSX and DasyLab
- One speed sensor V1
- Two displacement sensors H7
- One transformer EA-TWI 250-12



Picture 4.1. Assembling diagram of measuring equipments

- Transformer EA-TWI 250-12 transforms 220V (AC) of PC to 12V (DC) for the battery which is used for H7 and DATRON microprocessor.
  - Two H7 sensors are installed into two ends and two sides of the machine to measure displacements to the front, rear, left, and right.
  - V1 Sensor transmits signal to DATRON, from there it is divided into 2 and then transmitted to channel 4, 5.
  - Computer Dewetron and software DLSX and DasyLab
- Measured signals are transmitted to DasyLab as shown in figure 4.1.  
 Processed by mathematic functions, it is possible to have the following intermediate parameters:

$$\text{Pitching angle } \varphi = \arctg\left(\frac{z_1 - z_2}{b}\right) \qquad \text{Rolling angle } \theta = \arctg\left(\frac{z_L - z_R}{b}\right)$$

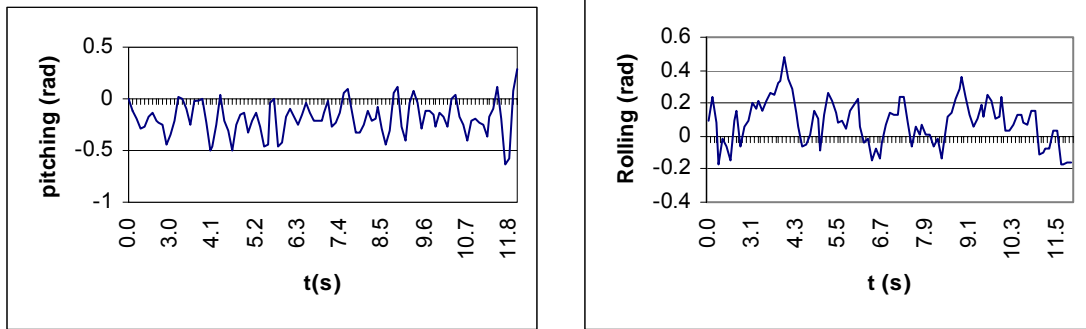
Among which:

- $Z_1(t)$ : front vertical transpose, which is the distance from sensor H7 to ground surface
- $Z_2(t)$ : rear vertical transpose, which is the distance from sensor H7 to ground surface
- $Z_R(t)$ : right vertical transpose, which is the distance from sensor H7 to ground surface
- $Z_L(t)$ : left vertical transpose, which is the distance from sensor H7 to ground surface
- $b$ : Distance between two sensors H7

### 4.2. Some experiment images



#### 4.3. Some experiment results



Picture 4.2. Simulation results of Rolling and Pitching

Experiment results, though are not the same as simulation results due to different input stimuli, show that it is possible to apply experiment procedures to survey minibus vibration in practical operation as well as in compatible simulation and experiment conditions.

#### 5. CONCLUSION

The above results show that Matlab\_Simulink is an effective tool for studying automobile vibration. The simulation of vibration that takes into account non-linear physical and geometric factors allows us to design and assess automobile vibration quality in a faster and more precise way. From there, structural parameters could be altered to improve vibration quality.

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