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Original Research Article

Improvement of the Vehicle Stability Using Suspension Optimization Methods

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Abstract

The number of studies that are supported suspension parameters with statistical methods is very small. However, many studies have been made on the dynamics of vehicle handling. For this reason, in this study, sensitivity analysis was performed using different stiffness values of the leaf spring and roll bar in the front suspension. The effects of the parameters on both static and dynamic analyzes were interpreted. In addition, the statistical t-test and Anova analysis were performed in order to compare means of two or three groups.

Keywords: Vehicle Dynamics, Suspension Optimization, Sensitivity Analysis, Anova

Definitions/Abbreviations

RCH - Roll Center Height of the Suspension

RS - Roll Stiffness

α - Roll Angle

Kø - Suspension Roll Stiffness

h - Center of Gravity Height

J - Cost Function of Optimization Algorithm

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1. Introduction

Heavy commercial vehicles may have stability problems when the driver must perform a quick maneuver to avoid an unexpected obstacle in the road. According to data of General Directorate of Highways, the number of mortal and injured traffic accidents was 162.512 in Turkey in 2014 [1]. 18,032 of the total accidents resulted in rollover and sliding. As shown in Table 1, accidents resulting in rollover and sliding is 10.70% of total accidents. It is the fourth most frequent accident type.

Table 1. 2014 Accident Analysis

KAZA OLUŞ TÜRÜ	Yerleşim Yeri		Yerleşim Yeri Dışı		TOPLAM	
	Kaza Sayısı	%	Kaza Sayısı	%	Kaza Sayısı	%
Yandan Çarpma veya Yandan Çarpışma	43.657	34,50	5.257	12,52	48.914	29,03
Yayaya Çarpma	30.411	24,03	1.364	3,25	31.775	18,86
Yoldan Çıkma	8.122	6,42	15.893	37,86	24.015	14,25
Devrilme, Savrulma, Takla	9.958	7,87	8.074	19,24	18.032	10,70
Arkadan Çarpma	11.756	9,29	4.358	10,38	16.114	9,56
Karşılıklı Çarpışma	8.051	6,36	2.548	6,07	10.599	6,29
Engel/Cisim ile Çarpışma	7.651	6,05	2.824	6,73	10.475	6,22
Duran Araca Çarpma	3.133	2,48	443	1,06	3.576	2,12
Yan Yana Çarpışma	1.865	1,47	371	0,88	2.236	1,33
Araçtan Düşen İnsan	1.029	0,81	197	0,47	1.226	0,73
Hayvana Çarpma	435	0,34	479	1,14	914	0,54
Zincirleme Çarpışma	227	0,18	76	0,18	303	0,18
Çoklu Çarpışma	190	0,15	60	0,14	250	0,15
Araçtan Düşen Cisim	52	0,04	31	0,07	83	0,05
TOPLAM	126.537	100	41.975	100	168.512	100

Vehicles are usually scaled based on a particular vehicle's SSF by the NHTSA. Calculating the SSF is a very compact and meaningful way of indicating rollover risk in single-vehicle crashes. As shown in fig. 1, the lowest-rated vehicles (1 star) are at least four times more likely to roll over than the highest-rated vehicles (5 stars) when involved in a single vehicle crash [2].

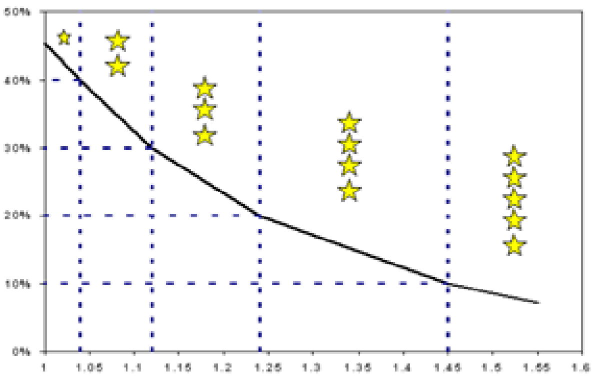

Fig. 1. SSF Score of the Vehicle [7]

In previous studies, active and semi-active suspension are directly prevent overturning [3]. In active suspension systems, the vertical forces are created by electrohydraulic dampers [4, 5]. The semi-active suspension system is based on the removal of a conventional damper located in passive suspension systems, which can be controlled by an actuator and provides a variable coefficient.

The type of suspension and its subsystems such as anti-roll bar, leaf spring and shock absorber were identified as critical in the overall stability of the vehicle [6, 7, 8, and 9]. Studies have shown that mechanical suspension containing leaf spring positively affected the stability of the vehicle because roll stiffness of the suspension is higher than the traditional suspension systems containing air bellow [10, 11]. Air suspension was identified as negatively impacting on stability. Therefore, if the suspension allows more roll to occur (low roll stiffness), the yaw-roll dynamic mode of the vehicle become weak and tendency of vehicles to roll over increases. For this reason, the stability of the vehicle will be improved by increasing the roll stiffness of the suspension. Using this simple relation, in this study, roll angle is reduced as much as possible by increased anti-roll bar and leaf spring stiffness via changing the diameter. Then the results are verified using static and dynamic tests. Stiffness of the anti-roll bar is calculated according to the SAE standards in each experimental analysis [5, 8]. Modeling studies were carried out using the technical features of the vehicle that shared in the Table 2.

Table 2. 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)
H_{CG}	779.43 (mm)
Gross Weight Vehicle	11500 (kg)

2. Virtual Model of the Adams/Car

Adams/Car software program was chosen to simulate static and dynamic tests because of the high level of computational ability as well as the detailed animation capability [11, 12]. A midi bus with 7 m length has been modeled using Adams/Car software program in order to obtain virtual vehicle for further stability analyses.

In the simulation, front suspension attachments points containing two leaf springs, shock absorbers and anti-roll bar are accurately represented. The standard front suspension models of the Adams/Car are shown in Figure 2. In addition to the standard model, leaf spring properties which are taken from supplier are modelled as non-linear characteristics.

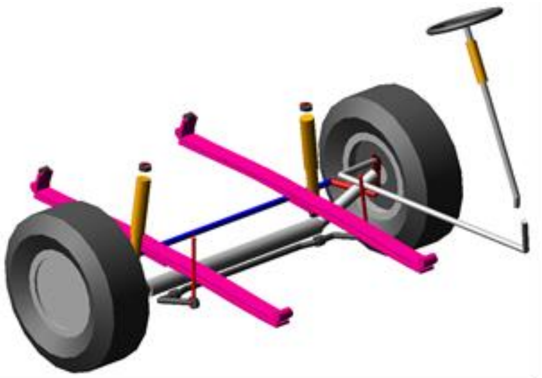

Fig. 2. Front Suspension Model of the Adams/Car

Rear suspension model of the Adams/Car containing two air springs instead of leaf spring, shock absorbers, trailing arm and Panhard rod is shown in Figure 3.

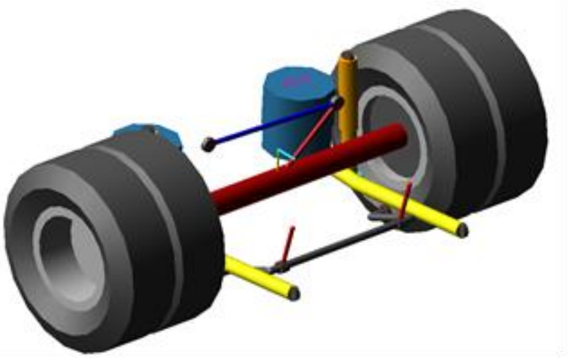

Fig. 3. Rear Suspension Model of the Adams/Car

Panhard rod is located on the rear axle is treated as a lever with the intermediate support at the rear axis and it prevents movement of the rear axle from right to left. Other sub-systems such as engine, brakes, anti-roll bar, chassis and body are also modelled in accordance with actual geometry and non-linear characteristics. Since they are not directly related to rollover-dynamics, they are not detailed herein.

The resulting multi-body system is connected by joints (translation and rotational motion), springs, dampers, and rubber bushings. These components are acted upon by external forces and moments or external control movements and can be described mathematically by a system of differential-algebraic equations (DAE). Full virtual vehicle model shown in Figure 4 has 487 DoF.

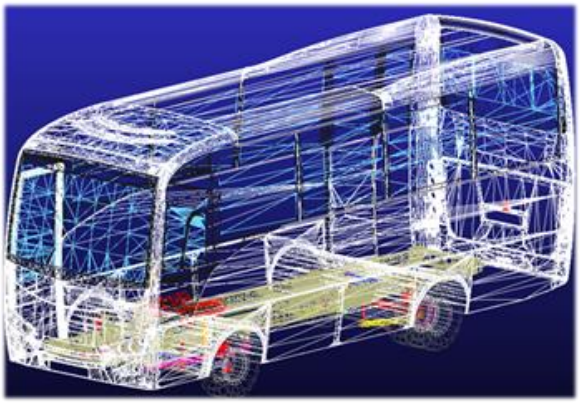

Fig. 4. Virtual Vehicle Model of the Adams/Car

The inertial properties of the virtual vehicle such as the location of center-of-gravity and the inertias are provided to match the actual vehicle. In this way, variable parameters containing force, torque, displacement and acceleration can be integrated to one of the subsystem.

3. Verification of the Adams/Car Model

This section will illustrate the various simulation tests that were performed to ensure that the model responses in an expected and predictable manner. Verification of the virtual model is essential in the development of analytical simulation models because virtual model is providing accurate results. Methodology of the verification of the Adams/Car model involves performing both static and dynamic cases. Furthermore, suspension and steering sub system models are validated prior to overall simulation. Therefore, the virtual model used in this work provides very accurate results.

3.1. Verification of the Virtual Model as a Static

The first step of the virtual model verification

is to specify a shock absorber for the vehicle. The minimum and maximum wheel travels are important in order to define what size of shock absorbers are suitable. In this model, the lengths of an already installed parts were given by the manufacturer, which was 142 millimeters for the front spring/dampers and 161 millimeters for the rear spring/dampers.
Parallel wheel travel test were available in Adams/Car that allowed the research team to analyze the characteristics of a suspension change throughout the vertical range of motion. The damper/shock absorber lengths of the rear and front suspensions were analyzed under the simulation test of parallel wheel travel. Parallel wheel travel test rigs are shown for front and rear suspension in Figure 5 and 6.

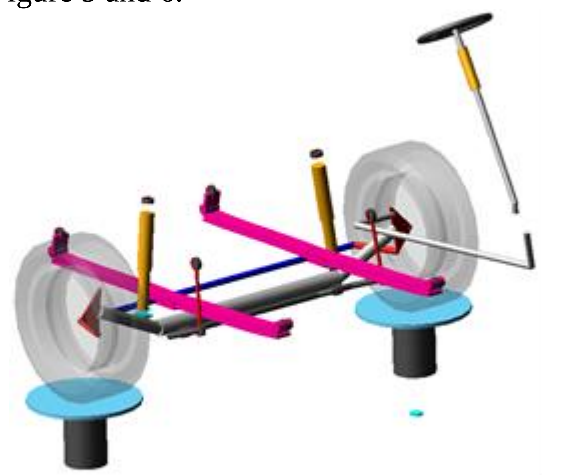

Fig. 5. Front Suspension Parallel Wheel Travel Test Rig

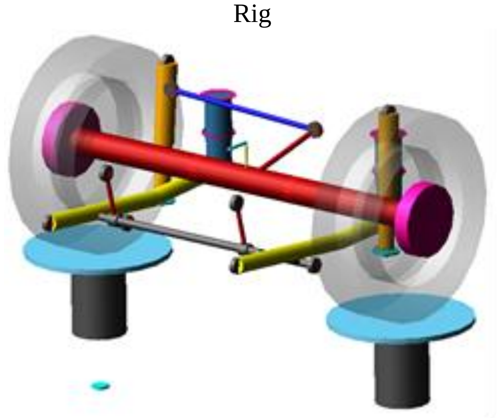

Fig. 6. Rear Suspension Parallel Wheel Travel Test Rig

The rebound of a suspension is the maximum downward displacement at ride height, and likewise, the jounce (4th derivative of position) of a suspension is the maximum upward displacement. The upper limit of wheel-center displacement relative to the input position is 70 (mm) and the lower limit of wheel-center displacement relative to the input position is -90 as shown in Figure 7. The horizontal scale (x axis) represents the response of the rebound and jounce values. The vertical scale (y axis) represents response of the damper length.
It is shown that damper length is measured as + 72.54 mm when vehicle is acting under jounce condition and is measured as - 70.15 mm when vehicle is acting under rebound condition. Front axle damper length of the actual vehicle is restricted as 142 mm. Since the measured values and predicted values agree well, the model is validated and it gives accurate predictions.

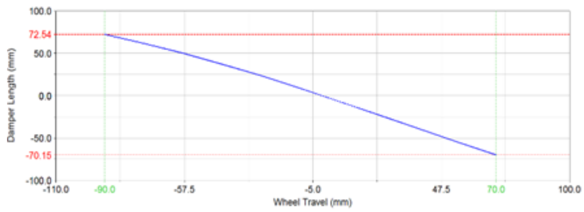

Fig. 7. Damper Length of the Front Suspension

The upper limit of wheel-center displacement relative to the input position is 90 (mm) and the lower limit of wheel-center displacement relative to the input position is -88 as shown in figure 8.
It is shown that damper length is measured as + 73.20 mm when vehicle is acted under jounce condition and is measured as – 87.80 mm when vehicle is acted under rebound condition. Rear axle damper length of the actual vehicle is restricted as 161 mm. the same accuracy level is obtained for the rear suspension model as it was previously obtained for the front suspension model.

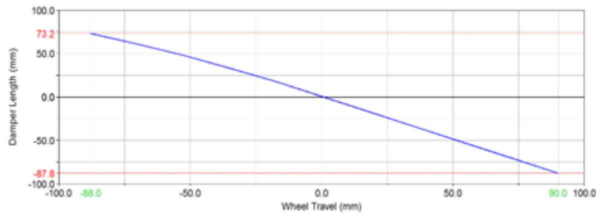

Fig. 8. Damper Length of the Rear Suspension

As the results show, wheel travel and damper length parameters of the virtual model were validated by the data obtained from the parallel wheel travel test rig.
The front and the rear axle loads were

measured at the stage of determining axle loads separately. The vehicle center of gravity "x and y" axes coordinates were identified by measuring of each wheel using load pads. Then, additional loads of each 2450kg were attached above the seats to reach the gross vehicle weight.

Finally, gross weight of the vehicle was measured at each wheel again. In this way, the coordinates of the center of gravity provide very accurate results. These steps were performed in order to determine the axle loads of the actual vehicle and verify axle loads of the virtual vehicle model. Axle loads of the actual and virtual vehicle were compared based on curb and gross weight vehicle conditions as shown in table 2.

Table 2. Axle Loads of the Midi Bus

	Axle	Actual Vehicle	Adams/Car Virtual Vehicle	Percentage Error (%)
Curb Vehicle Weight (kg)	Front Axle	3790	3800	0.26
	Rear Axle	4860	4900	0.82
	Total	8650	8700	0.57
Gross Vehicle Weight (kg)	Front Axle	4100	4130	0.73
	Rear Axle	7000	7040	0.57
	Total	11100	11170	0.63

It is seen that from Table 2, the difference between the simulated and measured values are less than 1% for all axle loads; the simulation model is considered as well-correlated and valid to apply for static tests. Parametric sensitivity analysis and all verification tests are performed when vehicle is loaded with gross vehicle weight. If the loading condition is like this, rollover threshold of the vehicle diminishes. When cornering, lateral load transfer ratio is increased and lateral acceleration acting on the CoG forces the vehicle to roll over. Therefore, the vehicle loaded with gross vehicle weight can be recognized as the worst case.

Tilt table test was performed with actual vehicle in order to measure the roll angle at the center of gravity corresponding to the tilt table angle in the last stage. Furthermore, the roll angle of the virtual model was measured in the same region as shown in Figure 9.

While the tilt table test was performed, the front leaf spring had 20 (kg/mm) as stiffness value and the front anti-roll bar were selected to be Ø38. Body roll angles of the actual and virtual vehicle were compared as shown in Table 3.

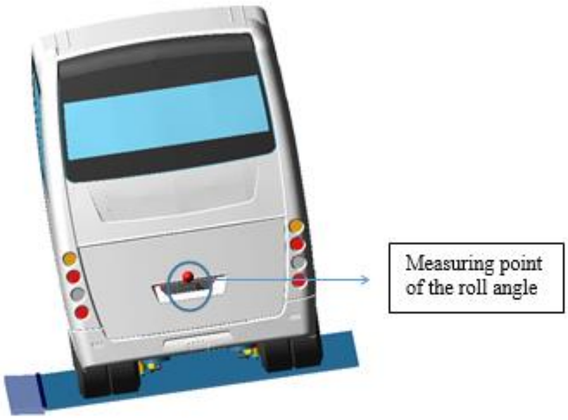

Fig. 9. Virtual Model Tilt Table Test Rig

Table 3. Tilt Table Test Results under Existing Components

Tilt Table Angle (degree)	Actual Vehicle Body Roll Angle (degree)	Virtual Vehicle Body Roll Angle (degree)
6°	7.30°	6.50°
18°	21.60°	19.01°
26.20°	31.60°	28.25°

It is observed that body roll angles of the actual and virtual vehicle are very close. Body roll angle of the actual vehicle is measured as 31.60° degree when the first tire lifts off the table. Tilt table angle is measured as 26.20° degree while the body roll angle was measured. After the vehicle was only verified as static, simulation results indicated 2% deviation from the actual test results at the time of rollover. When the vehicle was verified as dynamics, simulation results indicated a deviation of less than 1% compared to the actual test results.

3.2. Verification of the Virtual Model as a Dynamics

The maneuver with passing over the speed bumper was performed to evaluate the actual and virtual vehicle responses in terms of vertical acceleration and displacement.

In this validation method, overall results such as accelerations and displacements were analyzed instead of a detailed examination of each parameter such as toe, camber caster and kingpin inclination angles. The purpose of compliance design was to correlate the virtual suspension compliance response with the test data. Stiffness of the leaf springs, air springs and damping of the shock absorbers in specific orientations were tuned to achieve reasonable correlation.

While actual vehicle was driven over the speed bumper, vertical accelerations and displacements were measured on the wheel hubs. Next, actual test results were compared with the results of virtual test results previously calculated. Accelerometers and displacement transducers were placed in the corresponding points on the actual test vehicle listed as follows:

- Four 3D accelerometers on wheel hubs.
- Four 3D accelerometers on rigid brackets on the chassis above wheel hubs.
- 3D accelerometer on COG of test vehicle
- Four displacement transducers between wheel hubs and rigid brackets on the chassis.

After the instrumentation of the vehicle, two real rigid speed bumper models were made as shown in Figure 10.

The actual and virtual vehicle is driven over the speed bumper 3 times with a vehicle speed 20km/h under the same conditions as shown in Figure 11.

The suspension parameters of the simulation model are correlated using the actual test data. Vertical accelerations measured on the wheel hubs were compared as shown in Figure 12.

Fig. 10. Rigid Speed Bumper Models

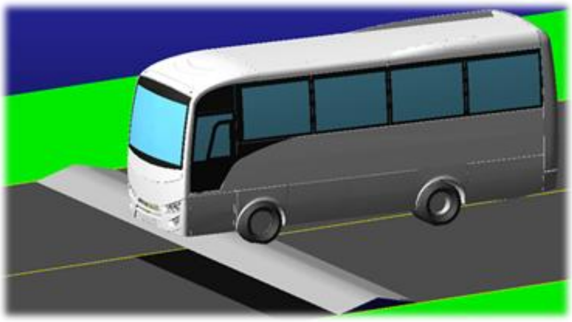

Fig. 11. Virtual Model Passed Over the Speed Bump

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.

Some of the significant parameters affecting the vehicle rollover threshold are lateral acceleration and roll rate. Therefore, these parameters of the virtual model should be verified. ISO lane change maneuver was performed in order to verify lateral acceleration and roll rate parameters. In this maneuver, the vehicle is driven at a speed of 40 km/h.

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 rate measured on the wheel hubs were compared as shown in Figure 13.

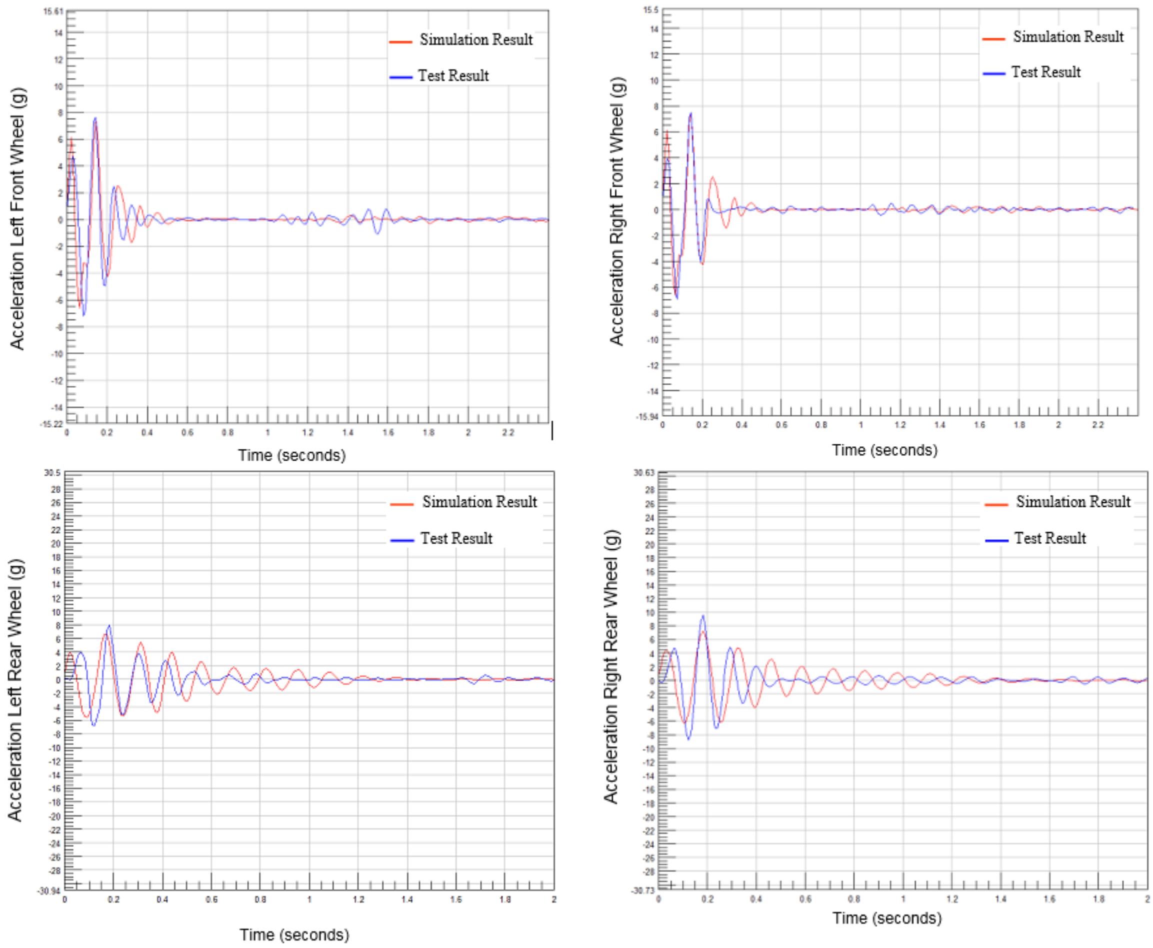

Fig. 12. Comparison of the Vertical Accelerations on Virtual and Actual Bus

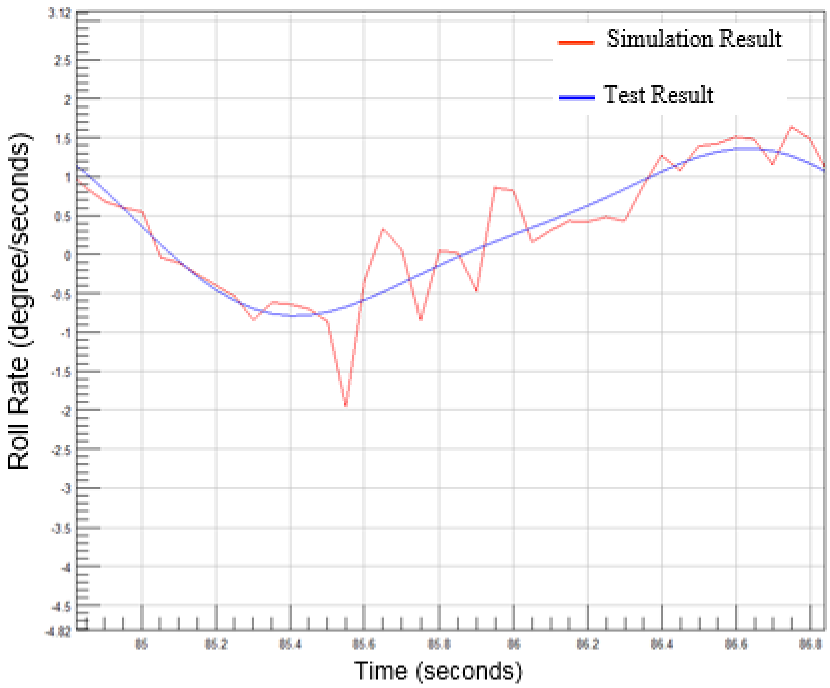

Fig. 13. Comparison of the Roll Rate on Virtual and Actual Bus

Lateral acceleration measured on the CoG were compared as shown in Figure 14.
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.
Multi body dynamic system analysis validated by the physical test is performed using Adams/Car software program using two different methods. One of these methods is based on static test simulation and the other is based on dynamic test simulation. Static test is chosen as tilt table test and dynamic test is selected as cornering and Fishhook test maneuvers.

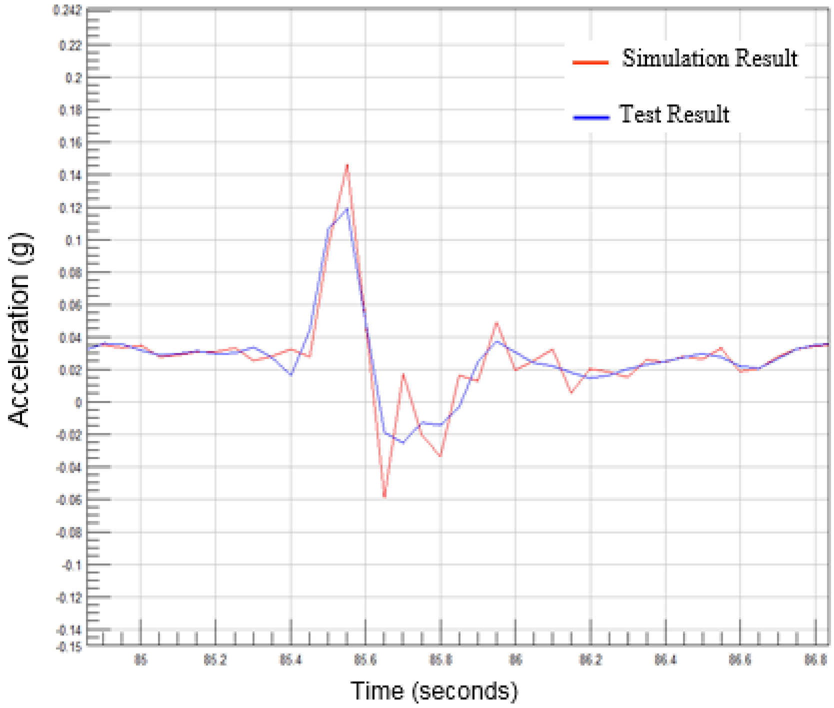

Fig. 14. Comparison of the Accelerations on Virtual and Actual Bus

4. Parametric sensitivity analysis based on static test

Sensitivity analysis approach is applied to efficiently tune the stiffness of the anti-roll bar and leaf spring at each axle in order to maximize rollover threshold. Studies are shown that suspension roll stiffness and roll center height are the most effective parameters for determining rollover behavior of the vehicle and roll angle threshold can be reduced by 8.3749 % if the rear suspension geometry is optimized [6, 8]. The Tilt Table test provides a measure of the level of lateral acceleration needed to lift the inside wheels off the ground and overturn a vehicle. In this analysis, one of the goals is to increase the rollover threshold value using static tilt table test. When tested using the existing parts, it was observed that right rear tire initially lifted off the table during the tilt table test as shown in Figure 15.
As it is seen that from figure 16, we can say that the roll stiffness of the rear suspension is higher than the front suspension because the point of roll instability of the rear suspension has not yet been reached. It was concluded, therefore, the front and rear roll stiffness should be balanced in order to have both wheels on the same side lift at the same time [13, 14].To achieve the intended design feature, we have to increase the roll stiffness of the front suspension in order to increase static stability factor because there is a linear relation between these two parameters.


Fig. 15. Adams/Car Tilt Table Test

As described in the previous sections,

suspension parameters including leaf spring and anti-roll bar actually affect the magnitude of the necessary lateral acceleration to produce rollover. Therefore, front leaf springs and front anti-roll bar were selected as design parameters. Three different diameters of the front anti-roll bar such as Ø38, Ø40 and Ø42 were selected as the first three set values of this parameter. If stiffness of the front anti-roll bar was more increased, this situation caused to increase in understeering. In a similar way, the front leaf spring is determined by the allowable bearing capacity of the front axle, ride height of the vehicle with the purpose of the ride comfort in mind as well. Therefore, the stiffness of the front leaf spring is limited to two different stiffness in the parametric sensitivity analysis such as 20 and 23 (kg/mm). Front leaf springs and front anti-roll bar were changed one by one in each experiment, so the combinations gave totally six different simulations. Moreover, validated Adams/Car virtual model was used in the parametric sensitivity analysis.

Front leaf spring and front anti-roll bar were selected like shown below as initial condition.

- The front leaf spring is selected as 20 (kg/mm)
- The diameter of the front anti-roll bar is selected as Ø38

Table 4. The Results of the Experiment 1

Front Leaf Spring	Front Anti-Roll Bar	Table Angle	Vehicle Roll Angle
20 (kg/mm)	Ø38	6°	6.60°
		18°	18.85°
		26.20°	29°
At the time of Rollover		**27.73°**	**33.57°**

The roll angle of the vehicle is obtained as 33.57° degree shown in Table 4 when the vehicle rolls over. Front leaf spring and front anti-roll bar used in this test is currently existing parts located on the vehicle. Therefore, the roll angle of the vehicle measured in subsequent tests will be compared with the results of this test.

- The front leaf spring is selected as 20 (kg/mm)
- The diameter of the front anti-roll bar is selected as Ø40

Table 5. The Results of the Experiment 2

Front Leaf Spring	Front Anti-Roll Bar	Table Angle	Vehicle Roll Angle
20 (kg/mm)	Ø38	6°	6.50°
		18°	19.01°
		26.20°	28.25°
At the time of Rollover		**28.11°**	**32.99°**

The roll angle of the vehicle is obtained as 32.99° degree shown in Table 5 when the vehicle rolls over. If it is compared with the results of the test-first test, table angle is increased while decreasing the vehicle roll angle by 1.72%.

- The front leaf spring is selected as 20 (kg/mm)
- The diameter of the front anti-roll bar is selected as Ø42

Table 6. The Results of the Experiment 3

Front Leaf Spring	Front Anti-Roll Bar	Table Angle	Vehicle Roll Angle
20 (kg/mm)	Ø42	6°	6.45°
		18°	18.91°
		26.20°	28°
At the time of Rollover		**28.54°**	**32.92°**

The roll angle of the vehicle is obtained as 32.92° degree shown in Table 6 when the vehicle rolls over. If it is compared with the results of the test-first test, table angle is increased while decreasing the vehicle roll angle by 1.93%.

- The front leaf spring is selected as 23 (kg/mm)
- The diameter of the front anti-roll bar is selected as Ø38

Table 7. The Results of the Experiment 4

Front Leaf Spring	Front Anti-Roll Bar	Table Angle	Vehicle Roll Angle
23 (kg/mm)	Ø38	6°	6.30°
		18°	19.07°
		26.20°	27.46°
At the time of Rollover		**28.17°**	**32.70°**

The roll angle of the vehicle is obtained as 32.70° degree shown in Table 7 when the vehicle rolls over. If it is compared with the results of the test-first test, table angle is increased while decreasing the vehicle roll angle by 2.59%.

- The front leaf spring is selected as 23 (kg/mm)
- The diameter of the front anti-roll bar is selected as Ø38

Table 8. The Results of the Experiment 5

Front Leaf Spring	Front Anti-Roll Bar	Table Angle	Vehicle Roll Angle
23 (kg/mm)	Ø40	6°	6.28°
		18°	18.90°
		26.20°	27.43°
At the time of Rollover		**28.56°**	**32.64°**

The roll angle of the vehicle is obtained as 32.64^0 degree shown in Table 8 when the vehicle rolls over. If it is compared with the results of the test-first test, table angle is increased while decreasing the vehicle roll angle by 2.77%.

- The front leaf spring is selected as 23 (kg/mm)
- The diameter of the front anti-roll bar is selected as Ø42

Table 9. The Results of the Experiment 6

Front Leaf Spring	Front Anti-Roll Bar	Table Angle	Vehicle Roll Angle
23 (kg/mm)	Ø42	6°	6.20°
		18°	18.87°
		26.20°	27.42°
At the time of Rollover		28.61°	32.60°

Table angle represents the rollover threshold. Therefore, increasing of the table angle means a certain amount of increase of the vehicle rollover threshold. In another sense, it means the vehicle will be rollover within later time. During this time, the roll angle of the vehicle decreases because, vehicle acts more stiff through tuned suspension roll stiffness. The roll angle of the vehicle is obtained as 32.60° degree shown in Table 9 when the vehicle rolls over. If it is compared with the results of the test-first test, table angle is increased while decreasing the vehicle roll angle by 2.88%.

The technical approach, from a suspension aspect, is to improve stability by increasing the roll stiffness of the suspension system in both static and dynamic response. Therefore, the vehicle rollover threshold can be increased maximum as 2.88 % from their nominal value as shown in Table 10 using static tilt table test.

Table 10. The Results of the Static Test

Front Leaf Spring	Front Leaf Spring	Reduction Rate of the Vehicle Roll Angle
20 (kg/mm)	Ø38	-
	Ø 40	1.72%
	Ø 42	1.93%
23 (kg/mm)	Ø 38	2.59%
	Ø 40	2.77%
	Ø 42	**2.88%**

Since static tests ignores the effects of tires and suspensions, a better model would include the possibility of the sprung mass because SSF ratings do not take the dynamic behavior of vehicles into account. If all parameters are grouped as a percentage of their own, efficiency percentages of the variable parameters are shown in percentage as seen in Figure 16.

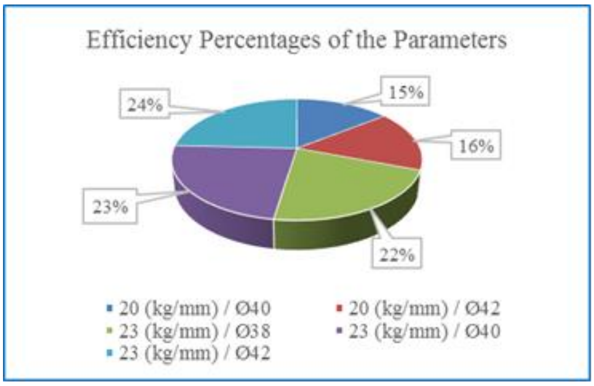

Fig. 16. Weight Ratio of the Parameters

It illustrates that the largest efficiency percentage is calculated as 24% If the stiffness of the front leaf spring is selected as 23 (kg/mm) and the diameter of the front anti-roll bar is selected as Ø42. Furthermore, it is observed that percentage distribution of each parameters is homogeneous. Therefore, the effect of parameter changes cannot be clearly seen.

5. Parametric sensitivity analysis based on dynamic test

The objective of this stage is to investigate the rollover threshold as function of anti-roll bar and leaf spring stiffness. Dynamic test series represents a different set of driving maneuvers and provides information based on real-world operations in which rollover is inevitable. Some of these maneuvers such as the Fishhook and cornering used in this work are described here. The Fishhook test series represent extreme cornering at relatively high speeds without braking. Accidents involving similar driving conditions to those demonstrated in Fishhook [15, 16, and 17]. In this maneuver, the vehicle is driven at a speed of 40 km/h in a straight line then the steering wheel is ramped at an angle of 550° in 3 seconds.

The steering wheel is held at this angle for 4 seconds then turned back to zero degrees at a steady rate during the following 2 seconds. Roll angle measured from the CoG is decreased in case of increasing the roll stiffness of the front suspension as shown in Figure 17.

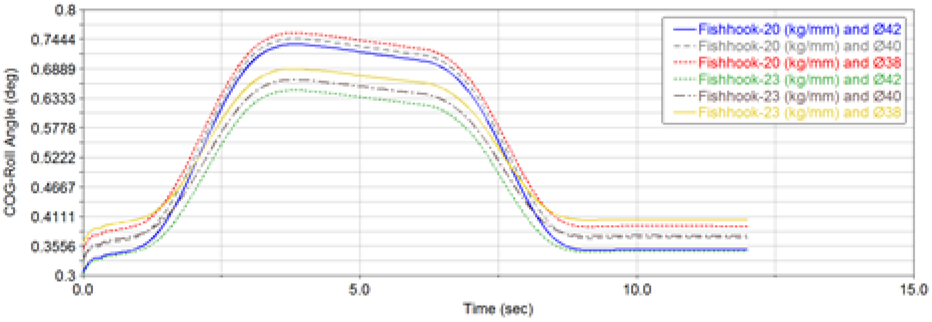

Fig. 17. The Roll Angles When Fishhook Maneuver

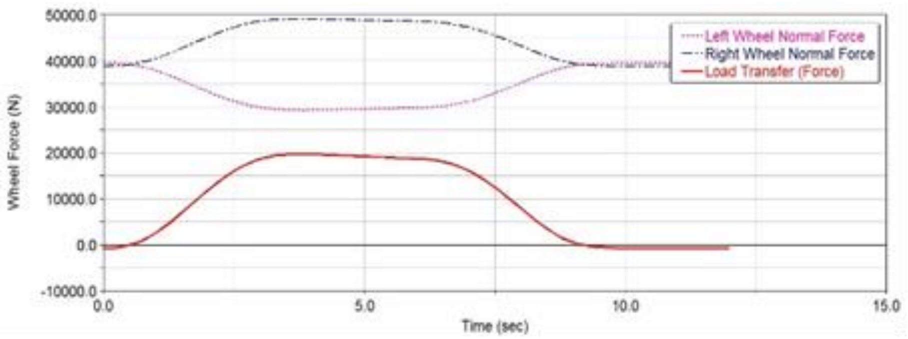

Fig. 18. Normal Forces of the Right and Left Tires

Figure 17 illustrates that rollover threshold can be increased by increasing suspension roll stiffness. Normal forces of the right and left tires were shared as shown Figure 18.

Purple line represents normal force of the sum of the left front and rear wheels. Blue line represents normal force of the sum of the right front and rear wheels. Red line represents the difference between left and right wheel forces.

In another sense, force transfer of the left wheel to the right wheel means the lateral load transfer ratio. It was measured as 20kN. This type of force can rotate the vehicle around its roll-axis by 0.75° degrees. Fishhook maneuver analysis results are shared in Table 12 for detailed examination.

Table 12. The Results of the Fishhook Maneuver Analysis

Fishhook Maneuver Analysis			
Front Leaf Spring	Front Anti-roll bar	Roll Angle (degree)	Percentage Change (%)
20 (kg/mm)	Ø38	0.75°	-
	Ø40	0.74°	1.55%
	Ø42	0.73°	3.26%
23 (kg/mm)	Ø38	0.68°	10.14%
	Ø40	0.66°	13.27%
	Ø42	0.64°	**16.71%**

It can be seen from table 12 that the vehicle roll angle threshold can be reduced by 16.718

% at maximum from their nominal value by increasing the roll stiffness of the suspension system. If Fishhook maneuver test results were to be compared with the cornering test results, it is seen that ratio of the percentage change in roll angle is equivalent to cornering test results. A persistent observation is that changing only the diameter of the front anti-roll bar is not sufficient to reduce rollover threshold. Efficiency percentages of the variable parameters described in the static test results are shown in percentage as seen in Figure 19.

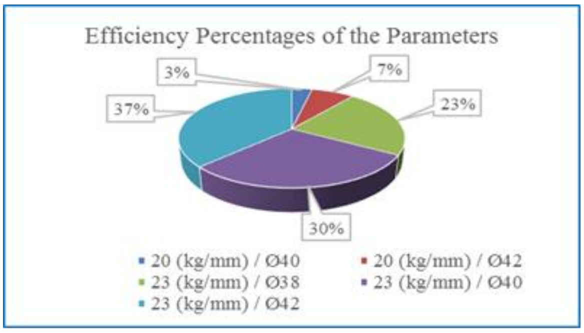

Fig. 19. Weight Ratio of the Parameters

Fig 19 illustrates that the largest Efficiency percentage is 37% .if the stiffness of the front leaf spring is selected as 23 (kg/mm) and the diameter of the front anti-roll bar is selected as Ø42, Furthermore, it is observed that percentage distribution of each parameters is heterogeneous. In case of the diameter of the front anti-roll bar is changed and stiffness of the leaf spring is kept fixed, rollover weight ratio is calculated as 7 %. Although, in case of the both leaf spring and anti-roll bar are changing, rollover weight ratio is calculated as 37 %.

6. Statistical parameter test

The most common way to determine whether there are differences in the means of a continuous DV across a set of three or more groups is to perform an analysis of variance (ANOVA). The Anova technique applies when there are two or more than two independent groups. The ANOVA procedure is used to compare the means of the comparison groups.

For the established model, the hypotheses are determined as follows,

- H_0: There is no significant difference between group averages
- H_1: There is a significant difference between at least two group averages

As a result of the one-way ANOVA analysis between each independent variable, the F value for each variable is calculated. The hypothesis (H_0) is determined by comparing the calculated F value with the table value.

If $F_{Calculated} < F_{table}$, (p > significant), H_0 Accepted

If $F_{calculated} > F_{table}$, (p < significant), H_0 Rejected

If the p-value is less than the significance level you conclude that the mean of the population is significantly different from the comparison value. If the p-value is greater than the significance level you conclude that the mean of the population is not significantly different from the comparison value.

While the t-test is limited to comparing means of two groups, one-way ANOVA can compare more than two groups. Therefore, the t-test is considered a special case of one-way ANOVA.

As a result of the t-test analysis between each independent variable, the t value for each variable is calculated. The hypothesis (H_0) is determined by comparing the calculated t value with the table value.

If $t_{Calculated} < t_{table}$, (p > significant), H_0 Accepted

If $t_{calculated} > t_{table}$, (p < significant), H_0 Rejected

If the p-value is less than the significance level you conclude that the mean of the population is significantly different from the comparison value. If the p-value is greater than the significance level you conclude that the mean of the population is not significantly different from the comparison value.

Tests of the equality of variances are sometimes used on their own to compare variability across groups of experimental or non-experimental conditions but they are most often used alongside other methods to support assumptions made about variances Therefore, It is essential to examine the equality of variances before conducting a t-

test and Anova. Levene Test are used to examine equality of variances.
Hypothesis are determined as follows,

- H_0: the variances are equal
- H_1: the variances are not equal

If (p > significant), H_0 Accepted
If (p < significant), H_0 Rejected

6.1. t-Test of the Front Leaf Spring

Levene Test, have been explained above, so we will only take the results ($\alpha_{sig} = 0.398 > 0.05 = \alpha$). So we have no reason to reject equality of variances-hypothesis at the significance level 5%. We can assume that the variances are equal for Dynamic analysis result. Therefore, only the differences of the leaf spring stiffness can be obtained according to dynamic analysis.
According to this finding, it was observed a significant difference in dynamic test result ($t=3.43$, $p<.05$). Based on the results of t-test analysis, differences of the leaf spring stiffness affect dynamic test results.

Table 13. The Results of the Front Leaf Spring Variances Test

	Levene Statistic	Sig.
Static analysis result	10.743	0.031
Dynamic analysis result	0.895	0.398

Table 14. The Results of the Front Leaf Spring t-test

	t Statistic	Sig.
Static analysis result	6.146	0.068
Dynamic analysis result	30.881	0.005

6.2. Anova Test of the Front Anti-Roll Bar

Levene Test, have been explained above, so we will only take the results ($\alpha_{sig} = 0.62 > 0.05 = \alpha$). We can assume that the variances are equal for Dynamic analysis result. Therefore, only differences of the Front Anti - Roll Bar stiffness can be obtained according to dynamic analysis.
According to this finding, it was observed a significant difference in dynamic test result ($F=17, 2$ $p<.05$). Based on the results of F-test analysis, differences of the Anti-Roll Bar stiffness affect dynamic test results.

Table 15. The Results of the Front Anti-Roll Bar Variances Test

	Levene Statistic	Sig.
Static analysis result	39	0.001
Dynamic analysis result	12.85	0.62

Table 16. The Results of the Front Anti-Roll Bar Anova test

	F Statistic	Sig.
Static analysis result	0.496	0,652
Dynamic analysis result	17.2	0,005

7. Results of the parametric sensitivity analysis

Static test results were compared with the dynamic test results as shown in Table 16. If one wants to improve vehicle stability significantly, it may not be effective enough to only change the anti-roll bar stiffness. Other factors may need to be accounted for, such as the leaf spring stiffness. Static test result is not sufficient in order to observe the effect of the parameters because threshold of the vehicle rollover can be reduced by 2.88 % at maximum from their nominal value.

Table 16. The Results of the Static and Dynamic Sensitivity Analyses

Results of the Static and Dynamic Sensitivity Analyses			
Front Leaf Spring	Front Anti-Roll Bar	Percentage Change of the Rollover Threshold (%) Static Test	Percentage Change of the Rollover Threshold (%) Dynamic Test
20 (kg/mm)	Ø38	-	-
	Ø40	1.72	1.55%
	Ø42	1.93	3.26%
23 (kg/mm)	Ø38	2.59	10.14%
	Ø40	2.77	13.27%
	Ø42	**2.88**	**16.71%**

8. Conclusion

This paper discusses the benefits of midi bus simulation for stability tuning at product development and demonstrates that it can greatly inform the designer. Virtual simulation can guide the designers to reach optimum suspension parameters for the vehicle.

In order to decrease the propensity of the rollover for the vehicle, soft suspension (front suspension) was optimized using vehicle parameters in order to reduce the risk of rollover. Furthermore, the effect of improved parameters were investigated using both static and dynamic tests. The analysis shows that it is possible to maximize rollover threshold and increase vehicle stability by tuning suspension parameters so much so that rollover threshold can be increased by 16.71 % at maximum from the nominal value using three different diameters of the front anti-roll bar and two different stiffness of the front leaf spring.

Front leaf spring stiffness used 23 (kg/mm) decrease the understeer gradient as shown in Figure 17. It will be outside of predetermined boundary conditions, if used 23 (kg/mm). On the other hand, the understeer gradient varies very little if front anti roll bar stiffness was increased. The use of the optimum spring stiffness found in the analysis is not appropriate since it is not a standard value. Therefore, rollover threshold can be only increased as 3.26 %. In addition to the results, it was observed a significant difference in dynamic test result ($t=3.43$, $p<.05$). Based on the results of t-test analysis, differences of the leaf spring stiffness affect dynamic test results. Moreover, it was observed a significant difference in dynamic test result ($F=17, 2$ $p<.05$). Based on the results of F-test analysis, differences of the Anti-Roll Bar stiffness affect dynamic test results.

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