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FAKULTA STROJNÍHO INŽENÝRSTVÍ

INSTITUTE OF MACHINE AND INDUSTRIAL DESIGN

ÚSTAV KONSTRUOVÁNÍ

TECHNICAL DIAGNOSTICS OF THE MOTORCYCLE SUSPENSION

TECHNICKÁ DIAGNOSTIKA ZÁVĚSU MOTOCYKLU

MASTER'S THESIS

DIPLOMOVÁ PRÁCE

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VEDOUCÍ PRÁCE

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Assignment Master's Thesis

Institut:	Institute of Machine and Industrial Design
Student:	**Bc. Ali Mohammed Ali Hasan Al-Qubati**
Degree programm:	Mechanical Engineering Design
Branch:	no specialisation
Supervisor:	**doc. Ing. Milan Klapka, Ph.D.**
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As provided for by the Act No. 111/98 Coll. on higher education institutions and the BUT Study and Examination Regulations, the director of the Institute hereby assigns the following topic of Master's Thesis:

Technical diagnostics of the motorcycle suspension

Brief Description:

The resonance – adhesion principle test was developed for passenger cars as fast and reliable solution to assess the overal technical condition of the suspension. However, chassis geometry of the motorbike is different from car, therefore it is required to verify applicability of such method on bikes. Identification of most critical operational parameters which has major impact on the test results should verify, whether such diagnostic method could be transferred from passenger cars to motorcycles.

Type of the thesis: Research
Output of the thesis: Publication (J, D)
Project: TACR MESTEC

Master's Thesis goals:

The main goal is to verify whether of the diagnostic method for testing wheel suspension based on the resonance – adhesion principle (EUSAMA) is apliccable on motorcycles.
Sub–aims of the master thesis:
– design of a simulation model for the resonance–adhesion test of the motorcycle suspension,
– identification of input parameters of the simulation model,
– sensitivity study and analysis of critical operational parameters by experiment and simulation,
– asessment of impact of the selected parameters on precision of the test results.

Required outputs: report, electronic data.
Thesis range: approx. 72,000 characters (40 – 50 pages excluding figures).
Time schedule, thesis structure and the template of the report must follow:
http://ustavkonstruovani.cz/texty/magisterske–studium–ukonceni/

Recommended bibliography:

COSSALTER, Vittore. Motorcycle dynamics. 2nd English ed. [S.l.: Lulu], 2006. ISBN 978-143-0308-614.

FOALE, Tony. Motorcycle Handling and Chassis Design: the art and science. Spain: tonyfoale.com, 2002. ISBN 978-8493328610.

DIXON, John C. The shock absorber handbook. 2nd ed. Chichester, England: John Wiley, c2007. ISBN 978-0-470-51020-9.

Deadline for submission Master's Thesis is given by the Schedule of the Academic year 2021/22

In Brno,

L. S.

prof. Ing. Martin Hartl, Ph.D. Director of the Institute	doc. Ing. Jiří Hlinka, Ph.D. FME dean

ABSTRACT

Vehicle suspension testing is very important to ensure a safe and comfortable ride. There are many testers, one of which is the resonant adhesion tester. It is an easy and non-invasive device that is used to assess the functionality of cars' suspensions. EUSAMA is one of the best known methodologies that this tester is based on. However, there is no such simple device for checking the technical condition of a motorcycle suspension. Therefore, this work examined the applicability of the EUSAMA methodology for the evaluation of motorcycle suspension. It also examined the specification and determination of the parameters that most affect the EUSAMA, as well as ways to minimize the negative impacts of these parameters on the accuracy and reliability of test results. In parallel, a simulation model was created for the Aprilia ETX motorcycle for sensitivity analysis, in which the range for the most important parameters of the motorcycle suspension was set, and their effect on EUSAMA was tested within their specified range. As a result of the simulation model and experiments, it was observed that the rider's weight, tire stiffness, and platform angle have a major impact on EUSAMA, where rider weight, stability during testing, tire inflation, and platform angle could lead to incorrect suspension evaluation. Subsequently, the modified platform and the stabilizing mechanism were experimentally verified on a selected sample of motorcycles. To eliminate the negative impact of the monitored parameters, several modifications were proposed in this work, including some test conditions and design accessories, including the addition of a stabilizing mechanism, checking tire inflation before testing, and specifying the driver's position during testing.

ABSTRAKT

Testování odpružení vozidel je velmi důležité pro zajištění bezpečné a pohodlné jízdy. Existuje mnoho testerů, přičemž jedním z nich je rezonanční adhezní tester. Jedná se o snadný a neinvazivní přístroj, který se používá k posouzení funkčnosti odpružení auta. EUSAMA je jednou z nejznámějších metodik založená na využití rezonančně adhezního testu. Avšak pro kontrolu technického stavu odpružení motocyklu neexistuje žádný takový jednoduchý nástroj. Tato práce proto zkoumala použitelnost metodiky EUSAMA pro zhodnocení odpružení motocykl. Dále zkoumala upřesnění a stanovení parametrů, které ji nejvíce ovlivňují a zároveň způsoby minimalizace negativních dopadů těchto parametrů na přesnost a spolehlivost výsledků testu. Paralelně byl vytvořen simulační model pro motocykl Aprilia ETX pro analýzu citlivosti, ve kterém byl nastaven rozsah pro nejdůležitější parametry odpružení motocyklu a byl testován jejich vliv na EUSAMA v jejich zadaném rozsahu. Výsledkem simulačního modelu a experimentů bylo pozorováno, že hmotnost jezdce, tuhost pneumatiky a úhel platformu mají velký vliv na EUSAMA, kdy hmotnost jezdce, jeho stabilita během testování, nahuštění pneumatik a úhel plošiny by mohly vést k nesprávnému hodnocení odpružení. Následně byl experimentálně verifikován upravený tester na vybraném vzorku motocyklů. Pro eliminaci negativního vlivu sledovaných parametrů bylo v této práci navrženo několik úprav včetně některých zkušebních podmínek a konstrukčních doplňků, mezi které spadá přidání stabilizačního mechanismu, kontrolu nahuštění pneumatik před testováním a upřesnění polohy jezdce při testování.

// KEYWORDS

EUSAMA, Resonance adhesion tester, Motorcycle suspension, Sensitivity analysis

KLÍČOVÁ SLOVA

EUSAMA, Rezonanční adhezní tester, Odpružení motocyklu, Analýza citlivosti

BIBLIOGRAPHIC REFERENCE

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STATEMENT ON THE ORIGINALITY OF THE WORK

I declare that I have written this thesis on the topic of *Technical diagnostics of the motorcycle suspension* on my own with the help of my supervisor *doc. Ing. Milan Klapka, Ph.D.* and using the references listed in the bibliography section.

Ali Mohammed Ali Hasan Al-Qubati

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ACKNOWLEDGMENT

Firstly I would like to thank my supervisor, doc. Ing. Milan Klapka, Ph.D. who has been there for guidance all the time. I would like to thank him for his time, patience, and advice that have helped me throughout my work on this thesis. I would also like to give my gratitude to doc. Ing. Ivan Mazůrek, CSc. who has provided great help for us during the experiments of this thesis.

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CONTENT

1 INTRODUCTION

The number of vehicles around us is increasing at a very fast pace, and alongside, the importance of routine testing for a safe and comfortable drive also increases. One of the elements needed to be tested regularly is the suspension system which is used almost in all vehicles with wheels, from small bikes to large planes. There are many ways for suspension systems testing, but they can generally be categorized as invasive and non-invasive testing. The non-invasive testing is performed without unmounting the spring-damper unit, which makes the testing easier, more time-consuming, and cheaper. However; there is no non-invasive tester or methodology that is broadly used in service shops which can be suitable for motorcycles' suspension testing.

The main problem that this centers on is the non-existence of a non-invasive suspension tester that is designed for motorcycles. This problem makes testing motorcycle's suspension very complicated and less periodic. Therefore, the riders tend to leave checking the state of their motorbikes' suspensions to their predictions and feeling, which is a very inaccurate way of suspension testing and may result in causing danger to motorcycle riders and other drivers on the road. On the other hand, there are many testers and testing methodologies that are designed for four-wheel vehicle suspension testing. However; motorcycles can't be tested by the non-invasive testers that are designed for four-wheeled vehicles because of the difference in the construction of motorcycles, where it would be difficult to apply the same conditions of testing. To illustrate, unlike cars, motorcycles can't be balanced while testing without the intrusion of a person or a device that would hold the motorcycle balanced and stable during testing. In addition, there is a big difference between the kinematics of motorcycle tires' movement and the kinematics of car tires' movements. Because of these differences, it would be difficult to test motorcycle suspension on a four-wheel vehicle tester without any modifications

For these reasons, this thesis will mainly verify the eligibility of using the EUSAMA methodology for testing motorcycles' suspensions on the resonance adhesion tester which is mainly used for car suspension checking. The verification of EUSAMA methodology will include specifying the major parameters that affect the EUSAMA value the most, and how to eliminate their effect on the outcomes of testing.

2 SUMMARY OF CURRENT STATUS OF KNOWLEDGE

2.1 Search methods

The literature in this thesis has been reviewed and chosen mostly from books and scientific articles that were issued after the year 2002 except for very few sources like literature number 4 and 16 that have issued a couple of years before that.

The language of the resources is mostly, except for thesis number 5 and 8, which have been written in the Czech language, and catalog number 18 of Aprilia ETX 350 motorcycle, which has been found only in german language.

The citation of the sources has been made according to ČSN 690, with the help of Citace pro and Mendly.

The main sources from which the review of this thesis was written are two books, presented as literature 1 and 2, and five other main scientific papers introduced as literature 3,4,7,9, and 10, and these are the sources from which most of the needed information for this thesis was taken. Others are sources for secondary information, pictures, and motorcycle parameters.

2.2 Critical search

2.2.1 Definition of motorcycles

Motorcycles are made up of a large number of mechanical parts and they can be described in many different ways, however; motorcycles can be generally defined as a machine consisting of four main parts [1]:

- The rear part which includes the frame, the saddle, and the transition unit.
- The front part which includes the steering axle, the fork, and the starring bar.
- The front wheel.
- The back wheel.

These parts are connected with each other by the three main joints (two joints connect the wheels with the front and back parts, and a joint connects the front part with the back part) [1]. See figure 2-1.

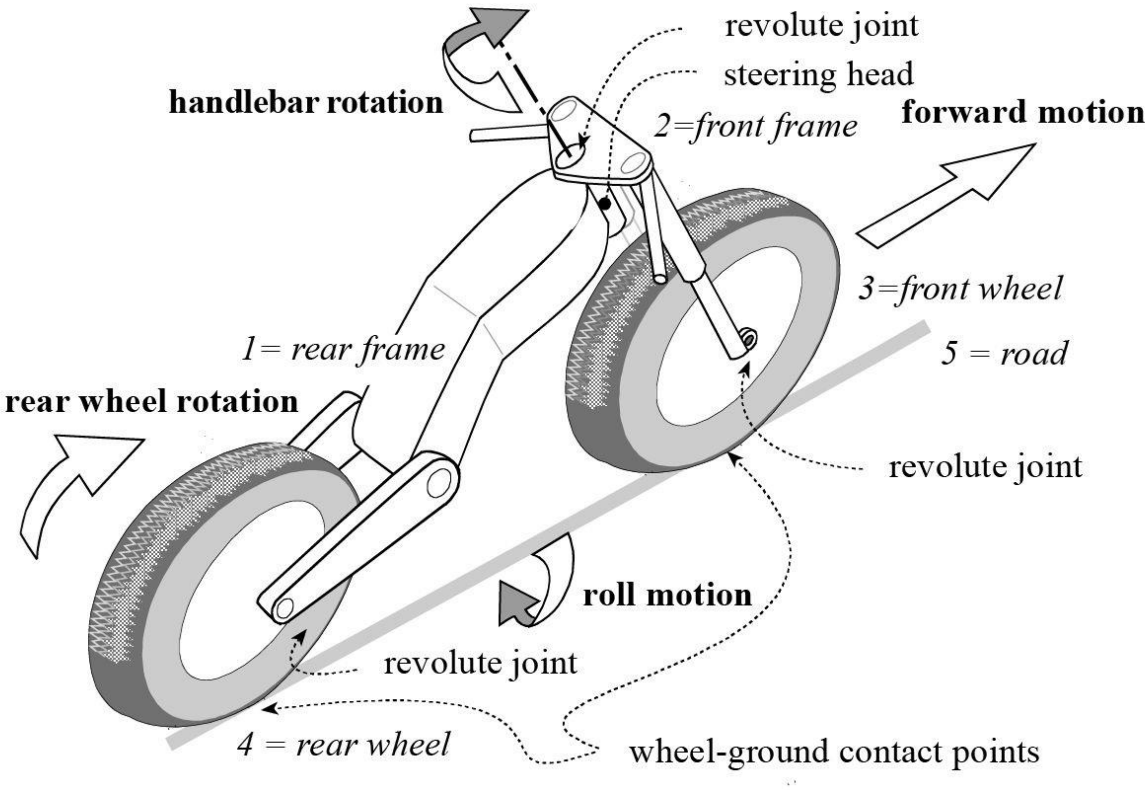


Fig. 2-1 Motorcycles Structure [1].

2.2.2 Geometry of motorcycles

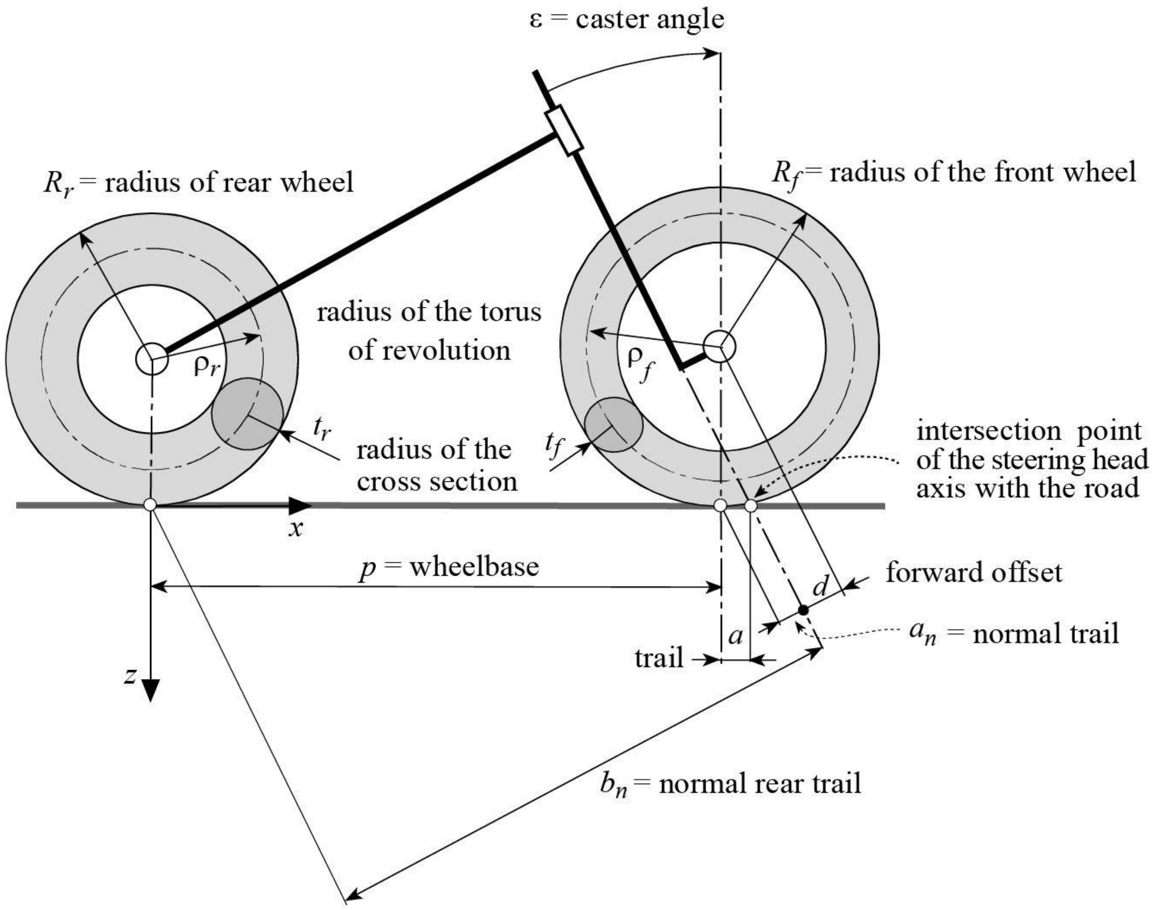


Fig. 2-2 Geometry of motorcycle [1].

The geometry of a motorcycle is a collection of important parameters that determine the handling of a motorcycle. There are a lot of parameters that identify the geometry of motorcycles like the diameter of the wheels, and the radius of their cross-section. However, some of the parameters that determine the geometry of a motorcycle the most are the caster angle, the wheelbase, and the trail where a small change of these parameters affects the stability and maneuverability [1].

- **Wheelbase**

The wheelbase is the distance between the contact of the front and back wheel with the ground. This distance varies between 1200 mm in small scooters to 1600 mm in tour motorcycles [1].

▪ **Caster angle**

The caster angle is the angle between the vertical axis of the front tire and the rotating axis of the steering head, see fig.2.1. This angle has a direct effect on the steering of the front wheel and stability in general. The angle varies mostly in the range of 19° - 27° according to the type of motorcycle. There are some motorbikes with extreme caster angles of even more than 37°, but these are rare motorcycles that are not included in our studies. [1].

The caster angle also directly affects the suspension unit and its technical state. To illustrate, small caster angles would put a lot of stress on the suspension unit while breaking, and therefore this will cause them to deform and thus cause vibrations on the front part of the motorbike [1].

The values of caster angle and the value of trails are very connected, where an increment of the caster angle should come to pass along with an increment of the trail in order to assure a safer and more comfortable drive [1].

▪ **Trail**

The trail is defined as the distance between the vertical axis of the tire and the point of intersection of the steering axis with the ground. It is directly affected by the caster angle, and the greater the caster angle, the greater the trail is. The trail varies from 75 mm in road motorcycles to 100 mm in sport and touring motorcycles. It is the most important character that assures the stability and controllability of the motorcycle. However, big values of the trails require greater efforts to steer. Therefore, sport motorbikers tend to reduce the trail to steer quicker, but this also reduces the stability of the motorcycles. For this reason, the trail should be set to a value where it isn’t so difficult to steer the wheel according to the purpose of the motorcycle, and stability is assured [1].

Figure 2-3, shows an example of the Aprilia ETX motorcycle’s geometry, and it is the main motorcycle that would be used for the simulation model creation.

Fig. 2-3 The Moto Aprilia ETX motorcycle [23].

Tab. 2-1 List of geometry parameters of the Moto Aprilia ETX and their measurements [11,18].

Geometry	Measurement
Wheelbase	1470 mm
Caster angle	26°
Front tire	90/90 x 21’’
Rear tire	130/80 x 17’’
Motorcycle length	2150 mm
Motorcycle width	560 mm
Saddle height	900 mm
Mass without filling	210 kg
Front suspension	telescopic hydraulic fork
Rear suspension	Monoshock absorber
Front Tire pressure for one person on a missed road(on/off-road)	1.6 bar
Back Tire pressure for one person on a missed road(on/off-road)	1.7 bar
Maximum tire pressure	2.5 bar

2.2.3 Types of motorcycle suspension

- **Suspension of the front wheel**

The front suspension of a motorcycle usually consists of a fork which consists of two identical sides that are connected with a yoke. Both sides consist of spring-damping units that work at the same time to assure the needed compression and the rebound of the motorcycle, and the most common type of forks in motorbikes is the telescopic fork [1].

The telescopic fork is the most common type of suspension in motorcycles due to its simplicity, low price, and beautiful design. It is made of two rods that slide inside the fork slider which are connected with the yoke (triple clamp). The other end of the rods moves inside cylinders that are connected to the tire of the motorcycle. Besides, the telescopic fork also consists of springs, oil, and a damping system made of pistons with holes for regulating the speed of the oil running through them [1].

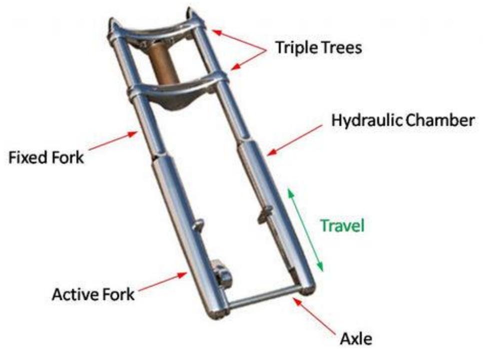


Fig. 2-4 Front telescopic fork [12].

- **Suspension of the back wheel**

The suspension of back wheels can also be made of two suspensions with two spring-damping units. Although this type of suspension could be simple in construction and could produce high amplitudes of the spring-damper motion, it has many problems and one of which is the possibility of the unevenness of forces in the two spring-damper units [1].

Another type is the mono-shock suspension, which consists of only one spring-damping unit. This type of suspension is easy to adjust and lighter which results in less unsprung mass [1]. The suspension of the back wheel can be connected to the motorcycle through a four-bar linkage or a six-bar linkage (see Fig.2-5) in order to control the progression behavior of the spring and the exact position of the suspension [1].

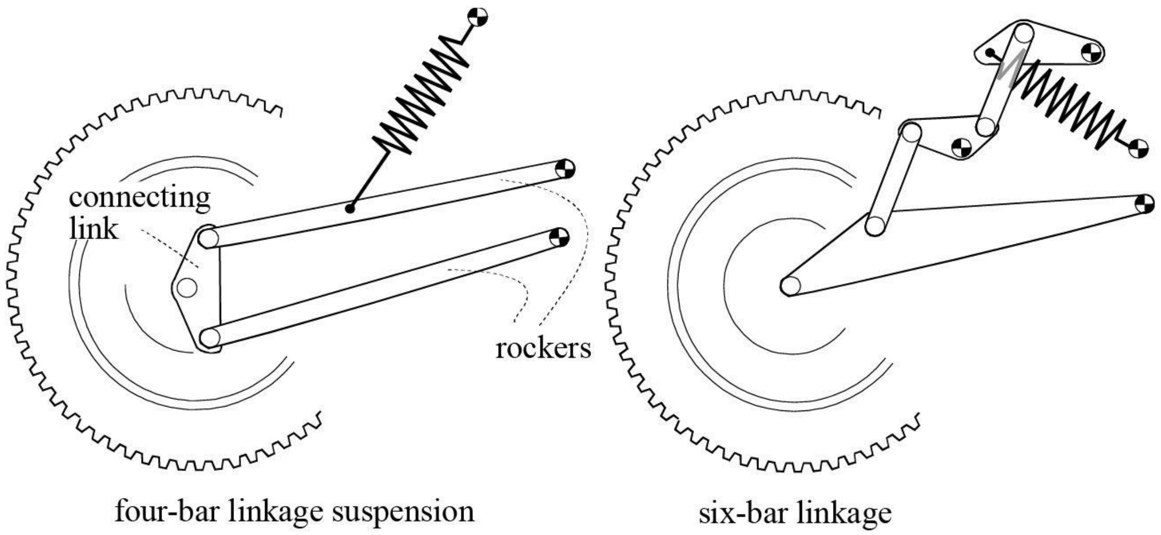


Fig. 2-5 Four-bar and six-bar linkage of back suspension [1].

2.2.4 Motorcycle suspension

The motorcycle suspension is one of the essential elements in motorcycles that affect the comfortability and the mechanical reliability of the motorcycle, and there are four main parameters that affect the performance of the suspension, summed up in the mass of the motorcycle, the springing, the damping, and the tire characteristics.

- **Tires**

Tires of motorcycles are the main elements that support the weight of the rider and the motorcycle chassis, and they transfer the power from the motorcycle to the road while accelerating, braking, turning, and cornering [2].

Along with the size of the tire and its cross-section area, the tire stiffness is one of the most important parameters that define the tire and its performance. The tire stiffness is defined as the ratio between the vertical load to the vertical deformation of the tire, and its value range between 100-350 kN/m [1].

The tire stiffness is highly affected by the inflation of the tire, where it changes the flexibility of the tire, and thus changes the ability of the tire deformation and accordingly the tire stiffness.

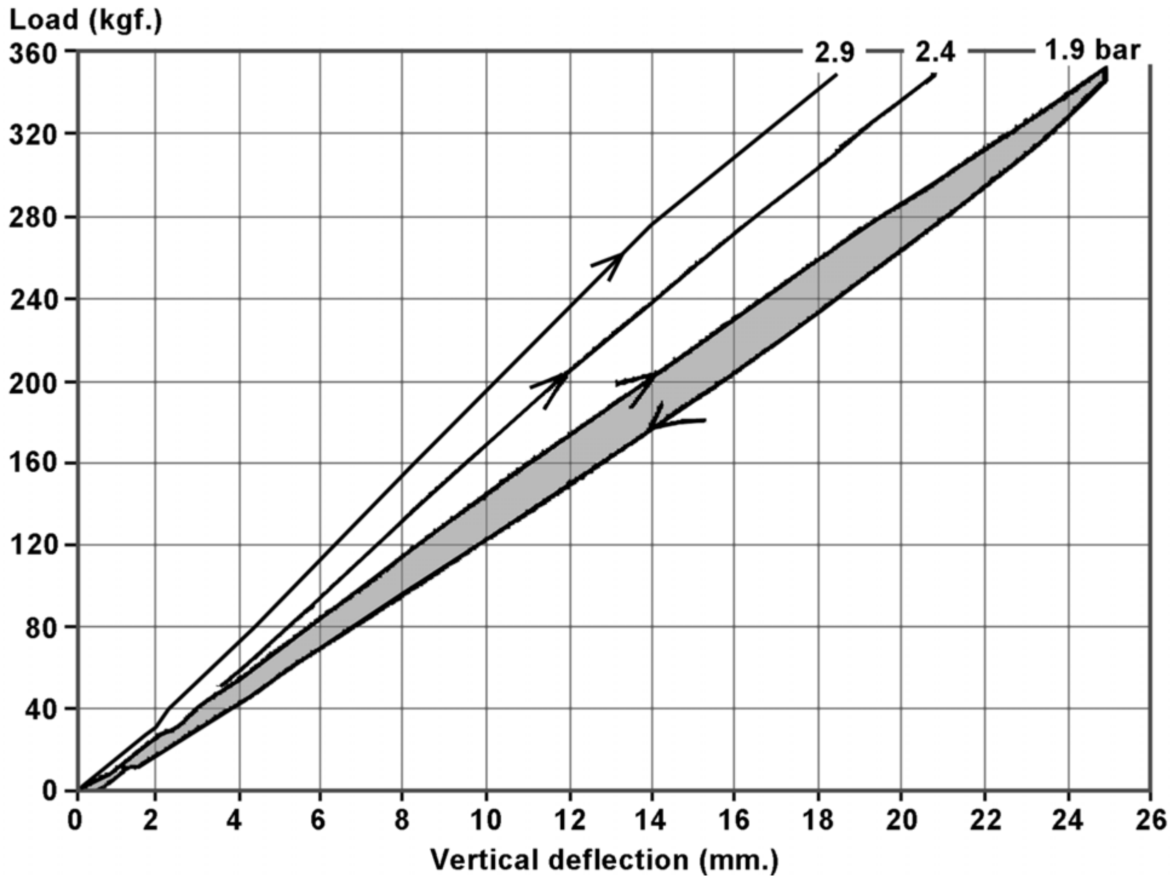


Fig. 2-6 Influence of tire inflation on tire stiffness [2].

In figure 2-6, the curves are taken from actual data, and they illustrate how the tire stiffness changes with the change of the tire pressure, where the tire stiffness is 137 kN/m (14 kgf. /mm) when the tire pressure is 1.9 bar and increases to 186 kN/m (19 kgf. /mm) when increasing the tire pressure to 2.9 bar [2].

There are many other factors that affect tire stiffness, like the temperature, the tire pattern, and the tire material, but tire inflation is the one factor that has the most influence on tire stiffness [2].

- **Spring**

The spring is one of the key components in suspension systems, and the most significant benefit of springs is that they store mechanical energy so that when a force is exerted on them, the spring will bounce back and forth till the energy is dissipated. The character that describes sprigs the most is the spring rate which is defined as the force needed to compress a spring of 1 mm [2].

The suspension stiffness of a motorcycle is nearly 6-12 times less than the tire stiffness [1].

There are many types of springs like rubber springs, gas springs, and leaf springs, but the most commonly used springs are coil springs. A coil spring is known as wire winded to several loops with a certain distance between each loop and the other, and the distance between the loops affects the behavior of the spring significantly. To clarify, if the distance between the loops were constant, then the rate of the spring would be constant, and the force needed for compressing the spring would also be constant, but if the distances between the loops were different, then the rate of the spring will differ accordingly [2].

- **Damper**

There are many usages of dampers like counteracting the kinematic motions of the tire while moving; however, the most important use of dampers is the dissipation of spring's energy. When the spring stores the energy absorbed from bumps and uneven roads, the damper will smoothly dissipate that energy so that the vehicle is not bouncing back and forth the whole time. This is done when the compressed hydraulic oil passes through a piston with small holes, changing the stored energy of the spring to kinetic energy and thus releasing as heat due to statical friction. The damper is directly affected by the velocity of the vehicle. To illustrate, if the vehicle moves fast when it hits a bump, the damper will dampen the force more so that the stored energy in the spring dissipates faster, and thus the dissipation of the energy is smoother, and driving is safer and more comfortable [2].

The damper is mainly formed by a fixed rod that is attached to a piston with small holes. Along with the fixed rod and the piston, there is another piston at the bottom of the moving part of the damper (the pressure cylinder along with the rod slides) with some holes for regulating the oil passing through them [2].

- **Sprung and unsprung mass**

The sprung mass is defined as the mass located above the suspension system and includes the rider and the chassis of the motorcycle, whereas the unsprung mass consists of the wheel mass, the bakes, suspension links, and the suspension components including the spring and the damper.

2.2.5 Types of motorcycles' suspension systems' testing

As stated before, suspension systems are essential parts of motorcycles, without which driving will be dangerous and uncomfortable. For that reason, testing suspension systems is very important to determine their technical state. Moreover, these testing suspension systems help simulate, compute, and analyze the dynamics of these suspension systems, as well as, estimate their condition and predict their durability with the lowest cost. There are a lot of testing methods, but they can all sorted into invasive and non-invasive methods.

- **Invasive testing**

This type of testing is done after removing the suspension system completely from the vehicle. To clarify, the test is generated on an oscillating motion generator like hydraulic actuators, pneumatic actuators, or motors with rigs. With these types of testers, one of the ends of the shock-absorber is fixed and the other end is then connected to the moving part of the machine. The moving part will then generate an oscillating motion whose frequency, amplitude, speed, and acceleration can be controlled. The results then will be sent to a computer that is connected to the machine where they will be analyzed and tested. The analysis is usually done by analyzing the change of the shock absorber's velocity or displacement with the force generated by it. Then by analyzing the graph of this change, we can decide the state of this suspension system and what deficits it has. The downsides of this type of testing are the high cost of such testing methods. Moreover, it is a difficult and time-consuming task to remove the suspension system of the vehicle, and the whole geometry of the motorbike is going to be changed.

- **Non-Invasive testing**

This type of testing is easier than the previous type, where the suspension system of the vehicle is tested without unmounting it from the vehicle. However, this type of testing is not very common with motorcycles, and few projects can perform motorbikes suspension testing without disassembling.

In disassembly testers, the tire(s) of the vehicle is placed on a rig that is actuated by a hydraulic/pneumatic piston or a motor that oscillates the rig with a given frequency. The dynamical force/ position/ velocity or acceleration of the motorbike is measured by a specific sensor whose data are saved as in the previous type to be analyzed and processed.

2.2.6 Resonance adhesion tester

One of the non-invasive testers that is used mainly for four-wheeled vehicle suspension testing, and which is available in the university is the resonance adhesion tester.

The resonance adhesion tester is a kind of disassembling testing method where the rig, on which the tire of the vehicle is placed, is actuated by a motor. So it mainly consists of a motor to generate the motion, an oscillating platform, a cam to transfer the motion to the oscillating platform, a dynamometer to measure the adhesion force of the tire that acts on the oscillating platform, and a computer that will be connected to the oscillating rig to analyze the measured data [7]. See figure 2.8.

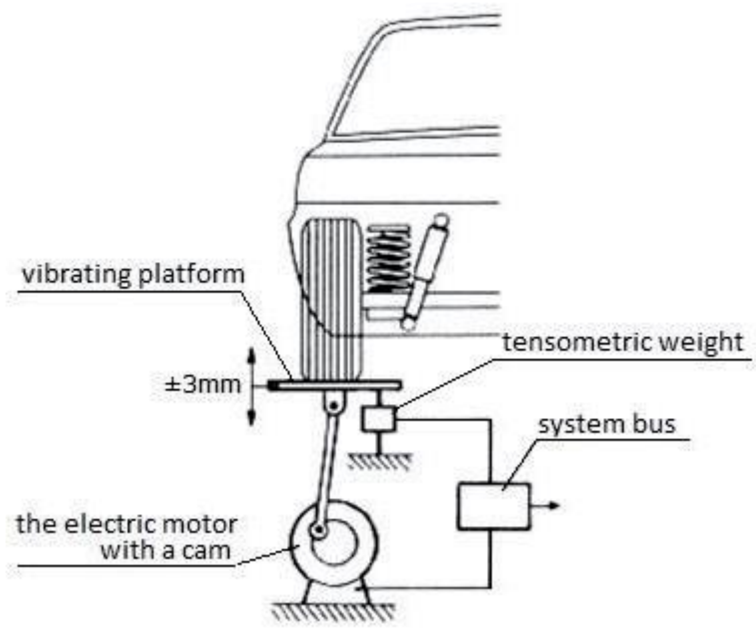


Fig. 2-7 Structure of resonance adhesion tester [7]

This type of testing is derived from the "European Shock Absorber Manufacturers Association" which is known as the EUSAMA methodology. The rig of the tester in the resonance adhesion tester is moving vertically with a stroke of ±3 mm. The test is done by actuating the oscillating platform with a frequency of 25 Hz, and then by using a frequency changer or by disconnecting the power supply, the frequency is decreased till it reaches 0 Hz. After that, the adhesion force which is acting on the oscillating platform from the suspension system will then be analyzed by a DAQ system [7].

One of the significant disadvantages of the adhesion tester is that the stroke is too high for today's roads, as they tend to be smoother with less unevenness than the roads of the seventeens when the EUSAMA methodology was originated [10].

2.2.7 Road simulator

In reality, there is no documented physical non-invasive tester designed for only motorcycles and the closest project that I have found to a non-invasive tester for motorcycles is the road simulation model.

The road simulation is used for determining the effect of the road on many characters and parts of the motorbike including the suspension, and one of these road simulators was built at Brescia University. The simulator was made of two oscillating rigs that are actuated by hydraulic pistons. It also contains an accelerometer in the wheel of each frame and a displacement transducer on each of the motorbike's suspensions as seen in figure 2-8. There is no loading mechanism in this simulator because they use the wight of the human riding on the motorcycle while oscillating. However, the holding mechanism is made of an aluminum frame, that is very light, and thus portable and easy to disassemble [3].

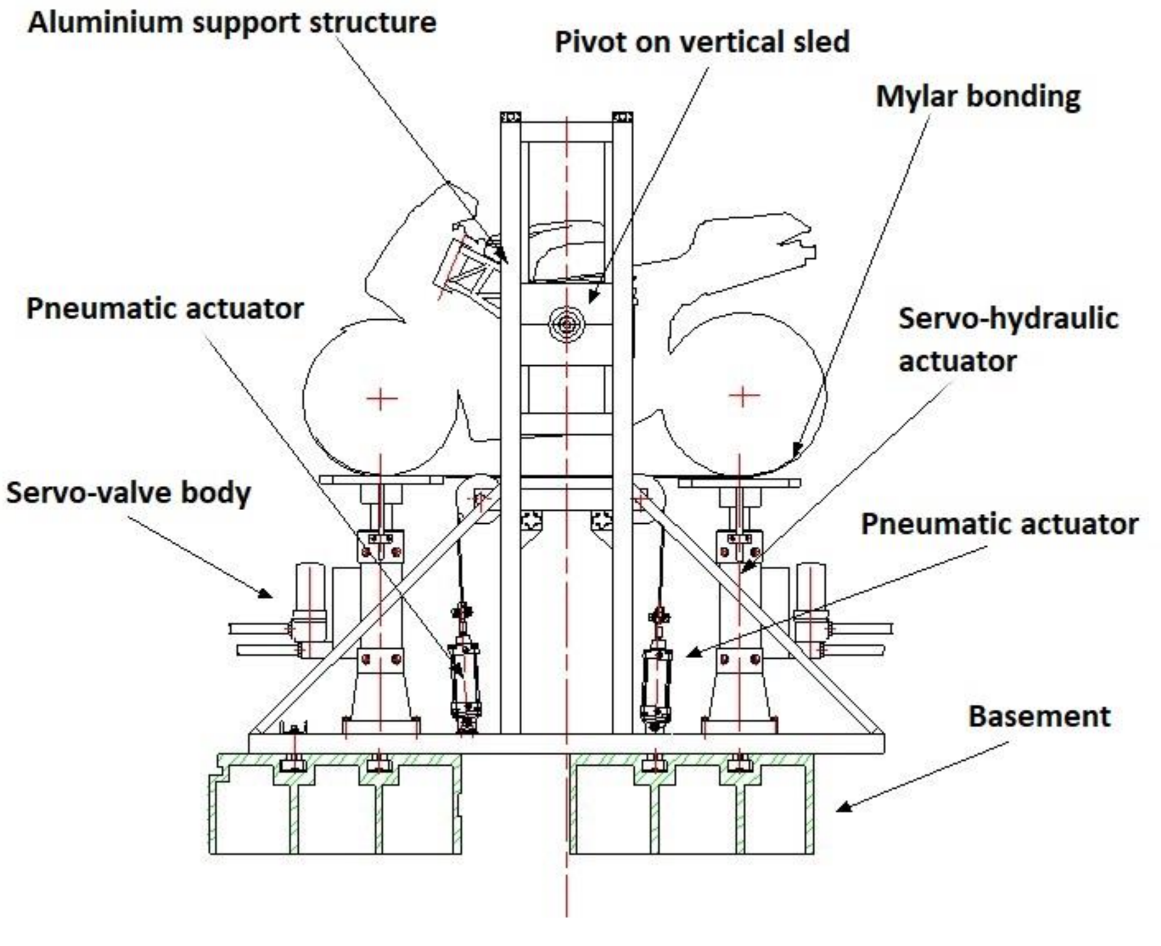


Fig. 2-8 Influence of tire inflation on tire stiffness [2].

This simulator is made to simulate the real behavior of motorcycles with regard to vertical loading, and the construction is quite complicated for suspension testing only.

2.2.8 EUSAMA Methodology

The EUSAMA methodology is a practical and well-known methodology for non-invasive suspension systems that is used for checking the technical state of cars' suspension systems.

The EUSAMA is an abbreviation of the European Shock Absorber Manufacturers Association that was established in 1971. This association has set the criteria for evaluating the functionality of suspension systems. This methodology uses the adhesion value to test the technical state of the suspension system, where the adhesion value is the percentage of the ratio between the minimum dynamical force acting on the oscillating platform and the static force that represent the proportion of the motorcycles' wight that is placed on the oscillating platform [4]. See figure 2.9.

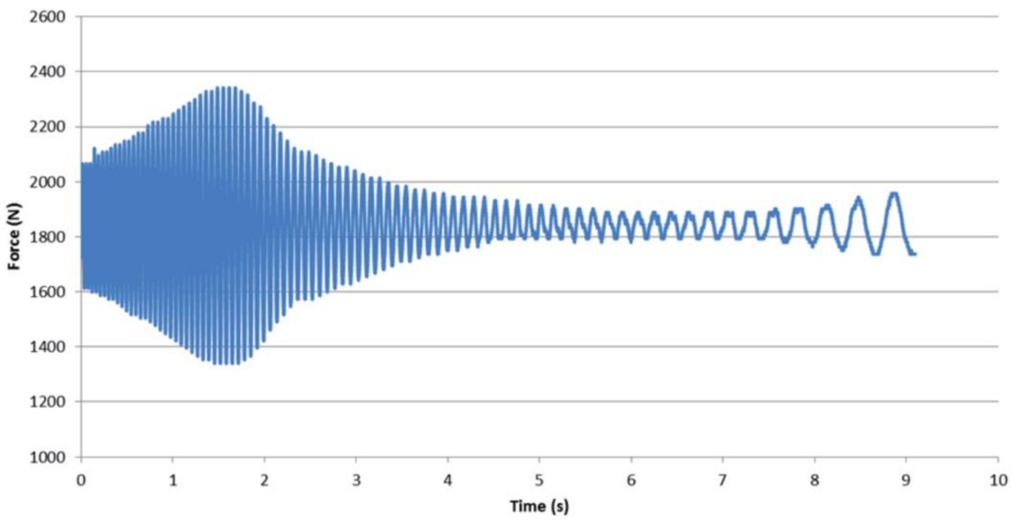


Fig. 2-9 actuating signal of the resonance adhesion tester [9].

$$EUS\,\% = 100\frac{F_{dyn\_min}}{F_{sta}}\,\% \tag{1}$$

Where the statical force is the weight of the sprung and unsprung mass.

$$F_{sta} = (m_1 + m_2)g \tag{2}$$

The minimum value of the dynamic force was chosen due to the fact that this force occurs on the natural frequency of the unsprung mass which is the frequency at which the system is vibrating the most [4]

Then we compare the percentage value of the adhesion value with the values in table 2-2.

Tab. 2-2 EUSAMA methodology criteria.

Minimum adhesion Value [%]	State of Shock absorber
More than 61	In an excellent condition
41 - 60	In a good condition
21 – 40	In a satisfactory condition
1 - 20	In an unsatisfactory condition
0	In an ineffective condition

EUSAMA methodology is a very easy, fast, and understandable method to assess the technical state of a suspension system [9]. Moreover, a huge range of cars can be evaluated by this method, regardless of the manufacturing company. To clarify, by EUSAMA methodology, there is no need for tabular data from the tested car, which might be impossible to get due to the big number of vehicle manufacturers. However, the value of the EUSAMA percentage is calculated by the minimum dynamic force and this force is directly affected by the weight of the vehicle, which can be a downside of the EUSAMA. To illustrate, a heavy vehicle with a suspension system in a bad condition might have a higher value of Eusama percentage than a lighter car with a suspension system in a good condition [5]

2.2.9 Phase shift

At the present time, resonance adhesion testers have some difficulties reliably assess the suspension of modern cars, and therefore there are efforts to adjust the analysis methods by considering other criteria [22].

One of the most advanced methods is the introduction of an auxiliary criterion called the phase shift, which detects the delay between the adhesion force and excitation.

The phase shift is the angular difference between the amplitude of the input sinusoidal signal that describes the position of the oscillating platform and the amplitude of the sinusoidal dynamical force between the tire of the vehicle and the oscillating platform [4]. See figure 2.10

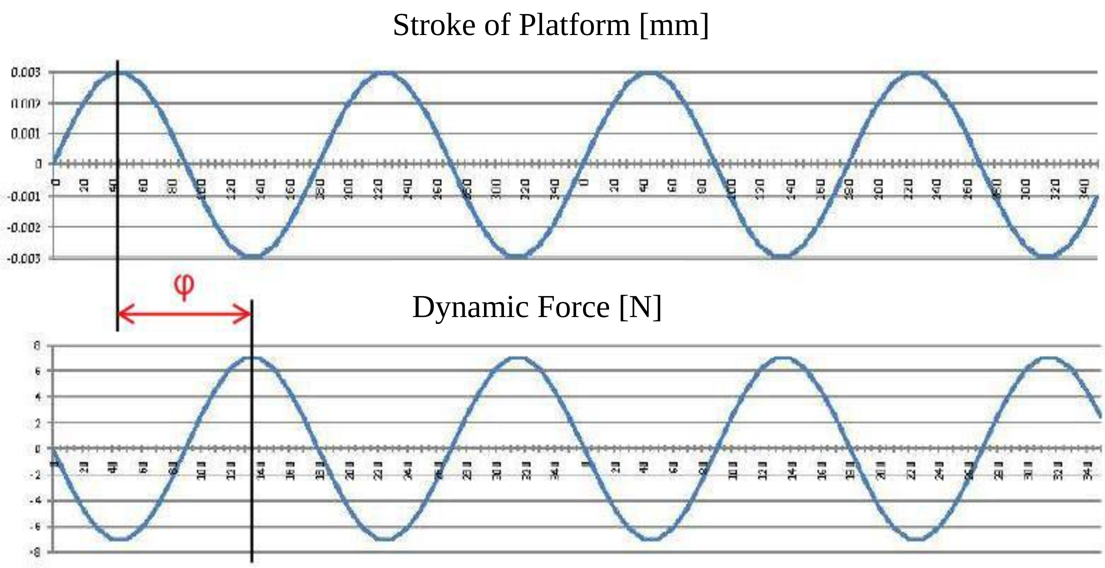


Fig. 2-10 Illustration of phase shift[5]

New resonance adhesive testers have the ability to draw the phase shift which can also be considered a criterion for checking the state of suspension systems. The minimum phase shift is observed in the range between the resonance frequency of sprung and unsprung mass, where the resonance frequency of the sprung mass can be between 1-3 Hz for automotive suspensions, and between 10-20 Hz for the unsprung mass [4]. If the minimum value of the phase shift is less than 40 °, then the state of the shock absorber is bad according to this criterion [5]

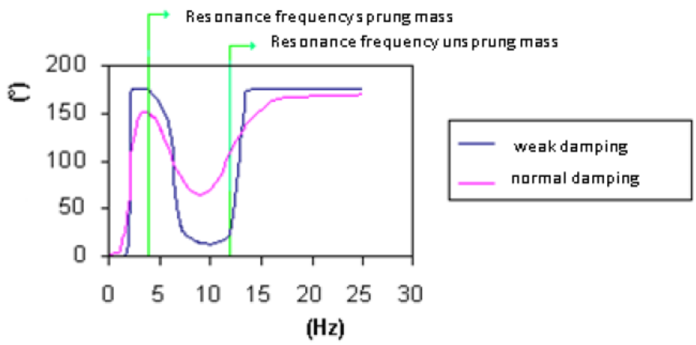


Fig. 2-11 The range of finding the minimum angle of phase shift [5]

For the phase shift, the damping constant of the suspension plays a very huge role. To illuminate, with the absence of damping in the suspension system, the phase shift will be 180° at the resonance frequency of the sprung mass and will equal 0° at the resonance frequency of the unsprung mass. Therefore, the increment of damping will decrease the phase shift at the resonance frequency of the sprung mass, and increase it at the resonance frequency of the unsprung mass [4].

2.2.10 Damping Ratio

The damping ratio is a very simple and clear additional criterion used by the EUSAMA methodology, that evaluates the efficiency of shock absorbers and their capability to achieve their purpose [10], and this criterion can be used for models with only one degree of freedom [19].

For calculating the damping ratio, the linear suspension damping is devided by the critical damping value [19], where the critical damping value is the minimum value of damping that allows the mass to stop at the equilibrium position after the first displacement [4].

$$\zeta = \frac{b}{b_{critical}} \tag{3}$$

where

ζ is the damping ratio

b is the damping of the suspension system

$b_{critical}$ is the critical damping of the system

Equation 5 and 6 are used to calculate the approximated values of critical damping of the sprung and unsprung masses [4]

$$b_{critical\_sprung} = \sqrt{K_{equ} m_1} \tag{4}$$

$$b_{critical\_unsprung} = 2\sqrt{m_2 (K_1 + K_1)} \tag{5}$$

Where

$b_{critical\_sprung}$ and $b_{critical\_unsprung}$ are the critical damping values of the sprung and unsprung masses respectively.

K_{equ} is the equivalent stiffness of the system which can be calculated by the following equation [4]

$$K_{equ} = \frac{K_1 K_2}{K_1 + K_2 (1 + \frac{m_2}{m_1})} \tag{6}$$

The minimum value of damping ratios of both the sprung and unsprung mass should not be less than 0.1 [7]

If the EUSAMA value was more than 20% and the damping ratio is less than 0.1, then the suspension system is safe. However, if the opposite happened, where the damping ratio's condition was met but the EUSAMA's not, the diagnosis might be inaccurate. The acceptable EUSAMA result is not attained by proper damping, and that is because a low damping ratio means that the damper is not able to dissipate the energy stored in the spring even if the test indicates good adhesion [7].

2.2.11 Characteristics of the suspension system in motorcycles [1]

Three main characteristics determine the suspension systems of motorcycles and their functionality, these characteristics are the stiffness of the spring, the damping coefficient, and the preload of the shock absorber [1]. However, there are a lot of parameters that control these characteristics, like the weight of the motorcycle, the weight of the rider, the center of gravity of the motorcycle, the moment of inertia, the geometry of the motorcycle, the type of the spring and the type of the damper.

This information is very important for making dynamical models of motorcycles' suspension systems, which will thus help us to verify the tasting methodology for determining the state of these suspension systems.

- **Preload**

The preload of a spring is the pre-compression of it. To illustrate, if a spring is preloaded, it means that the normal state of the spring is being compressed by a specific amount. Therefore, if a force is applied to a spring that is less than or equal to the pre-load, the spring is not going to be deformed at all. For that reason, the force exerted by a spring without a preload is calculated by equation 7, where K is the stiffness of the spring, and y is its deformation. Whereas equation 8 calculates the force exerted by the spring that is preloaded, where Δy is the deformation caused by the preload [1].

$$F_s = K\,y \tag{7}$$

$$F_s = K\,y + K\,\Delta y \tag{8}$$

the preload is used according to the weight of the rider and the type of the road to adjust the stiffness of the spring. It is also used to increase the limit of the maximum force that a spring can endure [1].

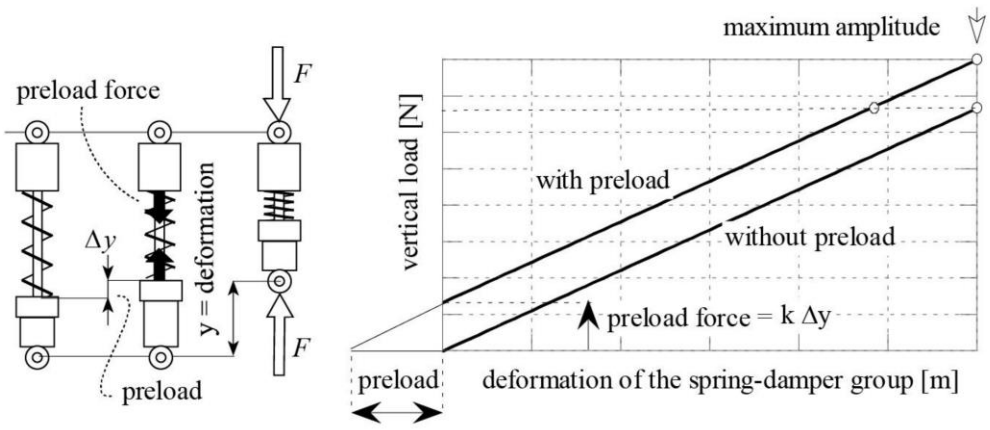


Fig. 2-12 Preload of a motorcycle's strut, and a graf showing the relationship between the vertical load and the deformation of the spring when preloaded and without preload [1].

As seen in figure 2.12, the maximum force of a spring when preloaded is more than the force of the same spring without preloading under the same maximum deformation.

To add, the preload also governs how much the spring of a shock absorber prolongate. To clear up, when going on a road with holes, the spring of the shock absorber of the motorcycle is supposed to elongate by the same length as the depth of the hole. Therefore, if the spring is not preloaded, the spring will be able to extend with a value equal to the deformation caused by the static load [1]. See figure 2-13.

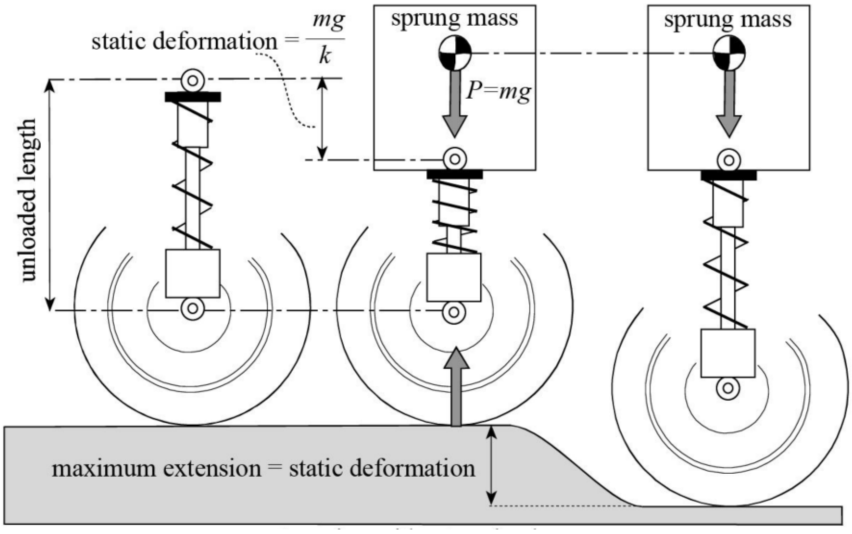


Fig. 2-13 The maximum extension of spring without preload [1].

$$y_{max} = \frac{m\,g}{K} \tag{9}$$

When the spring is preloaded, the maximum extension of that spring is going to be less, where it will be equal to the deformation caused by the static load minus the distance caused by the preload [1]. See figure 2.14 :

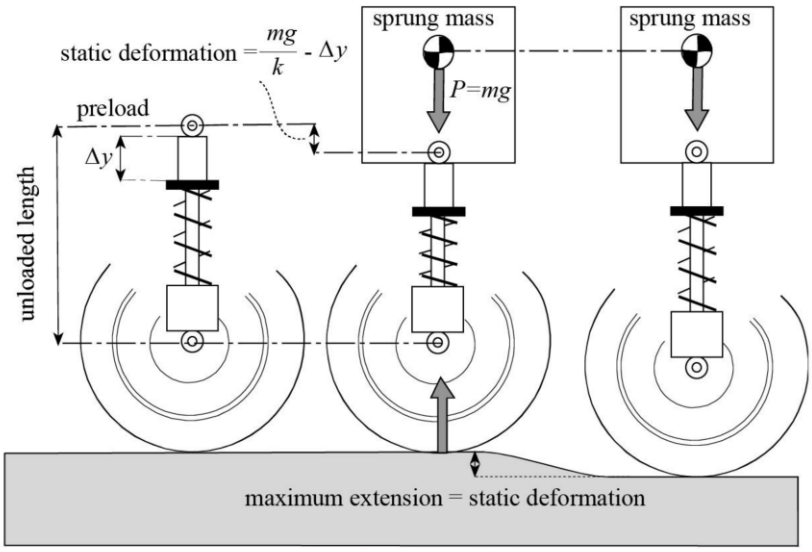


Fig. 2-14 The maximum extension of spring when preloaded [1].

$$y_{max} = \frac{m\,g}{K} - \Delta y \tag{10}$$

From equation 10, it can be seen that if the spring's deformation is equal to the deformation caused by the preload, the spring is not going to extend at all.

- **Force-Velocity Characteristics**

The F-V characteristic is a graf that represents the dependence of the force produced from a shock absorber on the velocity of it when tested on a hydraulic piston.

The tester is a hydraulic piston that tests a shock absorber by extending and compressing it with a certain frequency, stroke, velocity, and acceleration. After that, the tester will measure the reacting force of the shock absorber and send the data to a computer connected to the tester which will thus draw the dependence of the shock absorber on its velocity (F-V dependence) and stroke (F-Z dependence). See figure 2.15

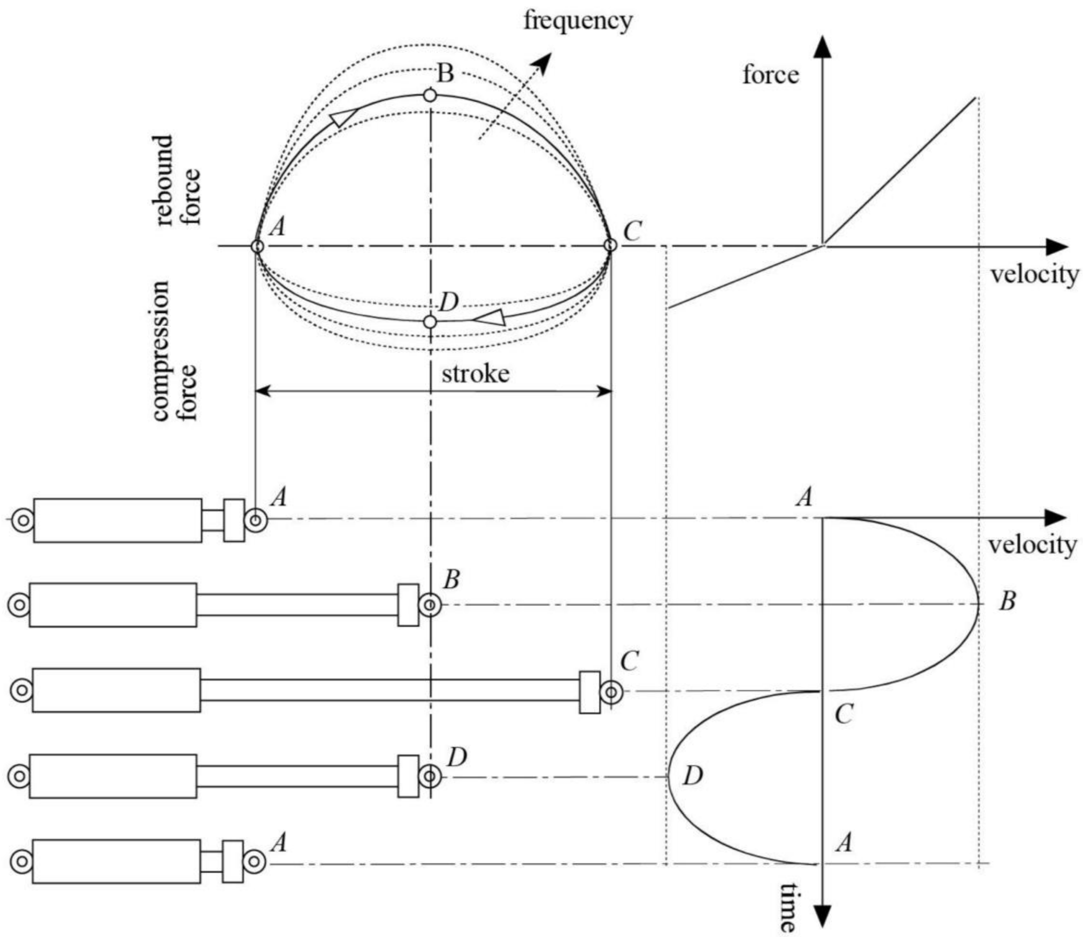


Fig. 2-15 The F-Z graph (left) and F-V graph (right) [1]

The closed area on the F-Z graph represents the dissipated energy of the shock absorber, where the upper part of the graph represents the dissipated energy while expansion and the lower area represent the dissipated energy while compression [1]

The damping constant in compression is less than the damping constant in extension (usually by less than half) so that when the motorbike hits a bump, the suspension system will not generate a lot of reaction force and thus, the tire will follow the profile of the bump or step. However, when the motorcycle goes over a small hole, the tire will jump over the hole within a very short time of losing contact with the ground due to the high reaction force of the shock absorber [1]

- **Stiffness**

As explained before, the stiffness of a spring is the most important parameter to identify the spring characteristics, and it represents the force needed to compress the spring by 1mm.

There are three main types of springs according to the stiffness:

1. Linear rate spring
   In this type of springs, the distance between the loops is the same, and therefore, the stiffness of the spring is then constant. These types of springs are good for predicting the behavior of the spring, as the vertical force of the spring increases linearly with the displacement [1].

2. Progressive rate spring
   This type of springs has different distances between the loops, which results in different stiffnesses on the same spring. These types of springs are used to calibrate the springs' rates so that the ride becomes more comforting with small bumps, but safer when bumps are big and dangerous. The rate of such springs increases exponentially with the increment of the displacement [1].

3. Degressive rate spring
   This type of springs is very similar to the progressive rate spring in the fact that they also have different spaces among the loops. However, in this type of springs, the rate decreases exponentially with the increment of the displacement [1].

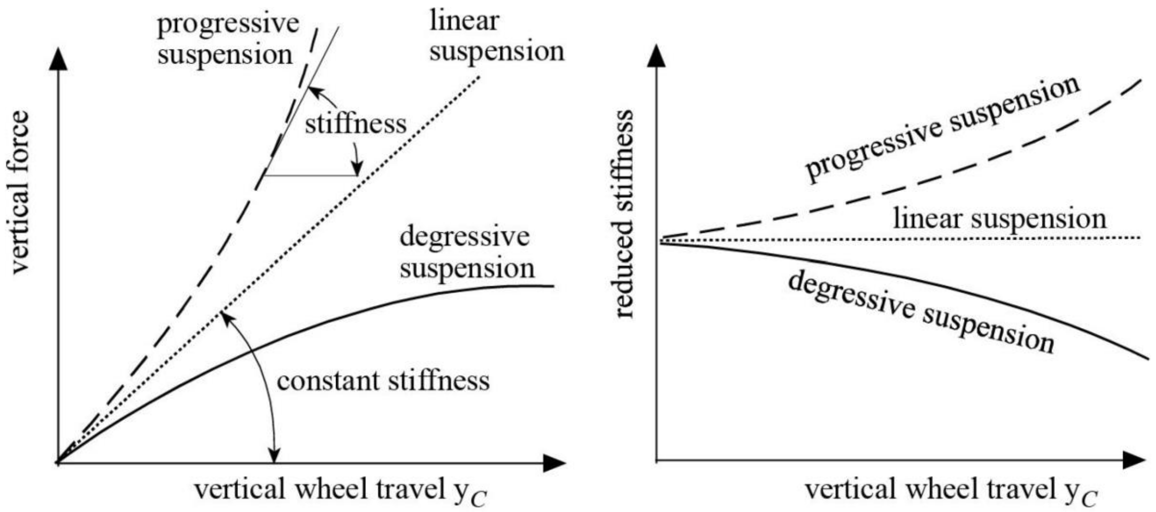


Fig. 2-16 Relationships between the vertical displacement with the stiffness and the vertical force of different types of springs [1].

- **Damping**

As stated before, the damping of a suspension system is responsible for the dissipation of energy, and the damping is very dependent on the velocity.

The damping can be linear, which means that the dependence of the damping force on the velocity is linear and can be calculated with the following equations [1],

$$F_d = \begin{cases} b_c \, v \\ b_e \, v \end{cases} \tag{11}$$

Where b_e is the damping constant in expansion, b_c is the damping constant in compression, and v is the velocity [1].

However, the damping can be non-linear and it can be progressive and degressive as shown in figure 2.17

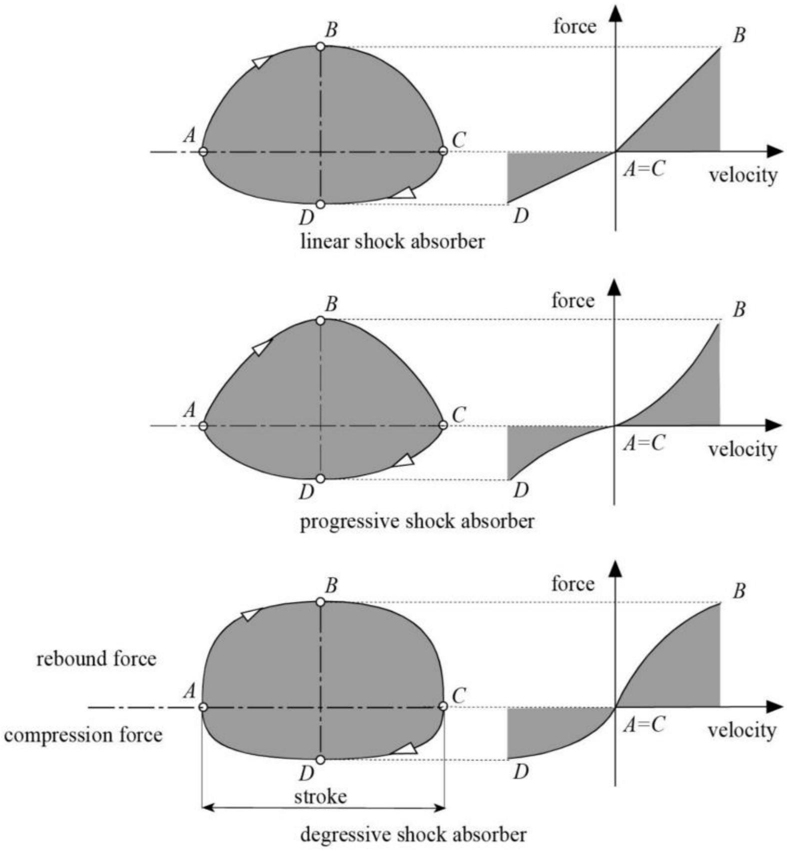


Fig. 2-17 Damping force dependence on the velocity of linear, progressive, and degressive damping [1].

The most significant advantage of a degressive damper is that it dissipates more energy than a progressive damper with the same force. This can be seen from the bigger closed area of the degressive damper on the F-Z graph [1].

The damper force of the degressive and progressive dampers can be calculated by equations 13 and 14 below [1].

$$F_d = \begin{cases} (c_c + \Delta c_c |v^n|)\, v \\ (c_e + \Delta c_e |v^n|)\, v \end{cases} \tag{12}$$

$$F_d = \begin{cases} (c_c - \Delta c_c |v^n|)\, v \\ (c_e - \Delta c_e |v^n|)\, v \end{cases} \tag{13}$$

Where n is the dependence degree of the damping constant on the velocity and Δ is the amplitude of the stroke of the shock absorber [1].

- **Stiffness and reduced stiffness of front suspension of motorcycles**

As know, the suspension units of motorcycles are tilted with a caster angle, and therefore it is quite difficult to deal with such tilted spring-damping systems. For that reason, reduced suspensions are introduced. It means that if a specific force is applied to a spring, the original spring will have the same deformation value as the one that will be gotten when the same force is applied to the same spring but with reduced stiffness [1]. See figure 2.18:

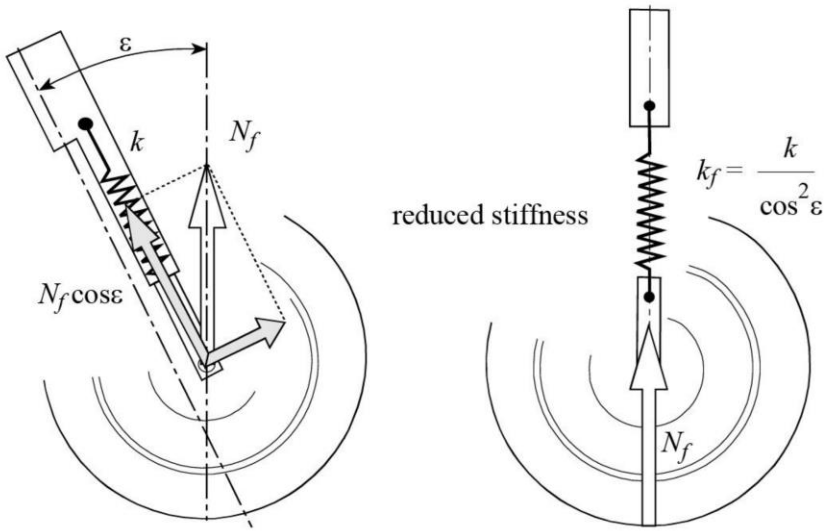


Fig. 2-18 The shape of an original suspension system and the shape of it when reduced [1].

$$k_f = \frac{k}{cos^2\ \varepsilon} \tag{14}$$

Where k_f is the reduced stiffness of the spring, k is the original stiffness of the spring, and ε is the caster angle.

To calculate the reduced damping of the front suspension system of the motorcycle, the following equation will be used [1]:

$$b_f = \frac{b}{cos^2\ \varepsilon} \tag{15}$$

Where b_f is the reduced damping coefficient of the spring, b is the original damping coefficient of the spring, and ε is the caster angle. And the reduced damping coefficient means that if a specific velocity is exerted on the spring-damper unit of the motorcycle, the original damper will generate a rebounding force that is equal to the one that will be generated by the vertically reduced damper if the same velocity is applied to it [1].

And since the fork consists of two spring-damping units, the resultant reduced stiffness of the springs will be:

$$k_f = \frac{2\,k}{cos^2\,\varepsilon} \tag{16}$$

And the resultant of both damping coefficients will be:

$$b_f = \frac{2\,b}{cos^2\,\varepsilon} \tag{17}$$

2.2.12 Simulating model of a motorcycle

It is a computer-made model that simulates the vibrations caused by the unevenness of the road. The simulating model simplifies the suspension system of the motorcycle and what affects it to a dynamic model that is easy to be studied. In other words, by using this model, it would be possible to simulate the effects of the vibration of the road on the suspension system, study the motorcycle's dynamics, and predict and analyze the changes that happen to the suspension system when optimizing it. This simulation is usually done with computer-aided programs like Matlab and Matlab Simulink.

For modeling the suspension system of a motorcycle, the spring-damper system is considered, the tire and its parameters, and the weight of the vehicle and the ride as well. See figure 2-19.

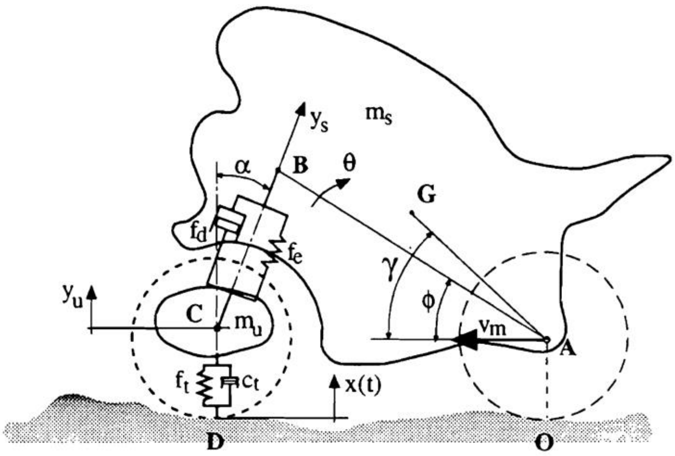


Fig. 2-19 The replacement of the tire and the suspension system in the dynamical model [16].

As seen in figure 2-19, the tire is replaced by a spring with a specific stiffness and a damper with a specific damping coefficient that can be determined according to the temperature of the tire and the air pressure indie it. Moreover, the spring damping unit is also replaced by a spring and damper with a specific stiffness and damping coefficient. However, the replaced spring and damper unit are tilted, and therefore, calculated reduced stiffness and reduced damping coefficient will be used.

Therefore, the model will be as in figure 2-20.

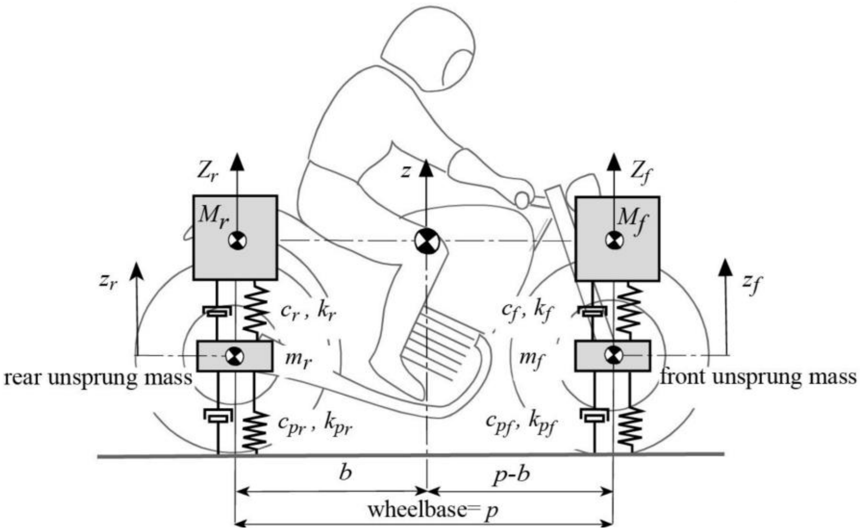


Fig. 2-20 The dynamical model of the front and back parts of the motorcycle [1].

As seen in figure 2-20, the mass of the rider and the mass of the motorcycle are distributed into two parts, the front, and the rear part, according to the position of the center of mass [1].

The same components will form the rear dynamical model of the motorcycle; however, in the course of this thesis, only half of the motorcycle is studied at the time, either in the simulation model or in the experiments.

To study the dynamic part of a motorcycle, the resonance frequency of the system is very important because the desired values to study are the results of testing when the frequency of the actuated signal meets the resonance frequency of the system.

The resonance frequency of the front dynamical system is calculated in Hz by the following equation [1]:

$$fo_f = \frac{1}{2\pi}\sqrt{\frac{K_f}{M_f}} \tag{18}$$

And the resonance of the rear dynamic system will be calculated in the same manner [1]:

$$fo_r = \frac{1}{2\pi}\sqrt{\frac{K_r}{M_r}} \tag{19}$$

2.2.13 2 Degrees of freedom model

The 2DOF model consists of three main masses:

- Sprung mass that represents the mass of the ride, the chassis of the motorcycle, and other accessories like a bag.
- Unsprung mass that represents the mass of the tire and suspension unit.
- The mass of the platform, that can be neglected in most cases.

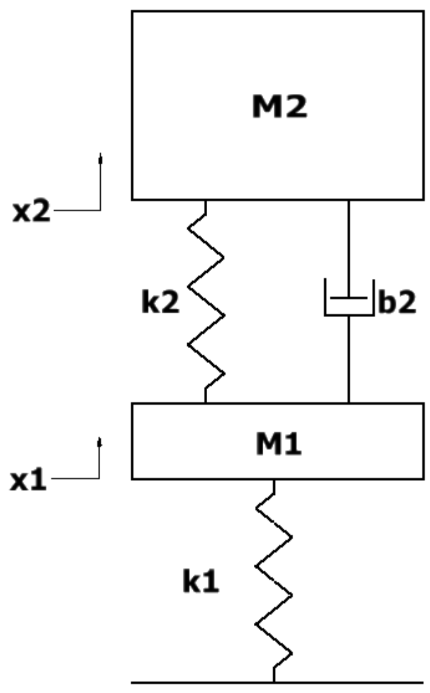


Fig. 2-21 2DOF motorbikes' model.

When applying a force on the rig, reaction forces from the springs and dampers will be generated as shown in figure 2-22

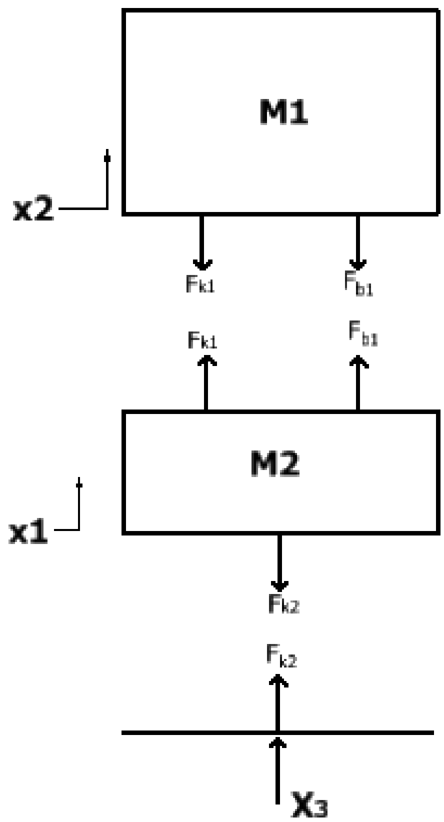


Fig. 2-22 Reaction forces on the dynamical model.

The reaction forces from the springs and dampers are calculated as the following:

$$F_{K1} = K_1(x_1 - x_2) \tag{20}$$

$$F_{c1} = C_1(\dot{x}_1 - \dot{x}_2) \tag{21}$$

$$F_{K2} = K_2(x_2 - x_3) \tag{22}$$

Where F_{K1} and F_{K2} are the reaction forces from the spring of the suspension system and the wheel of the motorbike respectively.

F_{c1} is the reaction force from the damper of the suspension system of the motorbike.

K_1 and K_2 are the stiffness of the spring from the suspension system, and the stiffness of the motorbike's tire respectively.

b_1 is the damping constant of the spring from the suspension system.

$x_1, x_2\ and\ x_3$ are the displacement of the sprung mass, unsprung mass, and the rig respectively.

$\dot{x}_1, \dot{x}_2\ and\ \dot{x}_3$ are the velocity of the sprung mass, unsprung mass, and the rig respectively.

Then each of the two masses will be analyzed individually according to Newton's law, where it reveals that the mass multiplied with the acceleration of it is equal to the summation of the forces acting on that mass.

The formulations of the dynamical system of a motorbike with two degrees of freedom:

$$m_2 a_2 = -K_2(x_2 - x_1) - C_2(\dot{x}_2 - \dot{x}_1) + K_1(x_3 - x_2) + C_1(\dot{x}_3 - \dot{x}_2) \tag{23}$$

$$m_1 a_1 = K_2(x_2 - x_1) + C_2(\dot{x}_2 - \dot{x}_1) + F \tag{24}$$

Where F is the input force that the oscillating rig will be actuated with.

2.3 Summary and analysis of the literature review

The Literature review started by defining motorbikes in order to define the scale of vehicles that are considered in this thesis. Then the geometry of motorcycles has been presented, by stating the elements that affect the dynamics of motorcycles the most, including the caster angle which is one of the main points in this thesis. After that, the main types of the motorcycles' front and back suspensions have been stated, illustrating the difference between them in terms of their construction. Later, the suspension of motorcycles has been explained in terms of its structure, where it was stated that each suspension is a combination of the tire, the mass, and the spring-damper unit of the motorcycle. Moreover, the elements that define and affect each of the suspension components were introduced, along with the ranges of some of these elements.

Then the testing methods have been presented, and it was clarified that testing methods can be classified into two main categories, the disassembly tester, and the assembly tester which is not suitable for service workshops due to its complexity. It has been also mentioned that there is no non-invasive tester for motorbikes' suspensions available for service purposes, where the only tester that has been found was a road simulator constructed in a university in Italy, that is used as a general raod simulator which is not specialized for motorcycle suspension testing and whose construction is quite robust for such specific purpose.

Conversely, one of the disassembly testers introduced in the literature review was the resonance adhesion tester which is one of the testers used in our university for car suspension testing. The resonance adhesion tester looks like a good platform to develop our motorbikes tester due to its simplicity and availability. It was stated that the resonance adhesion tester is based on the EUSAMA methodology which has been established in the seventeens for car suspension assessment. The EUSAMA methodology hasn't been used for motorcycles suspension testing, and therefore it is very essential in this thesis as a great part of this work will be related to it. In addition, other methodologies have been brought up in the literature review as they are used to support the results gotten from the EUSAMA methodology.

Lastly, the dynamic model has been demonstrated mathematically and structurally in a way that can be studied and simulated. To illustrate, the structure of the motorcycle dynamic model has been studied in terms of the mass distribution between the front and the back wheel, as well as the distribution of the sprung and unsprung mass. Moreover, the dynamic model also includes the replaced parameters of the suspension unit and the tire. To add, in the literature review, the simplified reduced stiffness and reduced damping that is caused by the tilt of the suspension unit have also been considered in order to simulate the dynamics of motorcycles more precisely. Besides the structural illustration, the mathematical equations have also been listed, which will change the graphical model into a mathematical model that can be studied and analyzed as a simulation of motorcycle dynamics.

In conclusion, it was clear that there is no documented motorcycle suspension tester that is specialized in testing the technical condition state of motorcycle suspensions and that the EUSAMA methodology, that is used for testing car suspension, could be a suitable choice to examine its eligibility on being used for motorcycle suspension testing.

3 AIM OF THE THESIS

3.1 Scientific question

What are the parameters that would affect the values of EUSAMA the most?

3.2 Objectives of the thesis

The main aim of this thesis is to verify the usability of the EUSAMA methodology that is mainly used for car suspension technical state assessment, and see if it can be for testing motorcycle suspension systems. This process will be achieved by both measuring experiments and dynamical simulations.

The partial objectives of the thesis:

- Design of a simulation model for the resonance-adhesion test of the motorcycle suspension,
- Identification of input parameters of the simulation model,
- Sensitivity study and analysis of critical operational parameters by experiment and simulation,
- Assessment of the impact of the selected parameters on the precision of the test results

3.3 Hypothesis

It is hypothesized that the weight of the rider would play a great role in the EUSAMA because of its direct impact on the contact force. Moreover, the tire stiffness and the platform angle should also affect the EUSAMA value due to their big impact n the dynamical force.

The effect of these parameters is supposed to be as the following:

- The increment of the weight increases the EUSAMA value.
- The increment of the tire stiffness decreases the EUSAMA.

4 SOLUTIONS AND METHODS

4.1 Methodologylogy

The problem of this thesis is relational, where it focuses on studying the relations of some of the motorcycle parameters on the Eeusama and how these parameters affect it. Looking at these relations will help in checking the eligibility of using EUSAMA for motorcycles, and what modifications are needed to be done to adapt the tester for motorcycles.

The scheme below shows the steps of the solution.

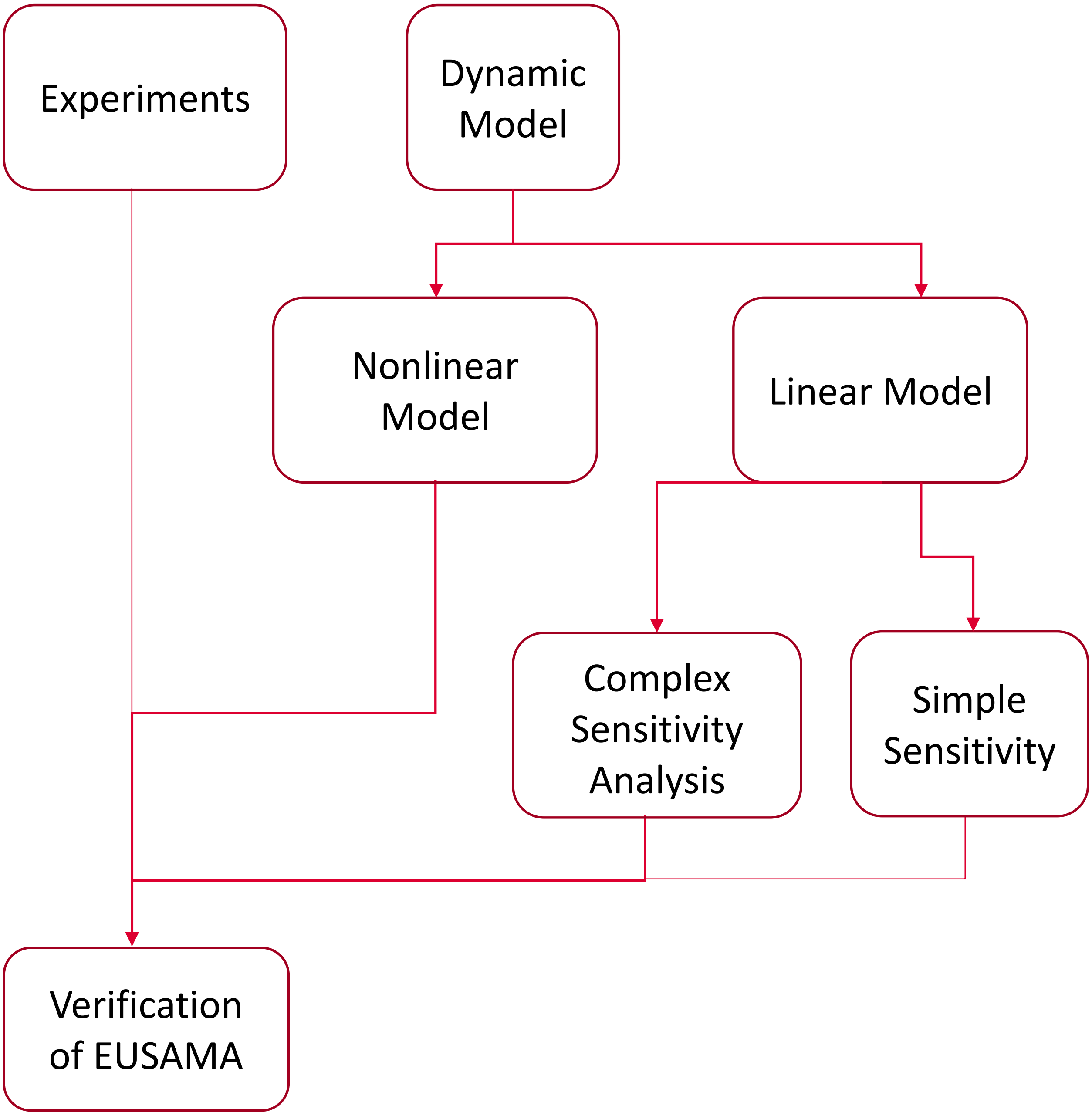


Fig. 4-1 Scheme ofsolutionn.

4.2 Experimental equipment and instrumentation

4.2.1 Tester's design

The regular platform used for car suspension assessment is a flat rig that moves in a vertical motion. However, the platform used for testing is an angle-adjustable rig which is a former project that has been constructed at the university. Due to the fact that the suspension units of motorcycles are tilted, the platform used for testing in this thesis is an adjustable angle platform whose angle can be adjusted in a range between 0° and 30° according to the caster angle of the motorcycle.

The platform has 6 different angle adjustments, starting from 0° up to 30° with a 6° increment in each step, for illustration see figure 4-2, 4-3

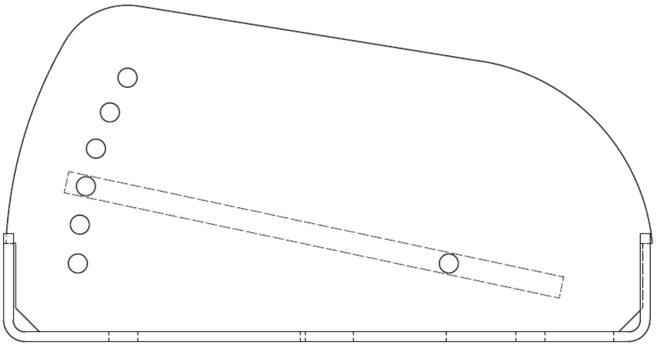


Fig. 4-2 Oscillating rig mounted on the tester.

Fig. 4-3 Oscillating rig mounted on the tester.

In the course of this thesis, I have helped to design and verify the stabilizing device during the experiments. The stabilizing mechanism consisted of a simple horizontal U beam with two vertical cylinders. A rope was tied between both cylinders for a steadier grab while testing. For more stability, weights were added on both sides of the horizontal beam. See figure 4-4 for a graphical understanding of the holding mechanism.

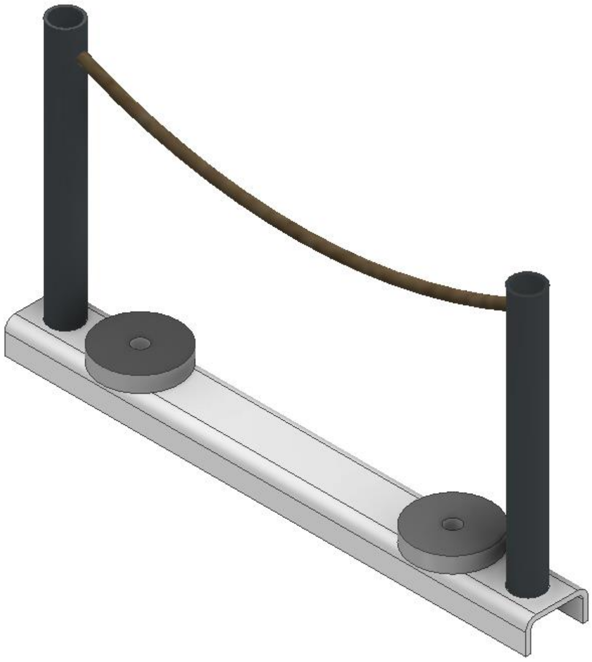


Fig. 4-4 Stability mechanism.

Experiments were performed on the resonance adhesion test ST400, which contains an electric motor with a power of 1.5 kW. The motor is connected to a rotating cam that is responsible for transforming the motion from the motor to the vibrating platform. The tester can endure up to 4000 kg as a drive-over weight and 1700 kg as a maximum testing weight. The tester primarily uses the EUSAMA methodology (0-100%) [17]. See figure 4-5

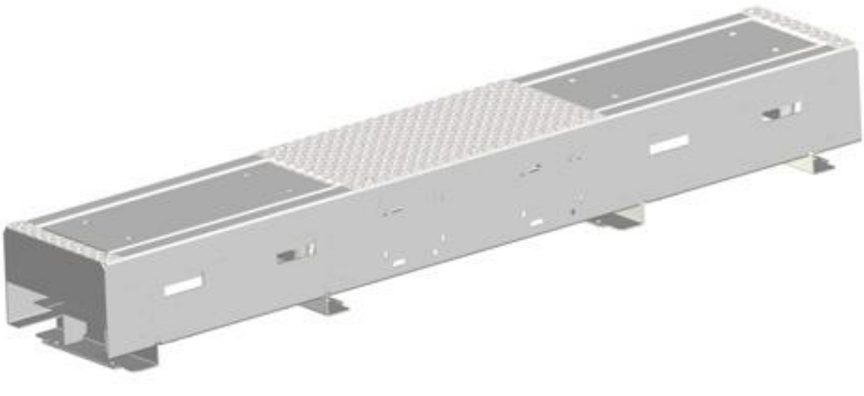


Fig. 4-5 Structure of the adhesion testing rig ST400 [17]

The oscillating platform was also connected to a tensometer that senses the pressure of the tire on it. The measured data are then sent to a computer with a driver, where they will be analyzed.

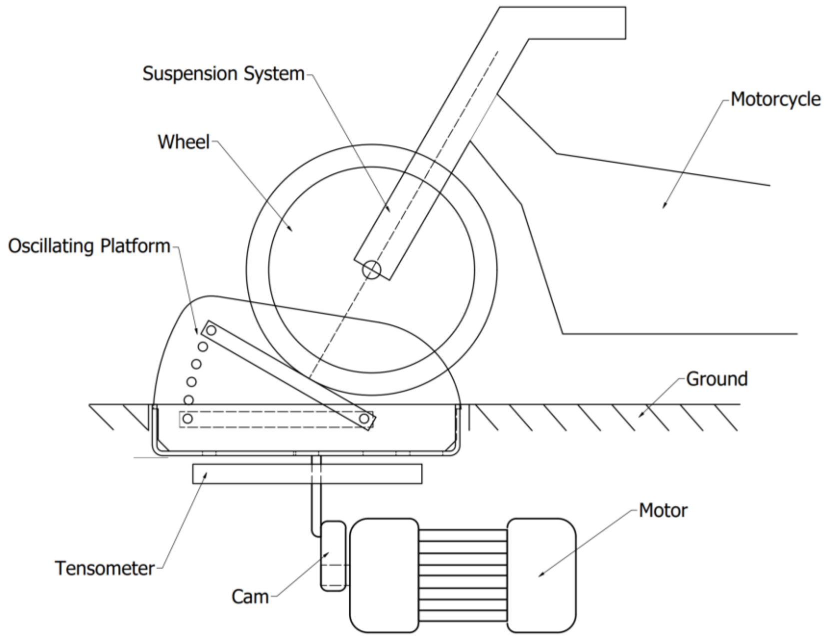


Fig. 4-6 Princip of tester's mechanism of working.

Figure 4-6 shows a scheme that illustrates the main components of the tester.

4.2.2 Dynamical model

The main purpose of the dynamical model is to simulate and predict the behavior of the suspension system when being tested. Moreover, the dynamical model is also used for the sensitivity analysis, where the effect of different parameters on the EUSAMA value will be studied.

In this thesis, Matlab and Simulink were used for creating the dynamic Model. Matlab is a programming language that uses matrices for numerical calculation [20], and Simulink is a graphical modeling program that uses block diagrams to represent the dynamical system [21], like the resonance adhesion tester dynamics.

For simulating the excitation signal of the resonance adhesion tester in Simulink the Chirp signal is used, which provides a sinusoidal wave with a decaying frequency from the initial frequency value to the final value within the target time. See figure 4-7, which displays the chirp block responsible for determining the decaying frequency of the tester, then this block is connected with the multiplier which represents the amplitude of the platform (0.003 m).

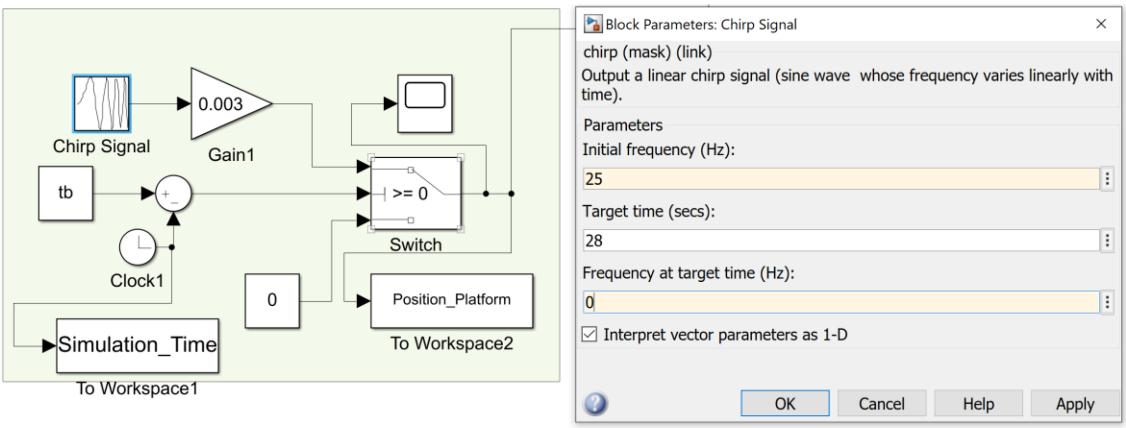
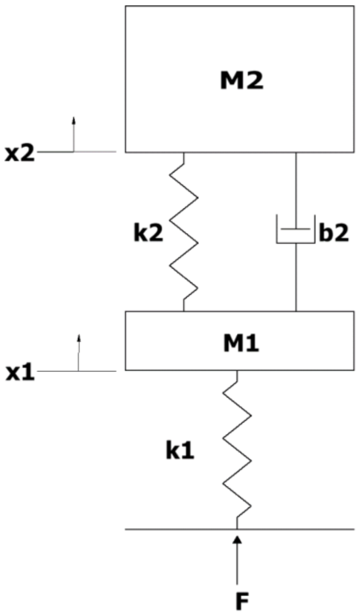


Fig. 4-7 Excitation signal in Simulink.

This input signal simulates the excitation signal used for the real tester in terms of the rig's position, where the rig will be lifted with a displacement of 6mm (Amplitude of $\pm 3\ mm$), and with a decreasing frequency until the rig stops.

For creating the dynamic model, the dynamic equations are then implemented as shown in figure 4-8.



$$m_2 a_2 = k_2(x_1 - x_2) - b_2(\dot{x}_2 - \dot{x}_1) - m_2 g$$

$$m_1 a_1 = k_2(x_2 - x_1) + b_2(\dot{x}_2 - \dot{x}_1) - k_1 x_1 + F$$

Fig. 4-8 Linear 2DOF model.

The most essential blocks used to create the dynamical equations in Matlab Simulink are the following:

- Integrator which integrates a_1 and a_2 into $v_1$and v_2, and in a similar way, $x_1$and x_2 are formed.
- Gain which is used to multiply the time alternating variables (a, v and x) with the constants like k_1, k_2, b_2, m_1, m_2

- To Workspace that sends data to the Matlab workspace which are then extracted in Matlab and used for the numerical calculations.

In the integrators, initial conditions were set with the values of velocities, and initial mass displacements, where these initial conditions are:

$$v_{1\_0} = 0 \tag{25}$$

$$v_{2\_0} = 0 \tag{26}$$

$$x_{1\_0} = -\frac{m_1+m_2}{k_1}g \tag{27}$$

$$x_{2\_0} = -\frac{m_2}{k_2}g + x_{1\_0} \tag{28}$$

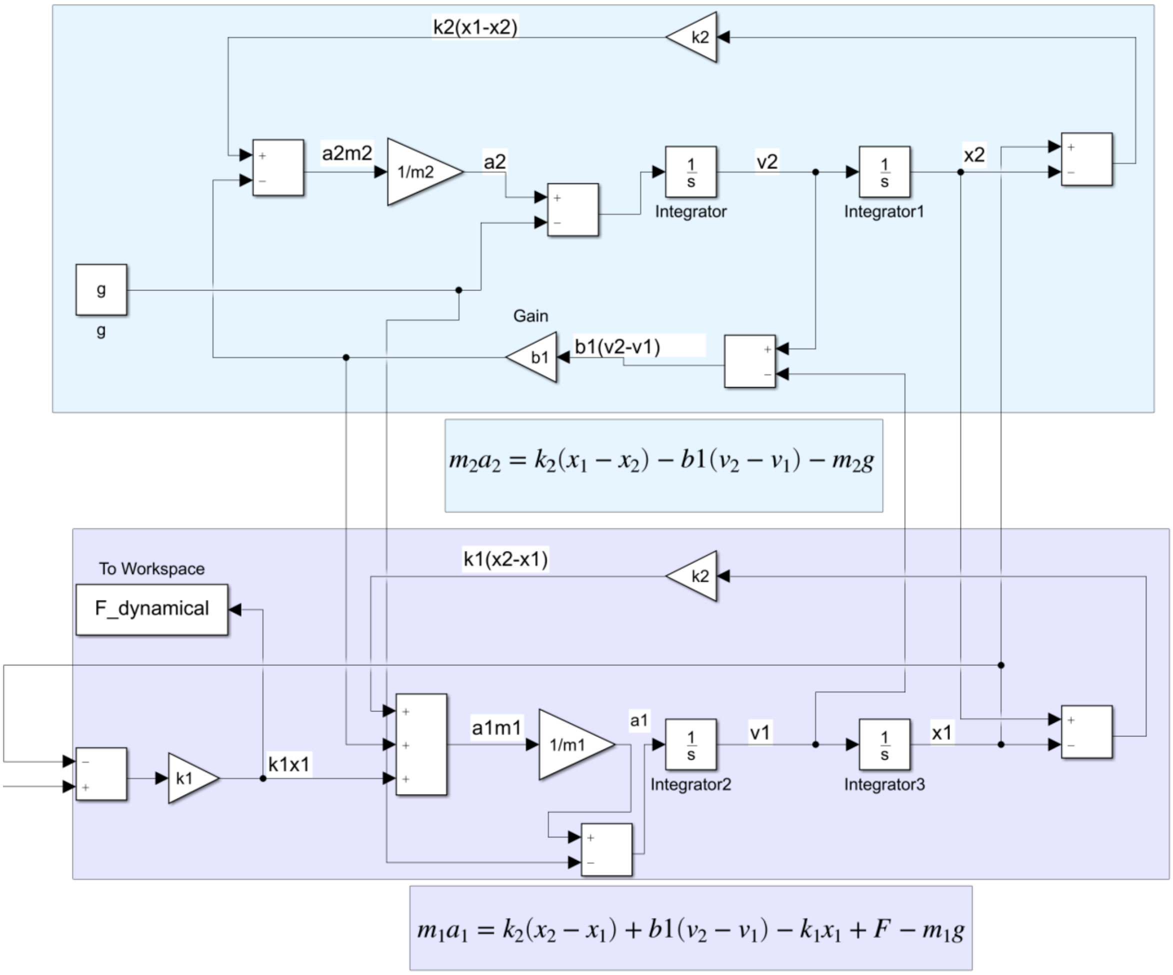


Fig. 4-9 The dynamic model in Simulink.

In that model, the parameters of the Aprilia ETX motorcycle will be used as a reference. These paraments have been previously measured by unmounting the suspension and drawing the F-V and F-Z graphs by the hydraulic tester.

4.3 Material and testing conditions

As stated before, I have used Matlab and Simulink for the simulation model, and for the experiments, I have used the resonance adhesion tester.

The resonance adhesion tester was run by the WinSTR program through which the tester was calibrated and run, and the data were extracted.

The main motorcycle of testing was the Aprilia ETX, for which I have run most of the experiment. This has been the reference motorcycle for the simulated model as most of its parameters are known.

Other tested motorbikes that were used in the experiments are the KTM LC 950, BMW GS 1100, Yamaha dragstar 1100, Yamaha XT 660 SM, and Beta Urban 200.

4.4 Solution method

This work started with testing the Aprilia ETX motorbike with multiple configurations for both back and front suspension. Firstly, to examine the effect of different parameters on the EUSAMA, experiments have been conducted with different riders, different platform angles for the front suspension, different tire stiffnesses, and others. After that, more motorcycles have been tested too, with different but fewer configurations than Aprilia.

Secondly, a simulation model has been created with Matlab and Simulink for sensitivity analysis and to verify the experiments. I started by specifying the input parameters for the sensitivity analysis, where some of which were taken from literature, some were calculated, and others were chosen according to their most reasonable range. Each of these parameters was tested by the simulation model, where I used the range of that specific parameter and the rest of the parameters were fixed with the Aprilia ETX data. Finally, the effect of these parameters on the EUSAMA was compared with data from the experiments

4.5 Testing prediction

The experiments have been run with multiple sets of configurations including a change of the rider, a change of tire stiffness and a change of platform angle, and more. I Expect that these experiments would support the hypotheses the weight of the rider, platform angle, and the tire stiffness would have the greatest effect on the EUSAMA and that a change in these parameters would cause an effect on the accuracy of the EUSAMA value. Moreover, I expect that the tester stabilizing system would also lead to some inaccuracies in the EUSAMA measurements due to its simplicity, as the rider would be struggling to keep the motorcycle in place while testing.

5 RESULTS

5.1 Simulation

5.1.1 Linear model

I have used the linear model to evaluate the Aprilia ETX front and back suspension using the linear model, and I have compared the resultant dynamics, and thus EUSAMA with the dynamics of the experiments.

The results that I got from the experiments are shown in figure 5-1.

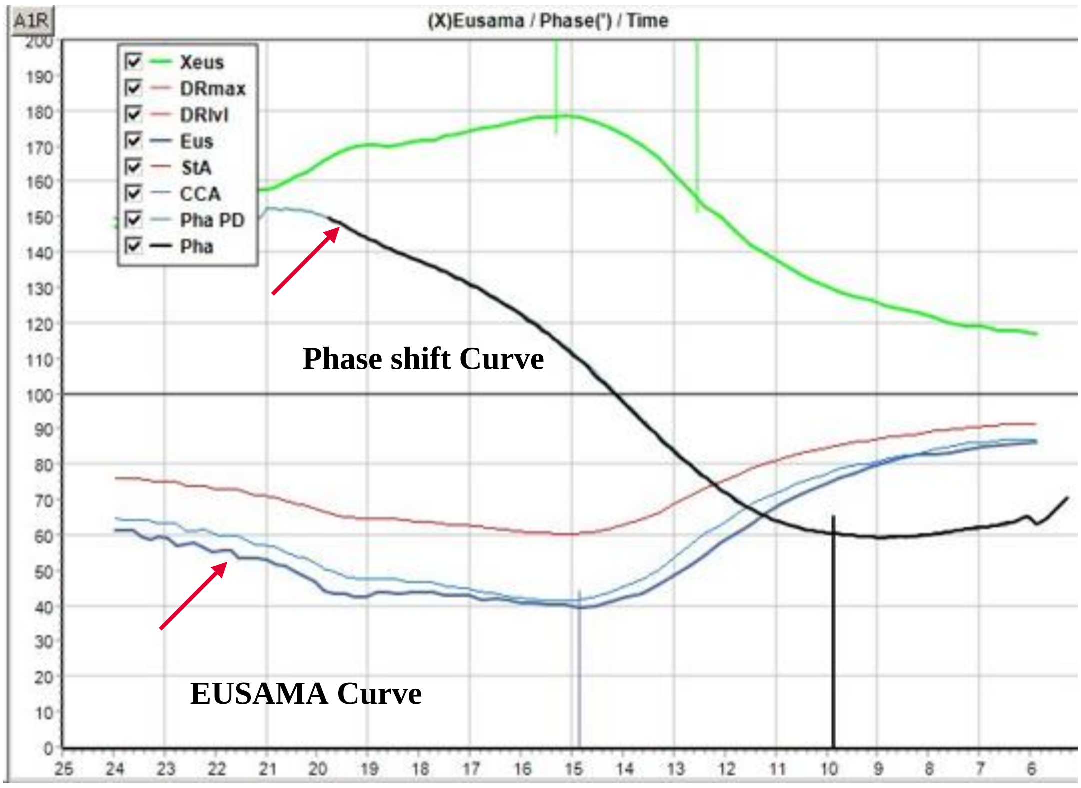


Fig. 5-1 Curves of different criteria of tested motorbike.

The result of the EUSAMA Value that I got from the simulation model is shown in figure 5-2

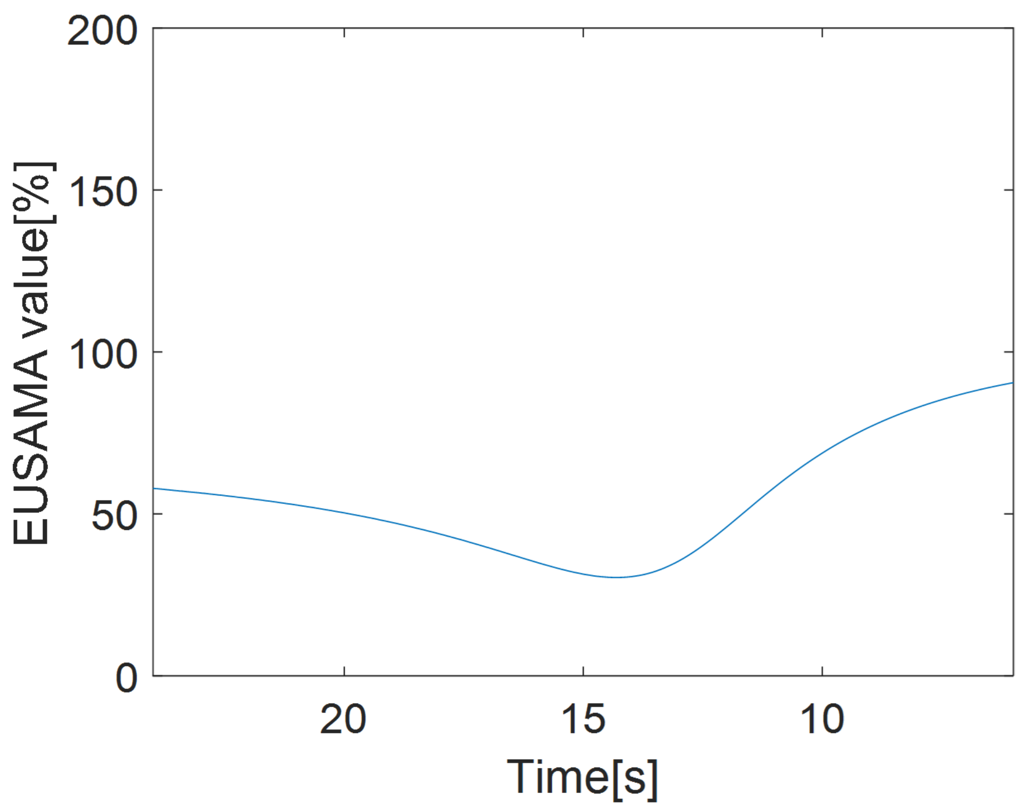


Fig. 5-2 Simulated curve of EUSAMA value for Aprilia ETX.

Comparing the two graphs, it is seen that the two graphs have a similar trend, where the EUSAMA curve starts with a decreasing tendency till it reaches the resonance frequency, after which the tendency starts to increase again. However, as stated in the literature review, the minimum EUSAMA value is the factor that determines the technical state of the suspension. Table 5-1 shows the minimum values of the EUSAMA obtained from the Aprilia ETX motorbike with different weight configurations.

Tab. 5-1 Comparison between the simulation model and experiments in terms of EUSAMA value.

Weight [kg]	EUSAMA Experiment [%]	EUSAMA Simulation [%]	Difference [%]	ResonanceFrequencyy Experiment [Hz]	Resonance frequency Simulation [Hz]
110,3	**44**	**35**	**22,8**	**14,7**	**14,30**
104,9	**43**	**32**	**29,3**	**14,3**	**14,30**
105,7	**42**	**32**	**27**	**14,5**	**14,30**
102,5	**40**	**30**	**28,6**	**14,8**	**14,30**
104,7	**35**	**32**	**8,9**	**14,6**	**14,30**

From table 5-1, it is seen that the resonance frequency is the same for all weight configurations, and that is because the resonance frequency in the model is calculated rather than measured. This can be seen in equation 18 where the resonance is calculated by the suspension stiffness and unsprung mass, but not affected by the variation of sprung mass on which table 5-1 is based on.

5.1.2 Non-Linear model

As explained in the previous chapters that the non-linear model of a motorcycle is a model in which the damper doesn't have a constant damping value which then results in a nonlinear increment of the damping force with the increment of the velocity.

The damping force used in the reference model has been measured in our institute, where I have received raw data, which I have analyzed and taken the outcomes needed for this thesis, including the damping force of the nonlinear trend of the damper. The trend shown in figure 5-3 is an approximated tendency of the measured data of Aprilia ETX front suspension.

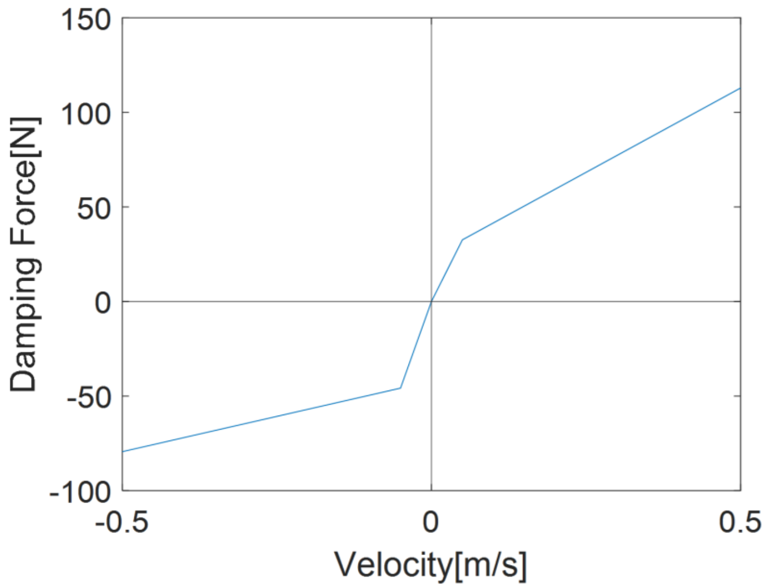


Fig. 5-3 Damping force of Aprilia ETX front wheel.

This approximation includes two different damping constants for the expansion of suspension which result in a negative damping force (the sign indicates the direction of the force), and two other different damping constants that result in positive damping values.

Figure 5-4 shows the cure of the EUSAMA value that I got from the nonlinear simulation model.

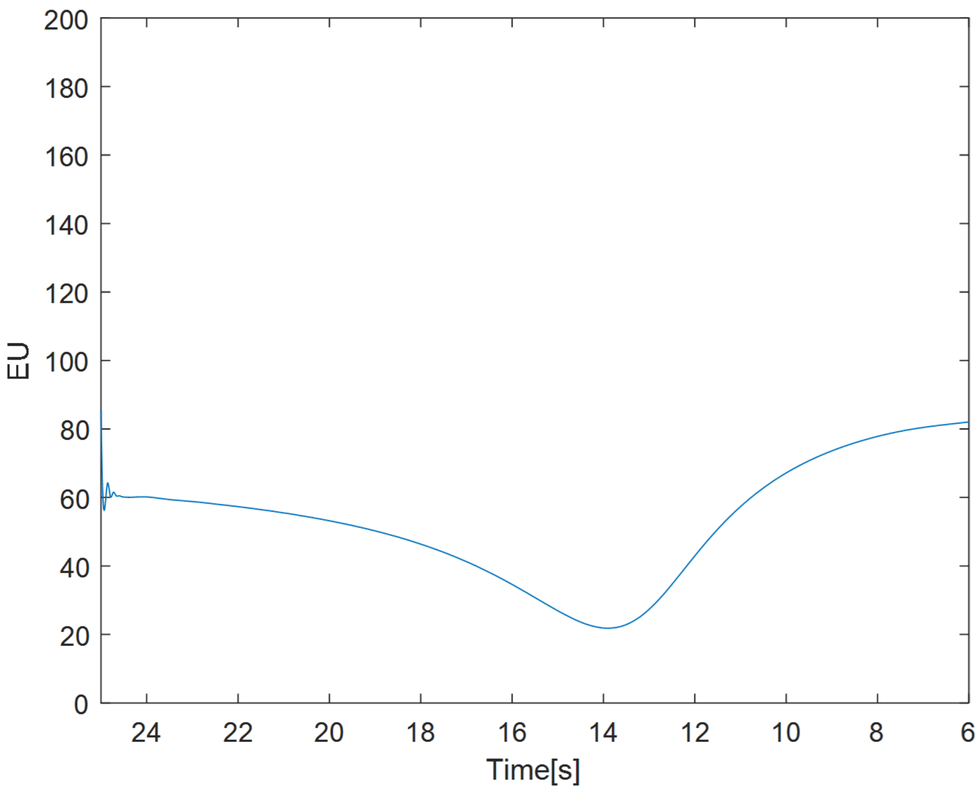


Fig. 5-4 EUSAMA curve of Aprilia front when simulated by a nonlinear model.

Similar to the linear model, comparing the curve of the EUSAMA value of the Experiment shown in figure 5-1 and the simulation shown in figure 5-4, I can say the trend is very similar. Still, the values of the minimum EUSAMA are less similar.

For more illustrations, see table 5-2.

Tab. 5-2 Comparison between the simulation nonlinear model and experiments in terms of EUSAMA value.

Weight [kg]	EUSAMA Experiment [%]	EUSAMA Simulation [%]	Difference [%]	Resonance Frequency Experiment [Hz]	Resonance frequency Simulation [Hz]
110,3	**44**	**27**	**47,9**	**14,7**	**13,90**
104,9	**43**	**23,2**	**59,8**	**13,9**	**13,90**
105,7	**42**	**23,8**	**55,3**	**14,5**	**13,90**
102,5	**40**	**21,5**	**60,1**	**14,8**	**13,90**
104,7	**35**	**23**	**41,4**	**14,6**	**13,90**

From the table, it is seen that the minimum value of the linear model is closer to the real value than the nonlinear model. Therefore, the linear model was used for the sensitivity analysis.

5.2 Sensitivity analysis

The study will be concentrated on studying the effect of different input parameters on the EUSAMA value, and this will be achieved by changing a specific input parameter within a set range and fixing the values of other input parameters.

For studying the effect of parameters on the EUSAMA value, a linear two degrees of freedom model has been used, which was generated by Matlab-Simulink, for the front motorbike suspension of the Aprilia Motorbike.

All parameters implemented in the simulation model are real measurements of the Aprilia front wheel, except for the tire stiffness which hasn't been measured, and therefore, I have chosen 100 kN/m to be used in the simulation model, and it has been selected according to its best fit on the simulation, when comparing the results of the simulation with real values from the experiment.

5.2.1 Effect of weight on EUSAMA Value

Most motorbikes have a weight range of approximately 100-350 kg, and the weight of the rider ranges mostly between 50 (young riders) -250 kg (when considering two riders and luggage),

Considering that the weight will be distributed (50% - 50%) between the front and the back wheel, and because our model is only a 2DOF model, which considers a half of the motorbike, then the weight range will be between 70 kg for light motorbikes and 300 kg for heavy bikes like cruiser motorbikes.

- **Simulation**

For the simulation, the following configurations have been used:

Tab. 5-3 Input data for Studying the weight effect on EUSAMA.

Parameter	Values
Range of sprung mass [kg]	70-300
Unsprung mass [kg]	14,4
Tire stiffness [kN/m]	100
Suspension stiffness [N/m]	3660 (reduced value for whole fork =8910)
Damping ratio	0,18
Caster angle [Degree]	26

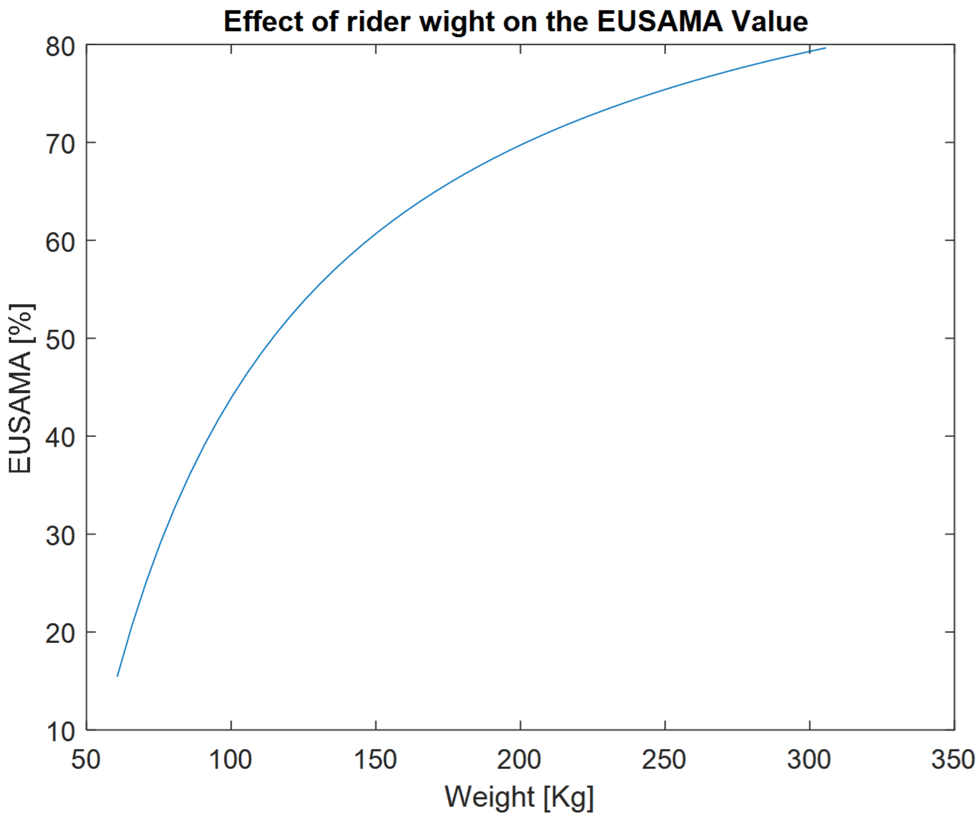


Fig. 5-5 Effect of sprung mass on the EUSAMA.

According to figure 5-5, the EUSAMA value is highly influenced by the weight of the motorcycle/rider, where the ESAMA increases with the increment of the weight, where the EUSAMA value increases from 15 % to 80 % within the set range of sprung mass.

The sprung mass mainly consists of the chassis of the motorcycle and the ridder, so the EUSAMA value will be affected by the weight of the rider, and the heavier the rider is, the higher the EUSAMA will be.

- **Experiments**

This has also been tested experimentally where the EUSAMA value has been measured for motorcycles with the same set of parameters, but with different riders, and table 5-4 shows the effect of different riders on the EUSAMA value.

For the Aprilia ETX, there were two different riders with different weights, where the first weighed 83 kg and the second weighed 95 kg.

The experiments started with the front suspension of the motorbike, where I have performed three different sets of experiments, and for each set of experiments, I have changed the platform angle. In each set of experiments, I have tested the minimum EUSAMA value with both riders multiple times, and the value shown in the table is the average value from these repetitions. After that, I tested the back wheel with the same method, with the difference that the back suspension doesn't have a caster angle, and therefore, the platform angle had only one zero setting.

For the Yamaha XT motorcycle, there were also two riders with different weights, and the with the same method as the Aprilia I have tested the minimum EUSAMA value for the front and back suspension.

Tab. 5-4 Input data for Studying the weight effect on EUSAMA.

Motorcycle	Rider weight [kg]	Wheel	Platform angle [Degree]	EUSAMA [%]	Difference [%]
Aprilia ETX	83	Front	18	35,1	11,4%
	95			39,6	
	83		24	40,6	9,2%
	95			44,7	
	83		30	33,9	15,5%
	95			40,1	
	83	Back	0	27,1	25,0%
	95			36,2	
Yamaha XT	83	Front	24	30	25,9%
	97			40,5	
	83	Back	0	65,6	6,6%
	97			70,2	

In table 5-4, I have also calculated the difference of the minimum EUSAMA value in percentage for different riders, and the difference is not constant as it ranged between 11 % and 25 %, and that is due to multiple factors including the stability of the rider which is going to be discussed in the next chapter. However, the EUSAMA value is highly affected by the weight of the rider, where it increases with the increment of the rider's weight.

5.2.2 Effect of Tire stiffness on EUSAMA Value

As stated in the literature review, the radial stiffness of the tire is the relation between the vertical load on the tire and the deformation that has been caused by this load [1].

For the simulation, the following configurations have been used:

Tab. 5-5 Input data for Studying the tire stiffness effect on EUSAMA.

Parameter	Values
Sprung mass [kg]	103
Unsprung mass [kg]	14,4
Tire stiffness [kN/m]	100-350
Suspension stiffness [N/m]	3660
Damping ratio	0,18
Caster angle [Degree]	26

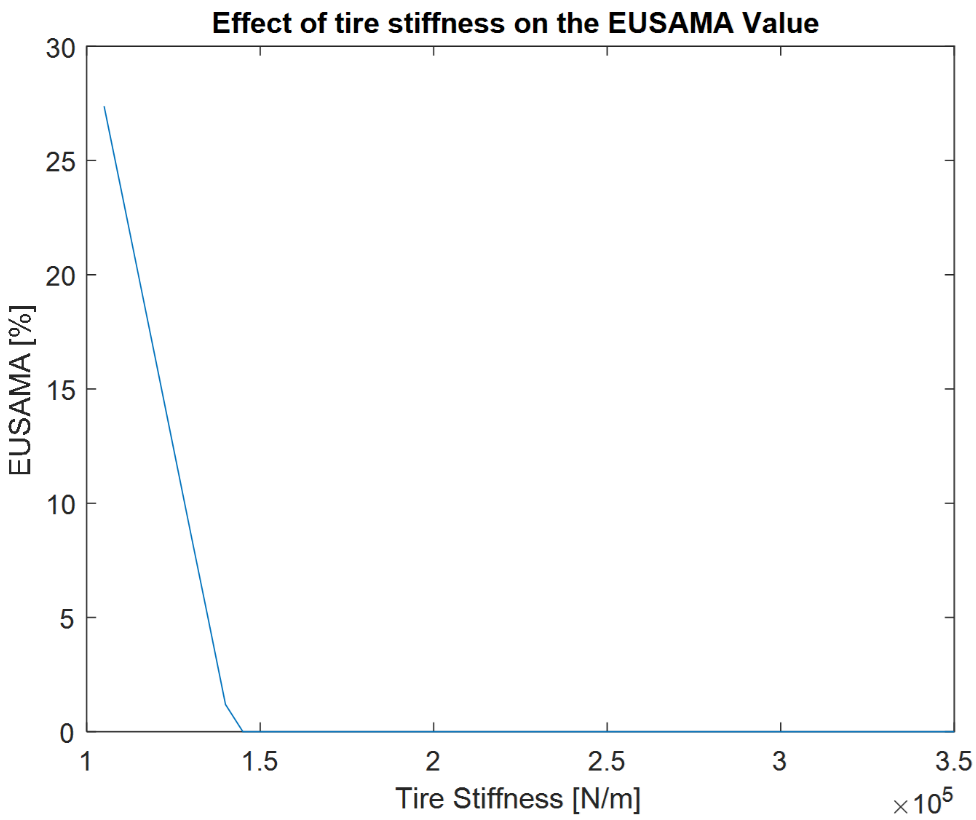


Fig. 5-6 Effect tire stiffness on EUSAMA.

Figure 5-6 shows that the EUSAMA is highly influenced by the stiffness of the tire, where the EUSAMA value decreases with the increment of the tire stiffness. In other words, the high stiffness of the tire decreases the contact of the tire, and thus the EUSAMA value will also be decreased.

As noticed in figure 5-6, in this specific motorbikes configuration, the EUSAMA value drops to zero after increasing the tire stiffness to a specific value, where the motorcycle losses contact with the ground, and thus, no dynamical force will be measured. However, this won't always be the case wherein heavy motorbikes; the weight would greatly increase the dynamical force as seen in the section before.

Here is a demonstration of how heavy motorcycles keep the tires in contact with the ground even if the tire stiffness is too high.

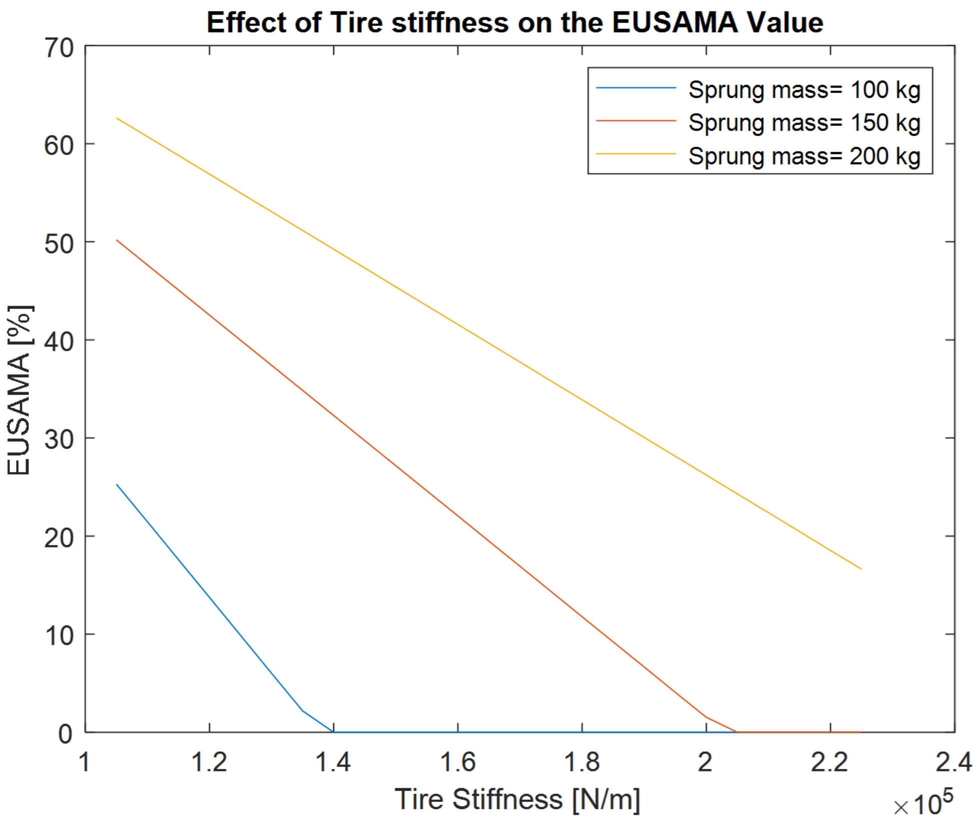


Fig. 5-7 Effect of Weight and tire stiffness on EUSAMA.

I have studied the effect of tire suspension on the EUSAMA value within the set range between 100-350 kN/m. The same range has been tested for three different weight configurations and the results of these three configurations are shown in figurer 5-7. From the figure it is seen that a motorcycle starts to bounce and loses contact with the ground at a specific value of tire stiffness, and a heavier motorbike will hold its contact at the same value of tire stiffness.

5.2.3 Effect of suspension stiffness on EUSAMA Value

It was mentioned in the literature review that the reduced stiffness of suspension is 6-12 times less than tire stiffness, and as seen previously, the tire stiffness value range between 100 and 300 N/m.

For studying the effect of suspension stiffness effect on the EUSAMA, I have considered the biggest range possible, where the lower limit will be 100000/12 [N/m] and the upper limit will be 300000/6 [N/m], which will give a range between 8333 N/m and 50000 N/m.

The resultant suspension stiffness is the effective suspension that I have used in the simulation model, and the resultant suspension is calculated by equation 16 which considers the two spring-damping units in the fork and considers the effect of caster angle. Therefore, to calculate the nominal range of one suspension, the following equation will be used:

$$k = \frac{k_{red} \cdot \cos(alpha)^2}{2}$$

in the simulation model, where I have used the configurations of the Aprilia motorbike whose caster angle is 26 degrees, then the nominal range will be between 3366 N/m and 20196 N/m.

The other parameters that have been used in the simulation model are shown in table 5-6

Tab. 5-6 Input data for Studying the weight effect on EUSAMA.

Parameter	Values
Sprung mass [kg]	103
Unsprung mass [kg]	14,4
Tire stiffness [kN/m]	100
Suspension stiffness [N/m]	3366-20196
Damping ratio	0,18
Caster angle [Degree]	26

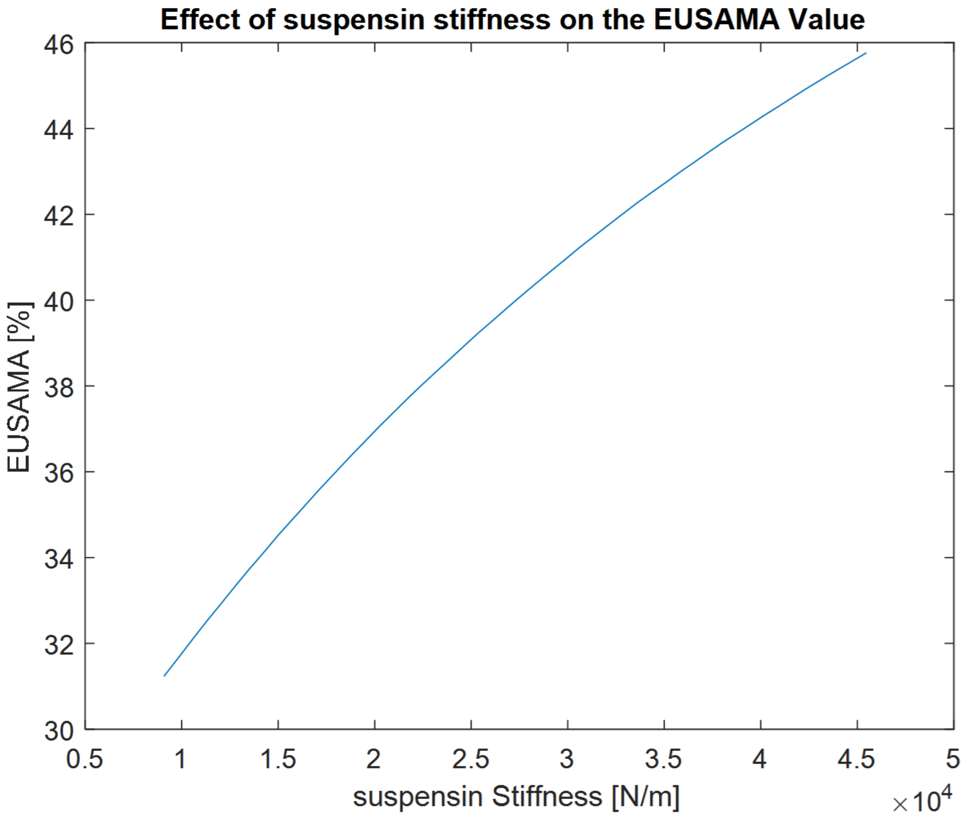


Fig. 5-8 Effect of suspension stiffness on EUSAMA.

In figure 5-8, unlike the tire stiffness, it is seen that the increment of the suspension stiffness affects the EUSAMA positively, where it increases with the rise of the suspension stiffness.

The suspension stiffness affects the dynamical force in a way that it affects the deformation of the tire, where it pushes the tire downwards which increases the tire deformations, or it pulls the tire upwards and thus decreases the tire deformation. This is shown in figure 5-9.

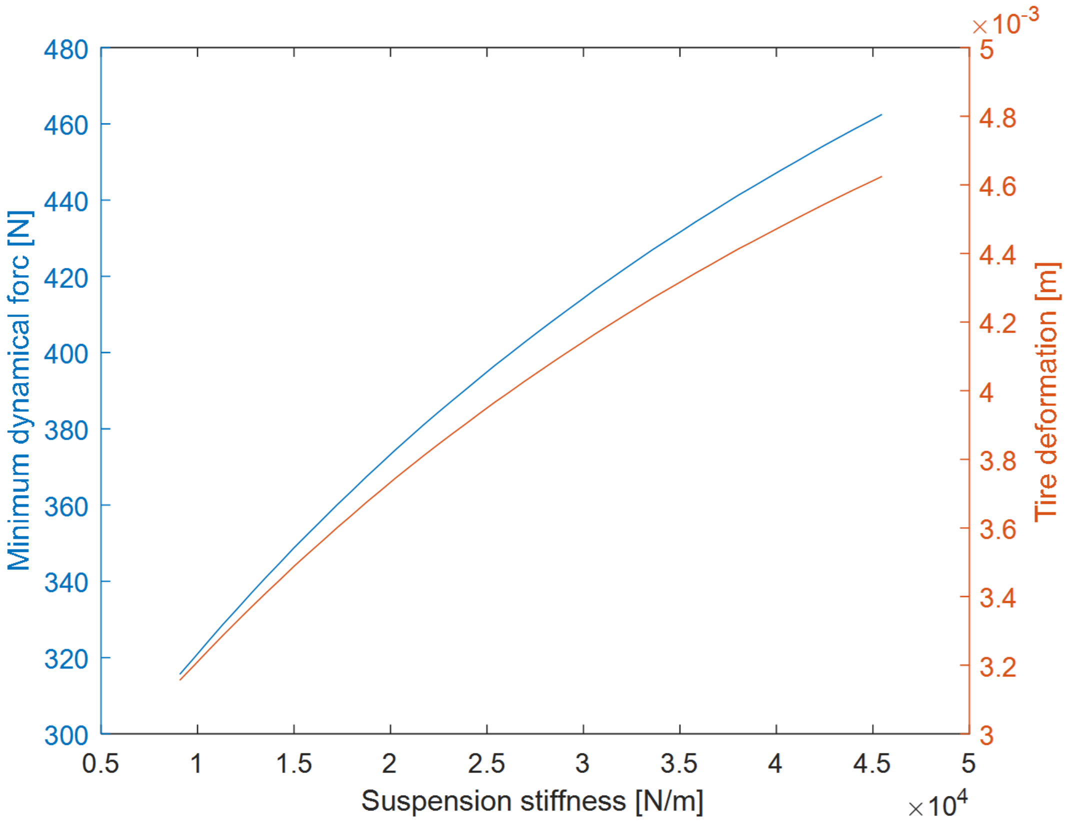


Fig. 5-9 Effect of suspension stiffness on Tire deformation.

From figure 5-8 and 5-9, it can be resulted that the effect of suspension stiffness on the tire deformation, and thus on dynamical force and EUSAMA value is not significant compared to the effect of tire stiffness.

The effect of the suspension stiffness can be very little compared to the effect of tire stiffness on the motorcycle dynamics to the point that it can be neglected with very low frequencies [1], and that is because the tire stiffness is 6-12 times greater than the suspension stiffness. Figure 5-10 shows the effect of tire stiffness and the suspension stiffness of a motorcycle (with a sprung mass of 150 kg) within 50 percent of their ranges, where the x-axis shows a percentage of the range of both stiffnesses and the y-axis shows the EUSAMA value.

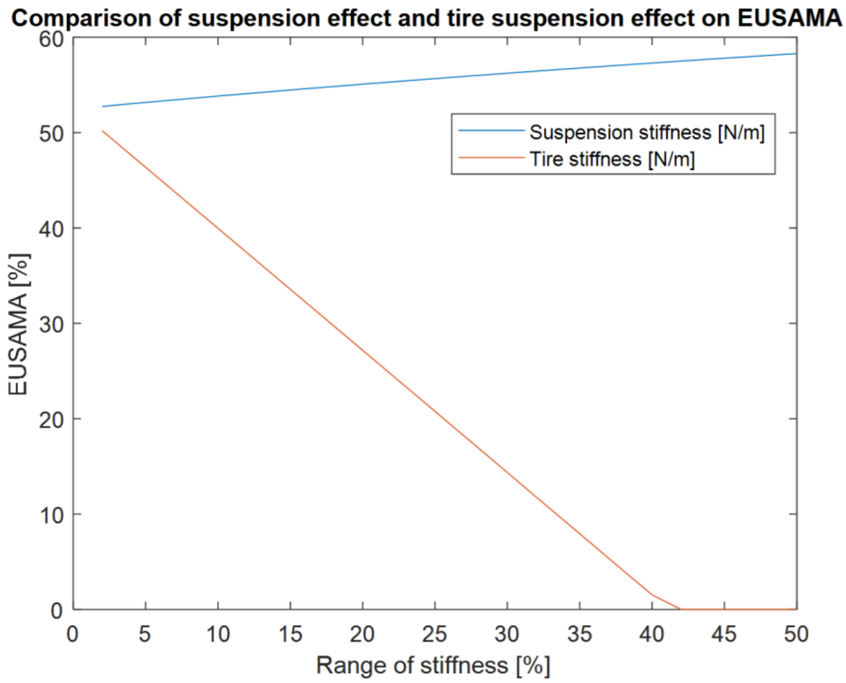


Fig. 5-10 Effect of suspension stiffness on Tire deformation.

5.2.4 Effect of damping constant on EUSAMA

Testing suspensions usually tells the state of the whole suspension system, but dampers can be one of the most vulnerable elements of the suspension system.

For calculating the range of the damping constant I have used the parameters for the Aprilia ETX motorcycle and the damping ratio which has been explained in the literature review and calculated by equations 4 and 6.

The damping ratio has a range between 0 and 1, and because the damping ratio is the ratio between the damping constant and the critical damping of the motorcycle, then the range of damping constant of any motorcycle will be between zero, and the critical damping which can be calculated using equation 6.

In the Aprilia ETX motorcycle, the calculated range of the damping constant was approximately 0-2500 N/(m/s), and this represents the range of both spring-damping units with considering the caster angle as the damping ratio is calculated for the whole fork.

For the simulation, I have used the calculated range with the rest of the Aprilia front suspension parameters listed in table 5-7, to examine the effect of the damping constant on the EUSAMA.

Tab. 5-7 Input data for studying the damping constant effect on EUSAMA.

Parameter	Values
Sprung mass [kg]	103
Unsprung mass [kg]	14,4
Tire stiffness [kN/m]	100
Suspension stiffness [N/m]	3366
Damping ratio	0-1
Damping constant [N/(m/s)]	0-2500
Caster angle [Degree]	26

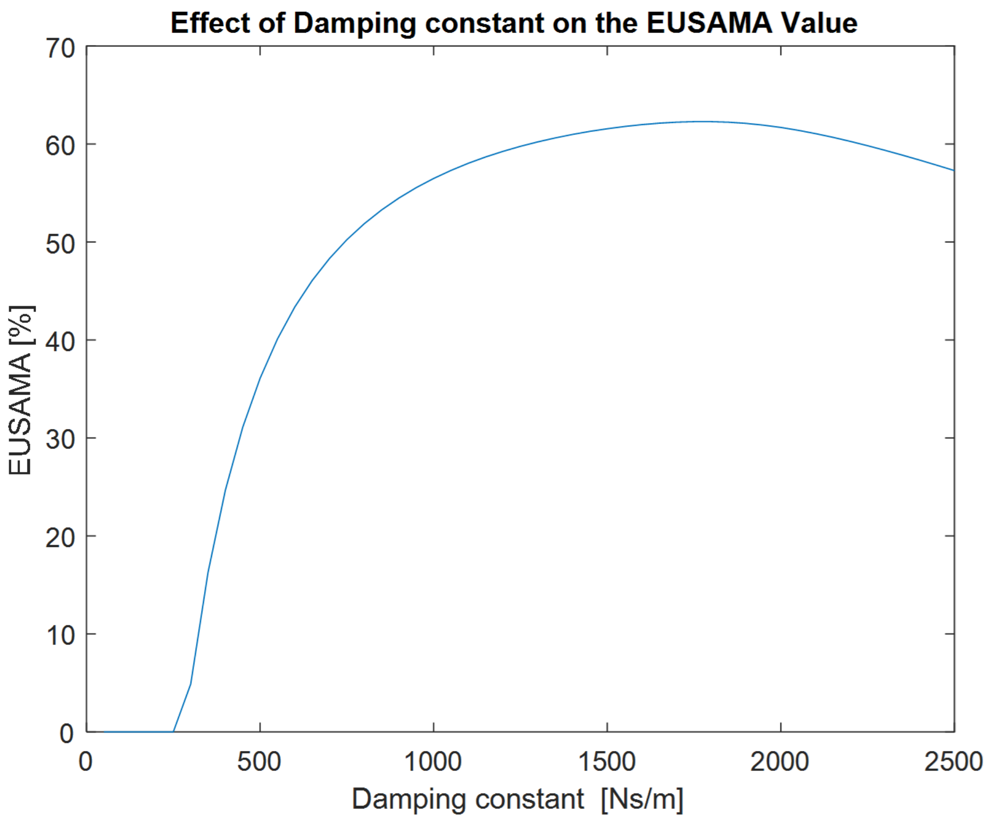


Fig. 5-11 Effect of damping constant on EUSAMA.

In figure 5-11, it is clear that the EUSAMA value increases with the increment of the damping constant until it reaches a specific value (peak) after which the EUSAMA value starts to decrease with the increment of the damping constant, and When the EUSAMA value reaches the peak, it means that the damping constant is equal to the critical damping constant

If the damping ratio is less than one, it means that the damper is underdamped where it will oscillate until it returns to its initial position, but if the damping ratio is more than one, then the damper is overdamped which means that the damper will be very stiff and will not oscillate, but rather go back slowly to its initial position without oscillations.

The damping ratio reaches one when the damping constant is equal to the critical damping; however, in the simulated model, the EUSAMA value started to decrease at 1750 Ns/m (reduced damping=2166.3 Ns/m). The damping ratio at the maximum EUSAMA value was 0,9 and that is because the simulation model is just an approximation with inaccuracies.

5.2.5 Effect of plattform angle on EUSAMA

In the literature review, it was noted that the caster angle could range between 19-24 degrees in sport motorcycles; however, in some extreme cases and with heavy motorcycles, the caster angle can be up to 34 in touring motorcycles.

For the purpose of the thesis, I chose the range between 19 and 30 for the simulation model because the platform that is used in the experiments ranges between 0-30 degrees.

The caster angle affects directly the suspension stiffness and the damping constants where the values that I have used in the model are the reduced values of suspension stiffness and therefore, by using the simulation, I will test the effect of the platform angle on both of these parameters and thus on the dynamic force and EUSAMA.

Firstly, I will examine the relationship between the suspension stiffness and the platform angle. The parameters used in the simulation are listed in table 5-8

Tab. 5-8 Input data for Studying the platform angle and suspension stiffness effect on EUSAMA.

Parameter	Values
Sprung mass [kg]	103
Unsprung mass [kg]	14,4
Tire stiffness [kN/m]	100
Suspension stiffness [N/m]	3366-20196
Damping ratio	0,18
Platform angle [Degree]	19-30

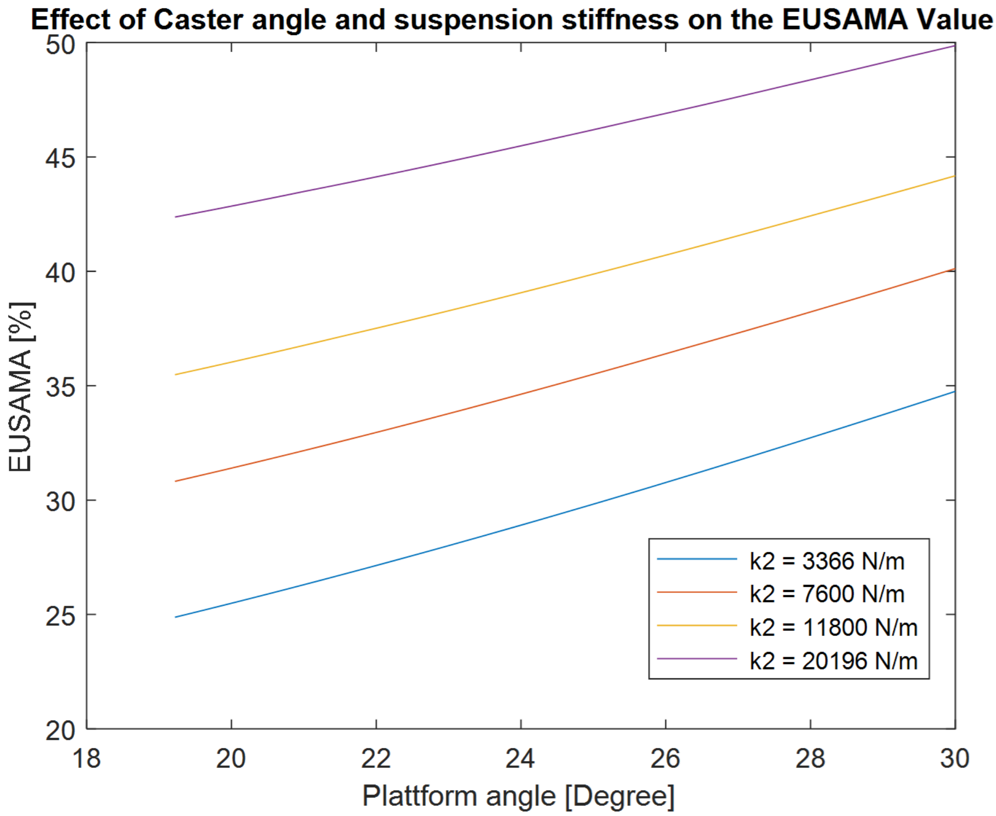


Fig. 5-12 Effect of platform angle and the suspension stiffness on the EUUSAMA.

Figure 5-12 shows how the change of platform angle and suspension stiffness affect the EUSAMA value, where I have applied the range of the platform angle on each of the four suspension stiffness configurations and reviewed their effect on the EUSAMA, noting that the suspension stiffnesses on figure 5-12 represent only one half of the fork. Therefore, the reduced I have considered the effect of caster angle and multiplied by two.

From figure 5-12 I can say that the effect of the platform angle on the suspension stiffness is notable but not significant, and the difference of EUSAMA within the range of platform angle is almost constant because the trend of the platform is almost linear and the increment of the suspension shifts the trend upward due to the semi-linear trend. For more illustration, see table 5-9.

Tab. 5-9 Range of EUSAMA within the range of platform angle for different suspension stiffness settings.

Suspension Stiffness [N/m]	Range of EUSAMA [%]	Difference [%]
3366 x 2	25-35	10
7600 x 2	31-30	9
11800 x 2	35-44	9
20196 x 2	42-50	8

For studying the effect of platform angle and damping constant on the EUSAMA, I have used the parameters listed in table 5-10

Tab. 5-10 Input data for Studying the effect of platform angle and damping constant on EUSAMA.

Parameter	Values
Sprung mass [kg]	103
Unsprung mass [kg]	14,4
Tire stiffness [kN/m]	100
Suspension stiffness [N/m]	3366
Damping ratio	0-1
Damping constant [N/(m/s)]	0-2500
Plattform angle [Degree]	19-30

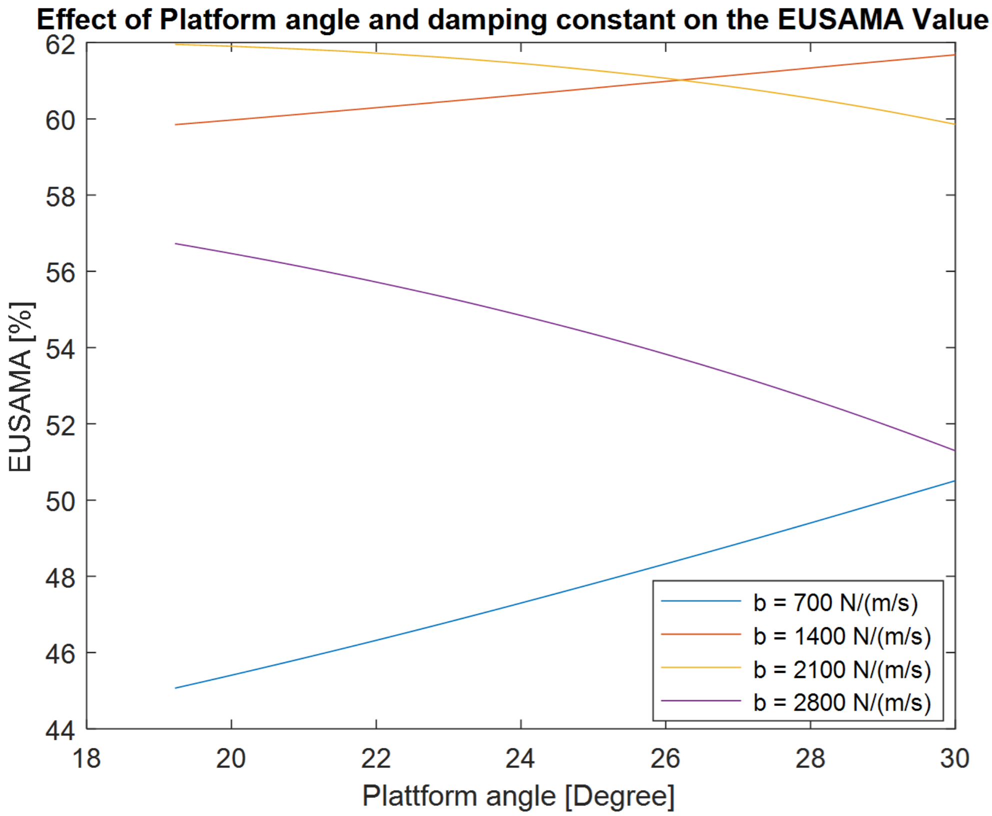


Fig. 5-13 Effect of platform angle and damping constant on the EUUSAMA.

Figures 5-13 and 5-11, where figure 5-11 showed the relation between the damping constant and EUSAMA, and it was seen that after the EUSAMA curve increases, it starts to decrease when the damping constant reaches the critical damping. This is the same thing that happens in figure 5-13, where the increment of the platform angle increases the EUSAMA value at b=700 N/(m/s) and 1400 N/(m/s), but decreases at b = 2100 N/(m/s) and 2800 N/(m/s) because when considering the reduced value of the damping constant, the 2100 and 2800 N/(m/s) within the range of platform angle will be more than the critical damping constant.

At b=2100 N/(m/s) and by using equation 17, the reduced damping constant at 19° platform angle is 2350 N/(m/s), and at 30° platform angle b= 2800 N/(m/s) which is greater than the critical damping, which means that effect of platform angle is significant enough to be considered in this work.

5.2.6 Effect of unsprung mass on EUSAMA

For identifying the range of the unsprung mass, I have used the range of the resonance frequency of the unsprung mass specified in reference [1], which is between 12-18 Hz, and the resonance frequency is calculated using the following equation (29):

$$f = \frac{1}{2\pi} \cdot \sqrt{\frac{tire\ stiffness\ (k1)}{unsprung\ mass(m1)}} \tag{29}$$

Therefore, for calculating the range of unsprung mass, I have used the range of tire stiffness, the frequency of 18 Hz, and equation 29, and the resultant range was between 8 – 30 kg, which is a reasonable range for unsprung masses.

Other parameters used for studying the effect of unsprung mass on EUSAMA are listed in table 5-11.

Tab. 5-11 Input data for Studying the effect unsprung mass on EUSAMA.

Parameter	Values
Sprung mass [kg]	103
Unsprung mass [kg]	8-30
Tire stiffness [kN/m]	100
Suspension stiffness [N/m]	3366
Damping ratio	0,18
Plattform angle [Degree]	19-30

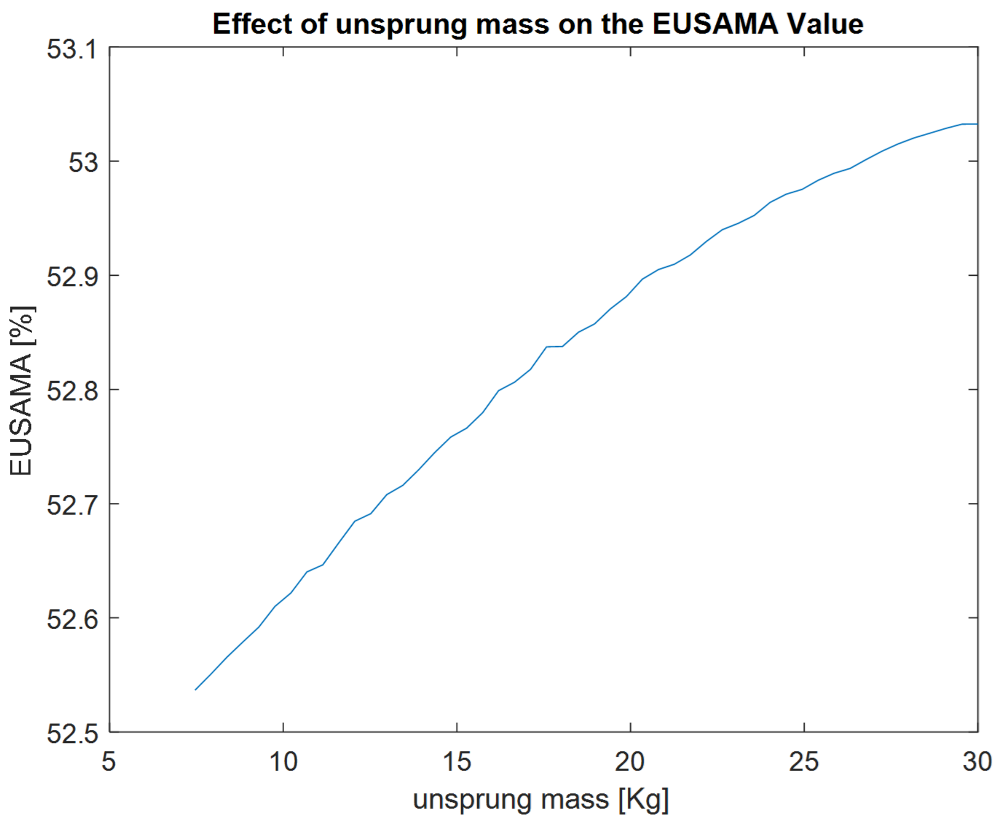


Fig. 5-14 Effect of unsprung mass on the EUUSAMA.

From figure 5-14, it is seen that within the set range, the unsprung mass has a very mild effect on EUSAMA compared to other studied parameters.

5.2.7 Summary of the parameters that affect the EUSAMA

This section is a comparison of 5 different parameters that affect the EUSAMA, excluding the damping constant, and that is because the damper is mostly the main element in the suspension that is being tested.

Each of the 5 parameters has been studied individually by varying the value of the studied parameter within its range and fixing the other parameters with the values shown in table 5-12

Tab. 5-12 Input data for Studying the effect of different parameters on EUSAMA.

Parameter	Range	Fixed Value
Sprung mass [kg]	70 - 350	150
Unsprung mass [kg]	7 - 30	14,4
Tire stiffness [kN/m]	100 - 350	100
Suspension stiffness [N/m]	3366-20196	3660
Plattform angle [degree]	19 - 30	26

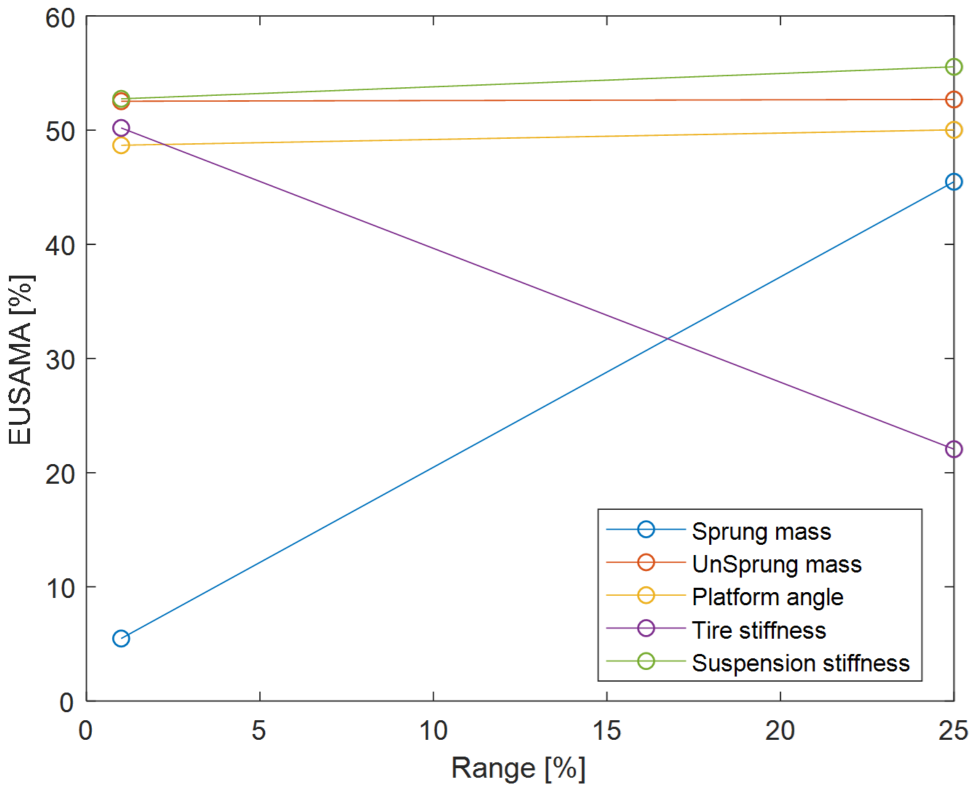


Fig. 5-15 Effect of different parameters on the EUUSAMA.

Figure 5-15 shows all the parameters that affect the EUSAMA within 25 % of their ranges (first quarter of their ranges), and the 25 % has been set because some of the changes of some parameters like the tire stiffness lead the EUSAMA to decrease to zero after a specific value, and some other parameters increase the Eusama and then decrease it. Therefore, the effect of only 25% has been considered to compare all the parameters equally where the EUSAMA either increases or decreases without having most of the range interval at zero EUSAMA.

The effects in figure 5-15 show the linear increment or decrement regardless of the curve of the effect of these parameters, and the curves that represent the change of the EUSAMA within the ranges of these parameters have been provided previously in this chapter.

From Figure 5-15, it is seen that the weight and the tire stiffness are the most significant parameters that affect the EUSAMA; however, the platform angle also might have a great effect on the EUSAMA depending on the value of the damping constant. Moreover, in figure 5-15, the platform angle seems not to have a great effect on the EUSAMA because 25% of its range is between 19° and 22,8° which is quite low. Besides, there are other important uses of the platform angle, which will be discussed in the next chapter.

6 DISCUSSION

6.1.1 Interpretation of results

The non-existence of a non-invasive testing method for motorbike suspension is the main problem that this thesis is helping to solve by verifying if the resonance adhesion tester that is based on the EUSAMA methodology could be expanded from car suspension testing only to motorcycle suspension testing as well. It started with experimental testing and a simulation model creation through which I have tested the main parameters that affect the EUSAMA the most. From the results, it was shown that the parameters that affect the EUSAMA value the most are the weight of the rider and motorcycle, the tire stiffness and the platform angle.

- **Weight effect**

As seen in the results, the weight of the rider highly affects the results of EUSAMA and therefore, the results of testing. So when testing the suspension of motorcycles and the person that performs the testing is light in weight, the evaluation of the suspension will be different from the evaluation when the person performing on the tester is heavier in weight.

The resonance adhesion tester is mainly designed for cars, and the weight is not an issue when testing because the weight of the car is not changed as the testing procedure does not require a rider while testing. However, this is not the case when testing motorcycles, as the testing performance requires a person riding the bike to stabilize it.

To solve this problem, a mechanism for weight settings can be used, like the one used in the road simulator mentioned in reference [3], in which the weight can be fixed to a specific value or changed as required. This solution can be quite complex for an easy tester designed to be in service stations and used only for the purpose of suspension assessment.

Therefore, I suggest that the easiest way is to set the criteria of testing, testing conditions, and calibrations according to a reference weight value so that when testing, a serviceman should be chosen to perform the testing whose weight is close enough to the reference weight value that has been set before.

Another factor that could affect the weight, which will thus affect the EUSAMA value, is the position of the rider on the motorcycle. To demonstrate, if the rider is leaning to the front while testing, this would increase the weight on the front side of the motorcycle, due to the fact that the weight is not going to be distributed evenly between the front and the bach weel (shift of the center of mass). On the other hand, if the rider leans more to the back, the weight on the front will be decreased, and thus the EUSAMA value will be measured inaccurately less. For that reason, a seating position should be set while testing where the test performer should keep his back straight and vertical to the seat of the motorcycle in order to ensure an even distribution of weight and thus a more accurate assessment of the suspension.

It was expected that there would be problems during testing regarding stability, and therefore stabilization device was designed. I have verified its functionality, and it was proven that the assumption about stability problems was reasonable due to its effect on the rider's position on the motorcycle and the position of his legs during the test, as would be seen in table 6-1.

Besides the previously mentioned factors, one of the most significant elements that affects the EUSAMA value during testing is the stability of the rider. If the rider is unstable and the motorcycle is fluctuating to the sides, the measurements of the weight are not going to be accurate, and thus the EUSAMA values. In other words, when the rider tries to stabilize the motorcycle, he will exert some force on the handles pressing the motorcycle down which will then affect the assessment of the motorcycle. In addition, during the stabilization, it is highly likely that the rider will lean to the front or the back causing inaccurate measurements of the weight and EUSAMA values.

Another issue that is related to the stability of the motorcycle during testing is the position of the rider’s legs if they are placed on the pedals or the ground. In other words, if the rider has his legs on the pedals, the stability of the motorcycle will be ensured by only the holding mechanism explained in chapter 4. The holding mechanism used in the experiments of this thesis is very simple and doesn’t provide enough stability.

When the rider keeps his legs on the ground, the tested motorcycle will be more stable, but the weight will be affected. To clarify, if the test performer has his legs on the ground, then part of his weight will not be measured because his weight is not fully placed on the motorcycle. This could also cause a decrease in the EUSAMA value because when the legs of the rider are on the ground, the motorcycle will not have a free motion when going upwards during testing. After all, the body of the rider will act as a resistor that opposes the motion of the motorcycle.

This effect has also been tested experimentally, where several experiments have been conducted on the Aprilia motorcycle with two different riders and with different platform angle settings. Table 6-1 shows the difference in the EUSAMA value for each platform angle setting and with different leg positions.

Tab. 6-1 Effect of legs position on the EUSAMA value.

Wheel	Rider weight [kg]	Caster angle [%]	leg position	EUSAMA [%]	Difference [%]
Front	95	18	Pedal	39,3	5,8%
			Ground	37	
		24	Pedal	44,7	5,1%
			Ground	42,4	
		30	Pedal	38,7	3,9%
			Ground	37,2	
	83	18	Pedal	35,1	9,1%
			Ground	31,9	
		24	Pedal	40,6	2,7%
			Ground	39,5	
		30	Pedal	33,9	11,2%
			Ground	30,1	
Back	95	0	Pedal	36,2	6,1%
			Ground	34	
	83	0	Pedal	27,1	7,80%
			Ground	29,4	

The difference in the EUSAMA value can reach up to 11 %, and therefore, ensuring stability by keeping the legs on the ground can lead to an inaccurate assessment of motorcycles' suspension. For that reason, a holding mechanism should be constructed to ensure stability without affecting the EUSAMA value.

- **Tire stiffness effect**

It was proven from the results that the EUSAMA can be affected by a slight change in the tire stiffness, and because the stiffness of the tire and the tire pressure are very dependent on each other, then the tire pressure has to be taken into consideration while testing. To clarify, when the tire pressure is low, then the tire stiffness will be decreased causing the tire to deform more easily. This would increase the contact force of the tire with the ground and thus increase the dynamical force and the EUSAMA value.

The relation between the EUSAMA and the tire pressure has also been tested experimentally, where the Yamaha XT back wheel has been tested with two different tire pressures (0,75 bar and 1,9 bar), and the result is shown in table 6-2, where the EUSAMA value decreased noticeably when increased the tire pressure from 0,75 to 1,9 bar.

Tab. 6-2 Effect of tire pressure on the EUSAMA value.

Tire Pressure [bar]	EUSAMA [%]	Difference [%]
0,75	65,5	26,10%
1,9	48,4	

In table 6-2, it is seen that a change of the tire pressure would result in a change of the EUSAMA, and therefore, the ideal tire pressure that is specified in the motorcycle's catalog has to be considered during testing. The tire pressure on the catalog has been set in accordance with the weight of the motorcycle, the type of motorcycle, the size and the material of the tire, and other configurations such as the damping behavior.

To conclude, the test performer should measure the tire pressure before testing and set it according to the reference value specified on the motorcycle catalog in order to have a more precise measurement of the EUSAMA value, and thus a better assessment of the suspension.

- **Platform angle effect**

As seen from the results that the platform angle can have a very great effect on the damping which affects the precision of the assessment of the suspension, and therefore it is very important to use the modified platform demonstrated in the previous chapters in order to have the right assessment of the suspension.

Using the modified platform reduces the stress and strain that acts on the suspension unit during testing. To illustrate, the tires of motorcycles don't move in a vertical way like cars, but rather along a diagonal line that alines with the steering axis which is inclined from the vertical line by the caster angle. So, if the original flat platform (which is used for cars) of the resonance adhesion tester was used, it will act on the spring-damper unit of the motorcycle with a vertical force that will then cause a lot of strain on the suspension units. This might not cause lots of problems at first, but with periodic testing, it might cause failure.

Before testing, the platform angle must be changed according to the caster angle. If the platform angle is not the same as the caster angle (or close enough), the reading of the dynamic model and thus the EUSAMA value would be affected. To clarify, if the platform angle was less than the caster angle, this would not decrease the resultant damping constant and thus the EUSAMA value. Besides, if the platform angle was less or more than the caster angle, this will act as a resistance to the movement of the suspension, and compression/tension of the suspension unit will not be as smooth as if the steering axis was vertically aligned with the testing rig.

This was also tested experimentally, where I have tested the Aprilia motorcycle with fixed parameters but different platform angles, and the results are shown in table 6-3.

Tab. 6-3 Effect of platform angle on the EUSAMA value.

Platform angle [deg]	Avrage EUSAMA [%]
18	38
24	43
30	38

From the table, it is seen that when the platform angle is close to the caster angle (26 °), the EUSAMA value is greater than the values when the platform angle is larger or smaller than the caster angle.

- **Dynamic model**

The dynamic that I created for this thesis did not give exactly match the real tester, so it should be more complex to mimic reality more. This can be done by firstly measuring the real tire stiffness of the front tire instead of choosing the value, as done in this thesis. In addition, when specifying the damping constants of the non-linear model, it was very difficult to specify the knees at which the damping changes trend, and this would also reduce the precision of the model.

To conclude, Only three parameters represented by the weight, tire stiffness, and the platform angle were the only parameters chosen to be discussed due to their major effect on the EUSAMA. The damping constant was excluded because the damper is mostly the element that is being tested, due to its complexity and susceptibility to failure. The unsprung mass and surprisingly the suspension stiffness had a mild effect on the EUSAMA, which were neglected multiple times in different models and calculations in reference [1]. For future work, the simulation model has to be improved, and more reference motorcycles should be used other than the Aprilia ETX. Additionally, more experiments need to be conducted with the addition of a stabilizing equipment and with respect to the testing conditions that have been introduced.

6.1.2 Verification of the hypothesis

This thesis helped in specifying the most important conditions of testing that need to be met and the changes that should be done to get more precise data of the EUSAMA. So, as hypothesized, it was observed that the measured weight affected the EUSAMA, where the EUSAMA increased with the increment of the rider, and if the increment of the weight was caused by the instability of the rider or his wrong position during the test, the increment of the EUSAMA would lead to a wrong assessment of the suspension. Moreover, this thesis verified the assumptions about the tire stiffness effect on the EUSAMA, where the increment of the tire stiffness affects the EUSAMA by decreasing it, so if the tire inflation was not set according to the testing conditions, the stiffness of the tire will be affected and thus the EUSAMA. In addition, the effect of the platform angle on the EUSAMA was also confirmed, and it was shown that if the platform angle was not adjusted according to the testing regulations, the movement of the spring-damping unit would be influenced and consequently, the EUSAMA.

7 CONCLUSION

This work investigated the suitability of using the EUSAMA methodology to assess the technical state of motorcycle suspensions besides the car suspension which this methodology was designed for. This investigation included experimental verifications and simulation verification. Both methods contributed to specifying the parameters that affect the EUSAMA value the most during testing, how they would impact the EUSAMA value, and how to prevent their negative effects on the testing results.

The first parameter that has a great impact on the EUSAMA is the rider's weight. When using the resonance adhesion tester with the EUSAMA methodology for testing motorcycle suspension, the weight of the rider plays a great role in determining the value of EUSAMA because a slight change in the rider’s weight will be accompanied with a change in the EUSAMA value. So if two riders with different weights are performing the suspension test for the same motorbike, there will be two different values of EUSAMA. Furthermore, the position of the rider could have an impact on the EUSAMA value, so if the rider is leaning forward during the test, the measured weight and thus EUSAMA value will increase, and if he is leaning to the back, The EUSAMA will decrease. Moreover, the instability of the motorcycle when testing is also a big influence on the measured weight and thus the EUSAMA. So if the motorcycle is not stable, the rider would exert some force by stabilizing the motorcycle which would be accompanied by leaning forward or backward, or by placing his legs on the ground instead of pedals, and all of this contributes to wrongly measuring the weight, and thus the dynamic force and EUSAMA as explained before. All of the mentioned problems don’t occur when car suspension testing because it doesn't require a rider to perform the test, and cars don’t need to be stabilized while testing.

Due to the difference in the design of motorcycles and cars, these problems occur with motorcycle suspension testing, and to avoid their negative effect on motorcycle suspension assessment, several precautions have to be taken into consideration. Firstly, a more complex stabilizing construction has to be made in order to prevent the shakiness of the motorcycle and the rider during testing. Moreover, all the testing conditions and calibrations have to be set according to a reference weight, and the serviceman performing the testing has to be chosen according to that reference weight. Additionally, the serviceman performing the testing has to respect some positioning conditions where he has to keep his back straight and legs on the padel for the aim of eliminating the weight factors that cause the miscalculations of the EUSAMA values.

Tire stiffness is also one of the parameters that impacts the EUSAMA value and might lead to miscalculating the EUSAMA value. Tire stiffness is determined by many factors that include the material and dimensions of the tire, but tire inflation is one of the factors that affects tire stiffness the most. To simplify, when the tire is overinflated, it will be very stiff, and its stiffness will be high, which will lead the tire to deform less and be more bouncy during testing. On contrary, if the tire was underinflated, then the deformation of the tire during testing would be bigger, and this would increase the contact force and thus EUSAMA. Therefore, for an accurate assessment of the EUSAMA, the tire should not be overinflated nor underinflated, and this is achieved by controlling the pressure of the tire before the tester, comparing it with the ideal tire pressure set for the motorcycle in its catalog, and adjust that if needed by inflating or deflating the tire.

The platform angle is also an important factor for an accurate assessment of the suspension. The effect of the platform angle differs according to the value of the damping constant, but the platform angle is also very important to prevent any damage to the suspension unit during testing. To clarify, unlike cars' tires, motorcycles' tires move diagonally (instead of vertically) along the steering axis inclined by the caster angle. Therefore, if the testing platform is a flat plate that pushes the motorcycle's tire to move vertically, it will create stress on the suspension unit of the motorcycle and would limit its movement causing a negative effect on the EUSAMA value. For that reason, the modified platform has to be used when testing the front suspension, and the right angle needs to be set according to the caster angle of the motorcycle.

In conclusion, the usage of the EUSAMA methodology for motorcycles is still new, and further work in this field should be done for a better investigation. Future work should include refining the simulation model and making it more complex to mimic reality more. Furthermore, it will be needed to measure the missing data from the Aprilia Etx motorcycle like the front tire stiffness, which will be used in the simulation model to compare the simulated and experimented data. In addition, in this thesis, the Aprilia ETX was the only reference motorcycle that verifies the simulation model and the experiments, and therefore, future works should consider using more reference motorcycles, starting by measuring their parameters and using them to verify the simulation model which could be used for further studies. Finally, more experiments should be conducted considering the mentioned changes and the testing conditions to verify their efficiency in eliminating the unwanted effects on the EUSAMA.

8 RESEARCH RESULTS ACCORDING TO RIV

Factors that affect the EUSAMA methodology when being used for assessing motorcycle suspension systems. 2022. Czech republic: Brno University of Technology, 5.

Abstract: The resonance adhesion tester is a non-invasive easy tester that is used to assess the functionality of car suspension, and the EUSAMA is one of the most significant methodologies that this tester is based on. The problem is that there is no easy and simple tester used for checking the technical state of motorcycle suspension, and therefore this thesis investigated the applicability of using the EUSAMA methodology for motorcycle suspension assessment by specifying the parameters that affect the EUSAMA value the most and suggesting guidelines to prevent their negative effects on the outcomes of the test.

The article was submitted for review.

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10 LIST OF ABBREVIATIONS, SYMBOLS AND QUANTITIES

10.1 Abbreviations

EUSAMA	European Shock Absorber Manufacturers Association
EUS	EUSAMA Value
DAQ	Digital data acquisition
F-V	Force velocity characteristics
F-Z	Force stroke characteristics

10.2 Physical quantities

F_{dyn_min}	Minimum Dtnamic force
F_{sta}	Static Force
$m_{1,2}$	Sprung or unsprung mass
ζ	Damping ratio
$b_{critical}$	Critical damping constant
k_{equ}	Equivelant suspension stiffness
$k_{1,2}$	Tire and suspension stiffness
F_s	Force exerted by spring
Δy	deformation caused by preload
v	Velocity
$b_{c,e}$	Damping constant at compresion and expansion
ε	Caster angle
k_f	Reduced stiffness of suspension spring
f_{of} / f_{or}	Resonance frequency of front and rear unsprung mass

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12 LIST OF TABLES

13 LIST OF ATTACHMENTS

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2. Non-linear simulation model/Simulink front suspension Aprilia
3. Non-linear simulation model/Simulink back suspension Aprilia
4. Linear simulation model/Simulink front suspension Aprilia for sensitivity Analysis
5. Data analysis from experiments
6. Raw data for Aprilia Motorcycle