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RIGID REAR AXLE DEVELOPMENT FOR A COMMERCIAL VEHICLE

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ABSTRACT

In this study, a rigid axle design for a city/intercity bus is developed. The design has focused on high comfort, high load capacity, availability to easy access and low cost. The design steps, which are determining concept, theoretical design, product development and final product, are summarized in this article. The rear suspension which is designed has been validated with bench tests and vehicle durability tests.

Keywords: Vehicle Dynamics, Product Development, Validation, Chassis, Suspension, Rigid Axle

BİR TİCARİ ARAÇ İÇİN RİJİT AKS GELİŞTİRİLMESİ

ÖZET

Bu çalışmada, şehiriçi/şehirdışı otobüs için arka aks geliştirilmiştir. Tasarım; yüksek konfor, yüksek taşıma kapasitesi, kolay erişilebilirlik ve düşük maliyet isterlerine odaklanmıştır. Bu makalede; konsept belirleme, teorik tasarım, ürün geliştirme ve nihai ürün gibi tasarım aşamaları özetlenmiştir. Tasarlanan arka süspansiyon fiziksel testlerle ve araç dayanım testleriyle doğrulanmıştır.

Anahtar kelimeler: Taşıt Dinamiği, Ürün Geliştirme, Doğrulama, Şasi, Süspansiyon, Rijit Aks

1. INTRODUCTION

Suspension; always hidden from view but nevertheless crucial for safety, handling and comfort. Commonly, most of commercial vehicles are equipped with off-the-shelf rear suspension systems of some main suppliers. An OEM brand in Turkey demanded a rear suspension development for 8-meters long, rear wheel drive, low-floor city/intercity bus with has rear axle load capacity of 6 tons. Within this scope of project, most known concepts of rear suspension is evaluated to meet the customer requirements with efficient and feasible:

- Rigid Axle and 4-Air Springs
- Rigid Axle and Leaf Springs
- Rigid Axle with 2-Air Springs, Longitudinal Links and V-Link
- Rigid Axle with 2-Air Springs, Longitudinal Links and Lateral Link

As a result of evaluation of **Table-1**, the concept of Rigid Axle with 2-Air Springs, Longitudinal Links and Lateral Link have been chosen as a feasible solution. The comparison has done according to requirements shown in **Table-1**. While creating a comparison chart between these four different suspension system solutions; packaging, unit weight, tuning period, performance, vehicle dynamics, investment cost, engineering-development-validation tests cost, development time and unit cost are the requirements which taken into consideration. Chosen suspension system solution is better solution in terms of unit weight, development time and unit cost. Especially with regards to packaging, it has many advantages to make a low floor bus has rear drive axle.

Table 1. Comparison of Rigid Axle Concepts

		Rigid Axle and 4-Air Springs	Rigid Axle and Leaf Springs	Rigid Axle with 2-Air Springs, Longitudinal Links and V-Link	Rigid Axle 2-Air Springs, Longitudinal Links and Lateral Link
Technical Performance	Packaging	+1	-1	0	+2
	Unit Weight	-2	+2	+1	+2
	Tuning Period	-1	-1	-1	0
	Performance	-2	-1	0	0
	Vehicle Dynamics	-1	-1	+1	0
Cost	Investment	0	-1	+1	-1
	Engineering, Development & Validation Testing	-2	+1	-1	0
	Development Time	-1	+2	-1	+1
	Unit Cost	-1	+2	0	+1
	TOTAL SCORE	-9	+3	0	+5

2. THEORETICAL DESIGN

By defining the concept, the main components of system are determined. The parametric model is settled in MSC ADAMS and basic characteristics of the suspension system are defined by benchmarking, literature and requirements.

During the theoretical design studies, it was evaluated to use anti-roll bar (yellow part in **Figure-5**) to overcome cornering controllability and stability, and increase roll stiffness of suspension. Firstly, anti-roll bar working principles are evaluated. When one wheel executes a motion that is opposite to that of the other wheel, however, the anti-roll bar is twisted. In this case, the anti-roll acts as a linear torsional spring, providing a torque that is proportional to its twist angle. This torque serves to counteract vehicle roll motion, reducing the roll angle of the vehicle's body. The use of an anti-roll bar not only reduces the vehicle's roll angle, but can also provide a further notable contribution to driving dynamics. When the body of a vehicle rolls, the wheels on either side are displaced vertically in opposite directions. This causes the anti-roll bar to twist, which results in a restoring moment about the roll axis, thereby reducing body roll. If both main springs on a particular axle are compressed or extended simultaneously. Unlike the main suspension springs that are loaded by static forces when the vehicle is at rest, anti-rolls are only loaded when the vehicle is in motion. When a force is applied to just one wheel by a road surface irregularity, the stiffness of the suspension on the other side of the vehicle is increased.

In order to reveal the effect of anti-roll bar, some analysis are performed. Roll gradient, suspension roll stiffness and total roll stiffness are calculated (shown in **Figure-2-3-4**) in order to see the differences.

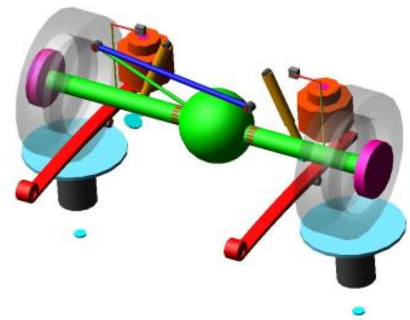

Figure-1 ADAMS Model of Rigid Axle

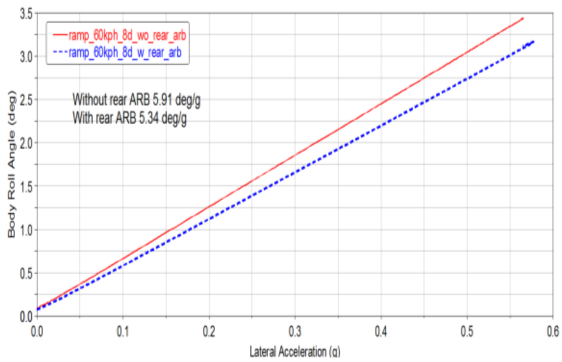

Figure-2 Roll Gradient Comparison in ADAMS (red: w/ ARB, blue: w/o ARB)

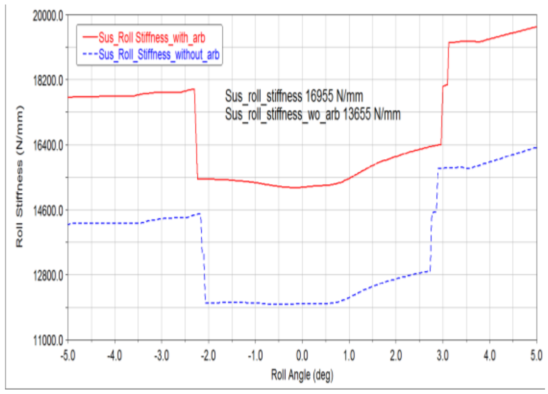

Figure-3 Suspension Roll Stiffness Comparison in ADAMS (red: w/ ARB, blue: w/o ARB)

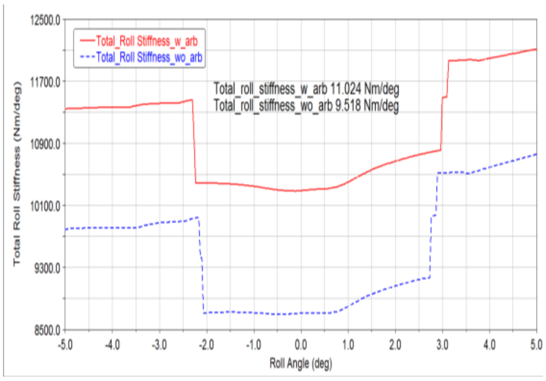

Figure – 4 Total Roll Stiffness Comparison in ADAMS (red: w/ ARB, blue: w/o ARB)

It is aimed that M3 class vehicles shall have reach averagely 0,55g lateral acceleration and sufficient handling performance during the cornering maneuver. The designed rear anti-roll bar provides this target with 0.58 g lateral acceleration. According to these calculations roll gradient, suspension roll stiffness and total roll stiffness are improved and target values are achieved.

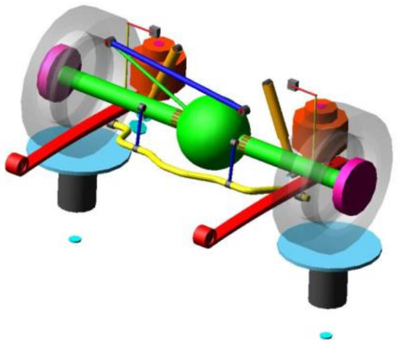

Figure-5 ADAMS Model of Rigid Axle w/ARB

In addition to anti-roll bar, some links are added to system such as panhard rod as lateral link (blue part in **Figure 5**) and air-linker as longitudinal links (red parts in **Figure 5**). Firstly, panhard rod are examined for this suspension system. Lateral forces are transmitted between the axle and the vehicle's body by a panhard rod or one of the other types of linkages. The motion of a panhard rod causes the vehicle's body to shift laterally during compression and rebound. Lateral control can also be provided by a Panhard rod. In order to prevent any steering motion during suspension compression, it is important to specify a linearly-acting vertical mechanism such as a panhard rod. Even if such a linkage is used, a slight steering motion still occurs during single-wheel compression, similar to a rigid axle with longitudinal links.

The height of the rod helps to determine the height of the rear roll center. The roll center is an imaginary point around which on rear axle of the car. The height of the rear roll center (and the front also) is critical for handling. When you lower the panhard rod the rear roll center drops. However, an extremely low roll center can generate excessive chassis roll which can cause suspension geometry problems. Also, excessive roll can delay corner exit acceleration. Raising the panhard rod causes to rise in the rear roll center. Generally, this adjustment causes corner entry handling to loosen and chassis roll to lessen. When adjusting for height, attachment points of panhard rod have to be changed from both ends.

During cornering the chassis exerts a side force on the rear axle and tires through the panhard rod. When the panhard rod is level, it transmits a whole lateral force to the rear tires. However, when the panhard rod is angled downward to the right, it transmits a partially downward force to the rear tires and rear traction is enhanced. Conversely, when the panhard rod is angled upward to the right, it transmits a partially upward force to the rear tires and rear traction is lessened. The effect of an angled panhard rod on rear tire loadings is brief but very important on handling. If the panhard rod is attached to the rear axle near the center of the rear trackwidth axis, the panhard rod will load or unload both rear tires by a similar amount during cornering. During cornering maneuver, wheel center deflection is decreasing with the usage of panhard rod.

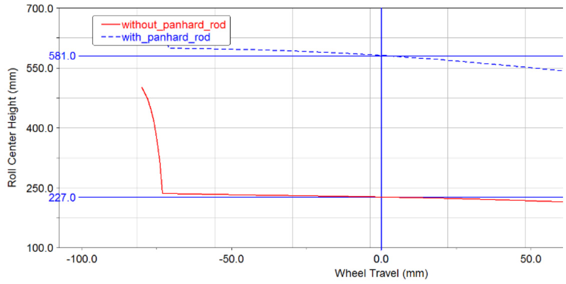

Figure-6 Diagram w/wo panhard rod_roll center height

Roll center height is gone up with the usage of panhard rod in order to increase handling performance during cornering.

3. PRODUCT DEVELOPMENT

According to theoretical design results, bill of material for related vehicle existed. Design verification plan is prepared and CAD modeling is studied in CATIA which is compatible with the ADAMS hardpoints. When the product design has reached the level of functionality and minimum strength targets, attribute prototypes (AP-level) are produced and equipped on prototype test vehicle.

Required vehicle dynamics tests are performed with AP-level prototype vehicle and results are correlated with the theoretical design to evaluate and study for vehicle dynamics tuning. According to tuning study results, CAD modeling studies are repeated by aiming required life cycle, strength and optimizing the vehicle dynamics both and CAE studies are done. As a result of these studies, confirmation prototype (CP-level) products are existed and manufactured (Figure 6 and 7).

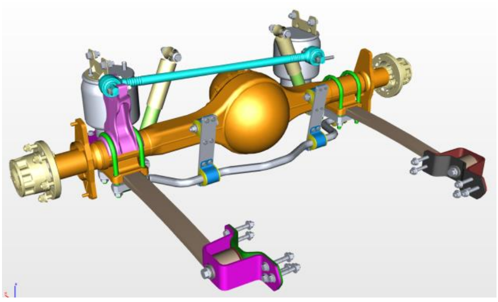

Figure-7 CP-level CAD

By supplying of CP-level products, fatigue and strength bench tests and proving ground vehicle durability tests are performed. Also objective vehicle dynamics tests and drive team event are performed and correlated with the theoretical studies. Subsystem and components are validated fully.

3.1. Rigid Axle, Component Bench Test

The rigid axle is co-designed by a supplier and sub-assembly bench tests are performed in its facility. Targets of tests are fully confirmed and validated as shown in **Figure 8, 9, 10** and according to Hexagon Studio Test Procedures:

Targets:

Static Vertical Strength:

- 6550kg load
- F > 2.2 x 6550kg
- No crack or loss of function until target value
- Permissible deflection = 1.7mm & 0.1 deg

Dynamic Vertical Loading:

- Fmin = 655kg, Fmax=6550kg
- Target Lifecycle = 250.000 cycles, no crack or loss of function
- Loading Frequency = 2-10 Hz

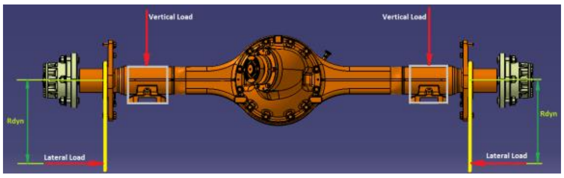

Figure – 8 Test Setup Plan

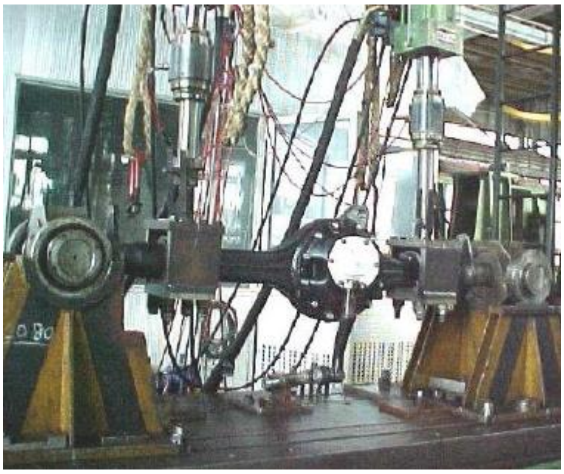
Figure – 9 Test Setup

Dynamic Lateral Loading:

- Fmin= 0kg Fmax=(rated max. lateral load) kg
- Target Lifecycle = 150.000 cycles, no crack or loss of function
- Loading Frequency = 0.5 – 1 Hz

Wheel End Test:

- Mmin= 380Nm Mmax =1362,5 Nm
- Target Lifecycle = 500.000 cycles, no crack or loss of function
- Loading Frequency = 0.5 – 2 Hz

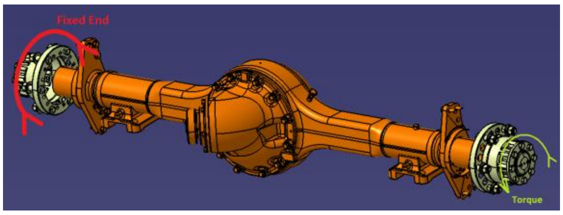

Figure – 10 Torsion Test Setup Plan

3.2 Air-linker, Component Bench Test

The air-linker is co-designed by a local supplier and component bench tests are performed in its facility. Target of tests are fully confirmed and validated as shown in **Figure 11** and according to Hexagon Studio Test Procedures:

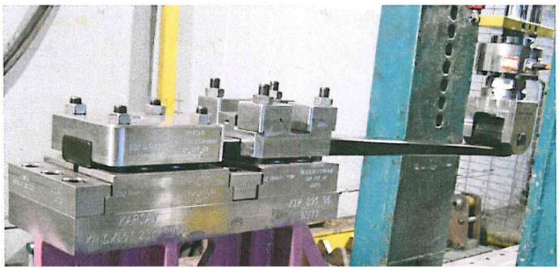
Figure – 11 Airlinker Test Setup

Targets:

- Sample Temperature = Max. 90^{o}C
- Fmax= 20132N
- Loading Frequency = 0.75Hz
- Target Lifecycle = 200.000 cycles, no breakage, crack, friction, plastic deformation

3.2.1. Air-Linker Pivot Bushing Development

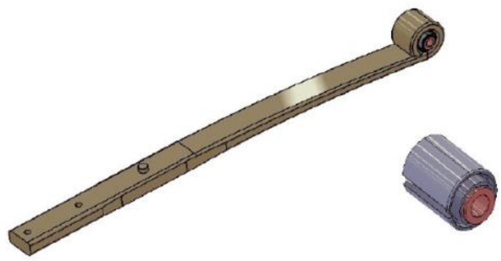
Figure – 12 Pivot Bushing

According to NVH CAE studies, air-linker pivot bracket area had insufficient stiffness by considering targets. Although the air-linker bushing stiffness was 35kN/mm and met the vehicle dynamics targets, this critical issue enforced to develop a new air-linker bushing and regarding the vehicle dynamics requirements and NVH targets the radial stiffness value has determined as below. As it is shown on the target progressive curve the linear stiffness was ≈2.8kN(mm until to 2mm displacement and ≈6kN/mm after that displacement.

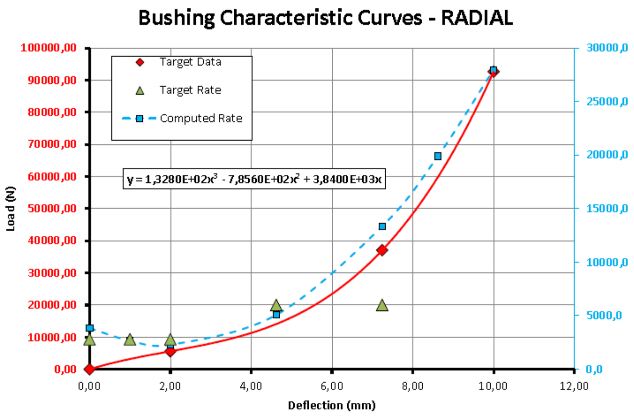

Figure – 13 Theoretical Curve of Pivot Bushing

In addition to VeD and NVH requirements there were some boundary conditions like outside diameter and rubber thickness value. The thickness of rubber must be at least 3 times more than the displacement value at maximum load on bushing. The tolerance of outside diameter was limited related to the push out load which defined as min. 1500kg before.

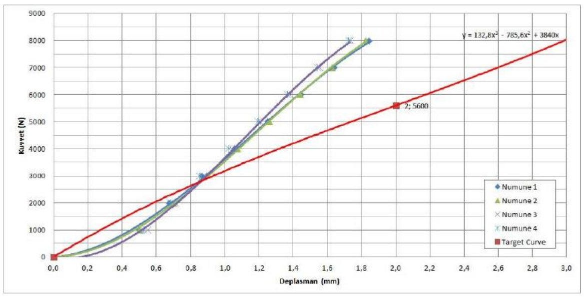

Figure – 14 Development result of actual pivot bushing

3.3 Air Spring, Component Bench Test

The air-spring is co-designed by a local supplier and component bench tests are performed in Hexagon Studio. Target of tests are fully confirmed and validated as shown in **Figure 15** and according to Hexagon Studio Test Procedures:

Targets:

- Sample Temperature = Max. 90°C
- Fmin= 10kN Fmax= 50kN
- Loading Frequency = 0.5Hz
- Target Lifecycle = 200.000 cycles, no breakage, crack, friction, laceration

Figure – 15 Air Spring Test Setup

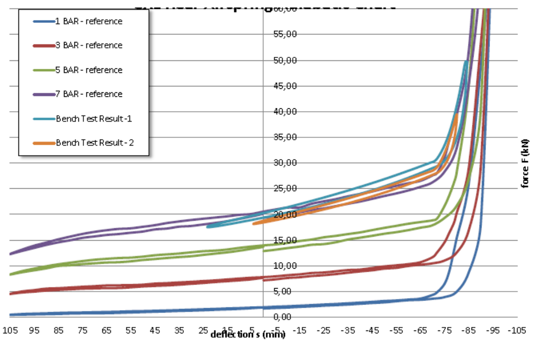

Figure – 16 Air Spring Test Result Correlation Diagram

3.4 Shock Absorber, Component Bench Test

The shock absorber is co-designed by a local supplier and component bench tests are performed in its facility. Target of tests are fully confirmed and validated according to Hexagon Studio Test Procedures. Performed test types are listed below:

- Full Stroke Operation
- High Frequency Operation
- Double Sinus Operation
- Muddy Water Test
- Noise Evaluation
- Breaking Strength Dynamic
- Oil Seal Durability

3.5 Bump stop, Component Bench Test

Figure – 17 Bump Stop Test Sample

The bump stop is co-designed by a local supplier and component bench tests are performed in Hexagon Studio. Target of tests are fully confirmed and validated as shown in **Figure 15** and according to Hexagon Studio Test Procedures:

3.6 Anti-roll Bar, Component Bench Test

The anti-roll bar is designed by Hexagon Studio and component bench tests are performed in its facility. Target of tests are fully confirmed and validated according to Hexagon Studio Test Procedures.

Targets:

- Test Displacement = +/- 22mm
- Loading Frequency = 1Hz
- Target Lifecycle = 250.000 cycles, no breakage, crack, friction, plastic deformation

3.7 Panhard Rod, Component Bench Test

The panhard rod is co-designed by a local supplier and component bench tests are performed in its facility. Target of tests are fully confirmed and validated according to Hexagon Studio Test Procedures.
Targets:

Survive Test

- Tensile Load = 90kN
- Compression Load = 90kN

Fatigue Test

- Fmin=0 Fmax=52kN
- Loading Frequency = 1 Hz
- Target Lifecycle = 150.000 cycles, no breakage, crack, friction, plastic deformation

3.8 Rear Suspension, Sub-system Bench Test

The test procedure and bench system is developed by Hexagon Studio and performed successfully. Complete rear suspension is tested and validated. The bench system is designed to simulate the in-use conditions and correlated with CAE studies and proving ground durability tests.

Figure – 18 Lateral Survive Test Setup

Targets:

Vertical Fatigue Test:

- Fmin= VeD Results, Fmax= VeD Results
- Loading Frequency = 0.5 Hz
- Target Lifecycle = 250.000 cycles, no loss of function, plastic deformation

Lateral Fatigue Test:

- Fmax= VeD Results
- Loading Frequency = 0.5 Hz
- Target Lifecycle = 150.000 cycles, no loss of function, plastic deformation

Vertical Survive Test:

- Fmax= 48000kg, Fsurvive= (depends on customer expectation) kg
- Target Lifecycle, plastic deformation is permissible but no loss of function or crack

Lateral Survive Test:

- Fmax= 12000kg, Fsurvive= (depends on customer expectation) kg
- Target Lifecycle, plastic deformation is permissible but no loss of function or crack

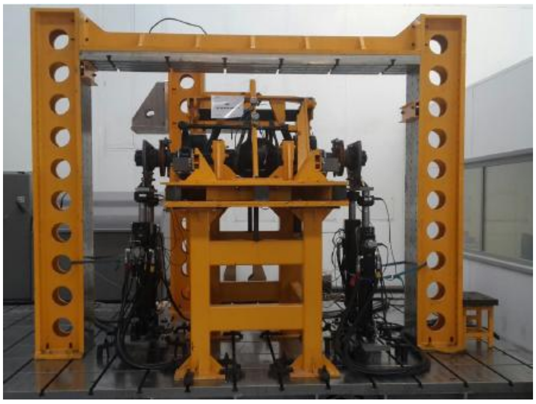

Figure – 19 Fatigue Test Setup

4. FINAL PRODUCT

The axle is validated fully, confirmed by customer and went on serial production.

Figure – 20 Final Product

5. CONCLUSION

In this study; a rigid axle that has 6tons load capacity is developed according to customer requests. All development steps; - concept design, product design and ergonomic, engineering and development, virtual product, prototype, test and validation, 3D and 2D release, production line and after sale support – has performed. The designed axle has advanced ride and passenger comfort level, availability to ride height level adjustment, kneeling, high handling control, low floor to provide easy access for disabled; the anti-roll bar is providing a better cornering maneuver performance than benchmark vehicles.

REFERENCES

1. Hessing, B., Ersoy, M. (2011), "**Chassis Handbook**", Vieweg+Teubner Verlag.
2. Gillespie, T. (1989), "**Fundamentals of Vehicle Dynamics**", Society of Automotive Engineers.
3. Reimpell J., Stoll H., Betzler J. W. (2001). "**The Automotive Chassis**", Reed Elsevier and Professional Publishing Ltd.