

Enhancement of Vehicle Handling Based on Rear Suspension Geometry Using Taguchi Method

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ABSTRACT

Studies have shown that the number of road accidents caused by rollover both in Europe and in Turkey is increasing [1]. Therefore, rollover related accidents became the new target of the studies in the field of vehicle dynamics research aiming for both active and passive safety systems.

This paper presents a method for optimizing the rear suspension geometry using design of experiment and multibody simulation in order to reduce the risk of rollover. One of the major differences of this study from previous work is that it includes statistical Taguchi method in order to increase the safety margin. Other difference of this study from literature is that it includes all design tools such as model validation, optimization and full vehicle handling and ride comfort tests.

Rollover angle of the vehicle was selected as the cost function in the optimization algorithm that also contains roll stiffness and height of the roll center. In order to form the cost function, five different geometrical factors have been selected as design variables. The ultimate aim is to minimize the cost function by increasing the roll center height and suspension roll stiffness. To run the optimization routine, a rigid rear suspension mechanism used on the 7 m bus has been modeled using Adams/Car software program. Opposite wheel travel analysis has been performed as an optimization test method in order to simulate the vehicle passing over the bump. Then, in order to reach the minimum value of the cost function, statistical Taguchi method was used to perform design of experiments (DOE).

In total, 27 experiments have been performed according to the selected design variables. Therefore, in each different experiment, the roll center height and the roll stiffness were measured. Then, the cost function was calculated and recorded to compare with the future iterations. The attachment points giving minimum cost function value are expected to be the optimal coordinates for installing the suspension mechanism.

CITATION: Sert, E. and Boyraz, P., "Enhancement of Vehicle Handling Based on Rear Suspension Geometry Using Taguchi Method," *SAE Int. J. Commer. Veh.* 9(1):2016, doi:10.4271/2015-01-9020.

INTRODUCTION

In Germany, 5% of the vehicle accidents have resulted because of rollover [1]. The studies conducted in the UK show that 13% of vehicle accidents were caused by rollover [2]. According to research conducted in the U.S, in 2001, the rollover accidents have accounted for 21% of total accidents [3].

While analyzing the cause of accidents, the rollover dynamics must be examined to identify the key vehicle parameters which might have potential for improvement. The trigger of the events causing rollover is often an uncontrolled lane change, abrupt cornering and obstacle avoidance maneuvers with sometimes excessive control inputs. In these cases, excessive lateral acceleration and forces are induced causing the vehicle to roll over. Furthermore, rollover dynamics have also more importance in the commercial vehicle

design, such as buses, because the vehicle inertia is greater and the center of gravity is higher than other vehicles. This condition leads to increased propensity of rollover and makes the vehicle more prone to severe accidents.

There are some active and passive safety systems that improve vehicle rollover dynamics such as anti-roll bar link, active suspension system, and real-time estimation of rollover. Among these, studies have shown that anti-roll bar placed on hydraulic actuator can increase real-time roll angle threshold so that vehicle cannot reach to the safety limit [4]. When hydraulic actuator operated with lateral acceleration measured continuously, roll angle reduction has been observed in [5]. In an active suspension system, vertical forces acting on the wheel are measured in real-time and roll angle threshold can be controlled [6, 7]. In almost all applications, the most accepted and

applied criterion is “Time to Rollover”. When the wheels are off the ground, it is considered that the onset of the rollover has started [8, 9]. Most of the studies on “Time to rollover” criterion are based on measuring real-time roll angle and lateral acceleration of the vehicle in order to predict rollover time.

Suspension roll stiffness (RS) and roll center height (RCH) are the most effective parameters for determining rollover behavior of the vehicle. The reason is that, the roll angle of a vehicle during cornering depends on the roll stiffness of the axle and on the position of the roll center. Different axle layouts may result in different roll angles at the center of gravity [10].

The roll center establishes the force coupling point between the unsprung and sprung masses [11]. The suspension roll center location determines lateral acceleration acting on the vehicle and consequently, changes the dynamic behavior of rollover. Roll stiffness is also one of the major parameters which has significant effect on ride and handling characteristics of vehicle [12].

In brief, within the scope of this article, unlike the other studies that have focused on changing suspension geometry, we include the roll center height and roll stiffness and how they affect the rollover behavior. In other words, this study explores the limits on how much it can reduce the tipping condition. Therefore, in this work, roll center height (RCH) and roll stiffness (RS) have been selected as major parameters for investigation of the rollover behavior. In addition to the evaluation of the two parameters separately, the relative change of the two parameters have been examined in order to observe the changes in cost function. This part can also be considered as small-scale sensitivity analysis.

The five design variables have been determined in suspension geometry in order to investigate their effects on RS and RCH, eventually affecting rollover propensity. Rollover angle equation can be described as a function of the RCH and the RS. Therefore, the parameters of RCH and RS have been evaluated using multibody dynamic system analysis and design of experiments (DOE). DOE is performed using statistical Taguchi method and multibody dynamic system analysis is performed using Adams/Car software program.

As a result of this optimization procedure, the optimal geometric dimensions yielding the minimum value of the cost function also satisfying the minimum rollover angle of the vehicle have been presented.

SUSPENSION GEOMETRY

Suspension compliance was tuned with regard to both handling and ride comfort. Suspension compliance was determined by the purpose of the ride comfort and handling. Actual tests take time to set up and perform, and if there are vehicle parameters that need to be changed between tests, even more time is necessary. In contrast, changing vehicle and suspension characteristics are simple and efficient if a software simulation such as Adams/Car is used. Vehicle parameters presented in the [Table 1](#) were used in optimization methods and simulations.

Table 1. Technical specification of the Midi Bus

Vehicle Parameters	Measurements
Wheelbase	3385 (mm)
Length	7305 (mm)
Width	2282 (mm)
Height	3350 (mm)
Front track width	1914 (mm)
Rear Track width	1650 (mm)
Center of Gravity Height	1250 (mm)
Gross Weight Vehicle	11500 (kg)
Roll Stiffness of the Suspension	34591*10 ⁴ (N-mm/deg)
Roll Center Height of the Suspension	470.5468 (mm)

The suspension system is modeled with rubber bushing, bump stop/rebound stop elements, anti-roll bars, shock absorbers, air springs and connections to the axes and sprung mass. Furthermore, parameters have defined as the non-linear function such as damping characteristics of the shock absorbers, spring stiffness, bump stop/rebound stop damping are integrated into the Adams/Car. The joints connecting the rigid bodies are modeled based on their physical representation. However, not all rigid bodies are kinematically constrained. Rather, bushings are often used to attach suspension components to satisfy the designer's need to improve the ride characteristics of a vehicle. Bushings are characterized by stiffness and orientation in all three axes. Therefore, translational/torsional stiffness of the bushings are integrated into the Adams/Car.

One of the most important features in a simulation is the tire model. The tire parameters involving Pacejka's Magic Formula parameters were taken from the supplier of the tire. Rear shock absorber is modeled and its characteristics as shown in [Figure 1](#).

In order to improve the rollover dynamics, some design parameters can be selected accordingly. These design parameters are mainly Panhard rod length and height, trailing arm length, the locations of shock absorber and air spring attachment points.

It is well known that the handling characteristics of vehicle can be changed by varying position of the center of gravity, suspension stiffness and roll center height of the suspension [13]. Therefore, it is defined a cost function containing suspension roll stiffness and roll center height of the suspension

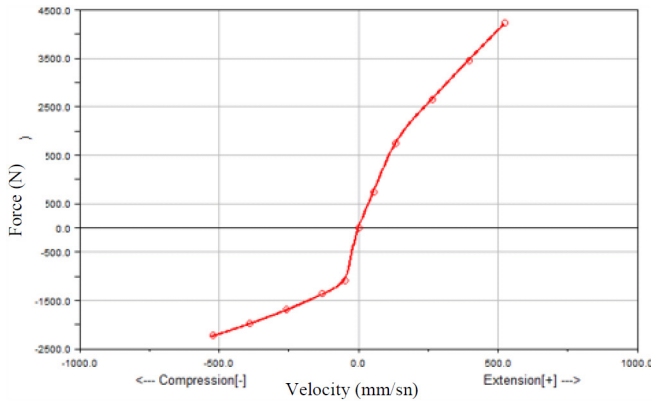


Figure 1. Rear Shock Absorber Force - Velocity Curves

Rollover angle equation selected as the cost function of optimization algorithm, which can be seen in Equation 1.

$$J = \alpha = \frac{M a_y h_s}{K_\phi} \quad (1)$$

As seen from the Equation 1, rollover angle of the vehicle was selected as the cost function of optimization algorithm that contains roll stiffness and height of the roll center.

- α Roll Angle (degree).
- K_ϕ Suspension Roll Stiffness (N-mm/degree).
- M Vehicle Mass (kg).
- a_y Lateral Acceleration (mm/s²).
- h_s The vertical distance between the roll center of the suspension and center of gravity of the vehicle (mm).

As seen from the results in Equation 1, the cost function depends on roll stiffness (K_ϕ), vehicle mass, lateral acceleration and roll center height. Therefore, roll stiffness (RS) and roll center height (RCH) were selected the main vehicle's parameters in order to know how these parameters influences and interacts made it possible to improve the roll angle threshold until the guidelines were fulfilled. The vertical distance between the roll center of the suspension and center of gravity of the vehicle is described as shown in Equation 2.

$$h_s = h - RCH = 1250 - 470.5468 = 779.4532 \text{ (mm)} \quad (2)$$

- h Center of Gravity Height
- RCH Roll Center Height of the Suspension

As seen from the results in Equation 2, h_s is calculated as 779.4532 (mm) in the design/normal value. The parameter study also reveals the interactions between parameters. It is very useful to examine these cross-effects as these interactions can lead to unexpected changes on other parameters. When the car corners a curvy road, the centrifugal force acting on the centre of gravity can be translated to the roll center and down to the tires where the reactive lateral forces are built up. The higher the roll center is the smaller the rolling moment around the roll centre. This rolling moment must be restricted by the springs.

If the roll center height (RCH) and the roll stiffness (RS) are increased, the cost function value will be decreased and will reach the main goal because roll angle is defined as a real time angle of the vehicle rotated about vehicle longitudinal axis. Rigid rear suspensions values and their coefficients were validated by physical test and modeled via Adams/Car and it has been selected for design of experiments (DOE).

A wheel-travel analysis allows looking at how the characteristics of a suspension change throughout the vertical range of motion of the suspension. When the vehicle was cornering, it was seen that right wheel center had a displacement of +30 mm and the left wheel center had a displacement of -30 mm. Therefore, the upper limit of wheel-center displacement relative to the input position was taken as 30 (mm) and the lower limit of wheel-center displacement relative to the input position was taken as -30 (mm). In this case, both roll stiffness and roll center height variation can be investigated. The opposite wheel travel test rig have been illustrated in Figure 2. In the system called as a half vehicle test, the pistons can move independently. In other words, one of the pistons can be moved upwards while the other one can be moved downwards.

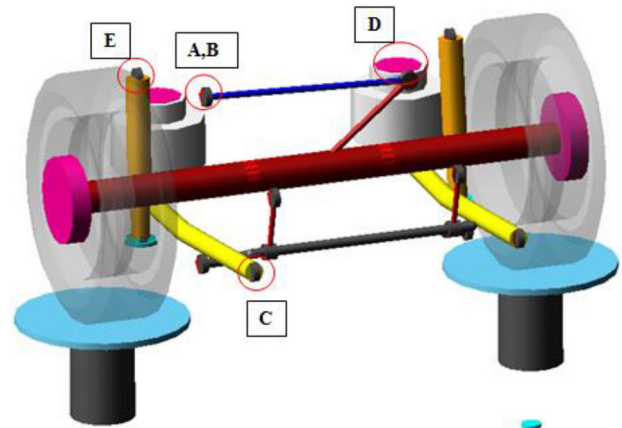


Figure 2. Rear suspension test rig

Rear suspension attachments points containing two air springs, shock absorbers and Panhard rod are accurately represented as shown in Table 2.

Levels are the values of the factors used to carry out the experiment. Specific factors that can be modified have been taken into account in the parameter selection. Therefore, five different design factors that are adjustable on the vehicle have been selected from rear suspension as shown in Table 2. Minimum and maximum values/levels cannot be changed because all values for each factors have been restricted by the suspension design criteria.

As seen from the results in Table 2, Panhard Rod attachment joints were moved in vertical and lateral direction at the chassis. Trailing Arm attachment joints were moved in longitudinal direction at the chassis, air spring and shock absorber attachment joints were moved in lateral direction at the chassis. The levels have been chosen in this manner so that they can be used as the adjustment

levels for the vehicle. The low levels then correspond to one endpoint of the wanted adjustment possibility and the high levels to the other endpoint.

Table 2. Design factors

Design Factors	Symbol	Coordinate	Minimum Value (mm)	Design Value (mm)	Maximum Value (mm)
Panhard Rod Attachment Point (Panhard Rod Length)	A	y (± 50)	302	352	402
Panhard Rod Attachment Point (Panhard Rod Height)	B	z (± 50)	979	1029	1079
Trailing Arm Attachment Point (Trailing Arm Length)	C	x (± 50)	6748	6798	6848
Air Spring Attachment Point	D	y (± 50)	342	392	442
Shock Absorber Attachment Point	E	y (± 50)	460	510	560

VALIDATION OF THE ADAMS/CAR VIRTUAL MODEL

Verification of the virtual model is essential in the development of analytical simulation models because virtual model is providing accurate results. While validation of the virtual model, vertical and lateral accelerations, displacements, pitch and roll angle were measured on the wheel hubs. Next, actual test results were compared with the results of virtual test results previously calculated.

When the instrumentation of the vehicle, position a triaxle (X, Y, Z) accelerometer and displacement sensor were placed on the four different wheel hubs in order to measure the vertical and lateral accelerations that are a parameter representing comfort and handling. V-Box data logger that measures pitch and roll angle should be fixed to an appropriate place on the floor closed the CoG of the vehicle and must remain still during the test.

Vertical and lateral acceleration can be manually measured by the program on the four different wheel hubs. Pitch and roll angle, roll rate can be manually measured by the program on the CoG of the vehicle. The Adams/Car automatically calculates suspension roll center height and suspension roll stiffness of the vehicle pre-determined from the installed length of the suspension system during the test.

The actual and virtual vehicle is driven over the speed bumper 3 times with a vehicle speed 20km/h under the same conditions. Vertical acceleration was measured during the test.

Red line represents simulation results and blue line represents actual test results. It is observed that virtual model provide very accurate results when compared to the results of actual measurements.

Simulation results shows 0.1% deviation from the actual test results in the best case and shows 1.1% deviation from the actual test results in the worst case.

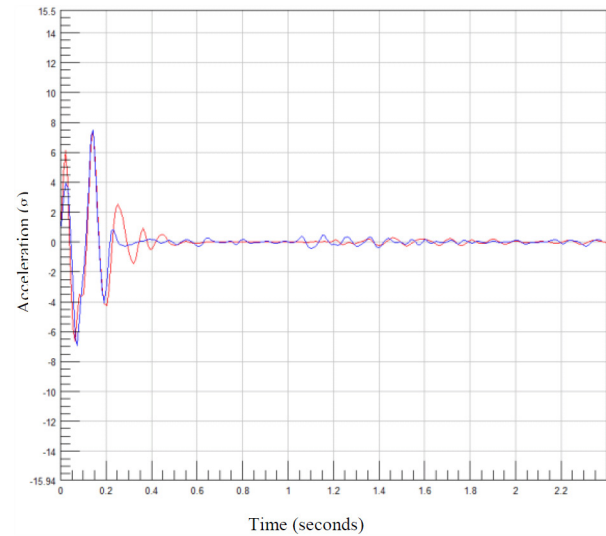


Figure 3. Comparison of the Vertical Accelerations on Virtual and Actual Bus

Two parameters were measured at the time of lane change. Bushings and rebound stop and bump stop parameters of the virtual model were tuned to achieve reasonable correlation. Roll rates measured on the wheel hubs were compared as shown in Figure 4.

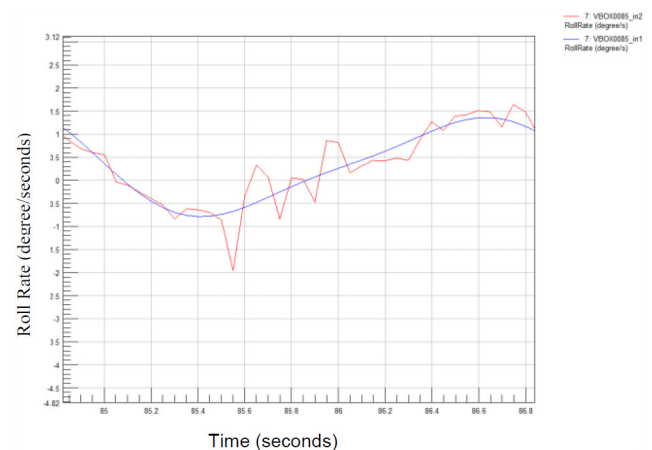


Figure 4. Comparison of the Roll Rate on Virtual and Actual Bus

Lateral accelerations measured on the CoG were compared as shown in Figure 5.

Red line represents actual test results and blue line represents simulation results. However, actual vehicle data were not passed through the filter. As a result of this, white noise increased throughout the data of the test, especially at the peak points. It is observed that virtual model behaves like an actual vehicle.