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CONSTRUCTION OF A TEST BENCH FOR THE DEVELOPMENT OF EXPERIMENTAL METHODS FOR THE REPRODUCTION OF ROAD INDUCED VIBRATIONS DURING INDOOR CYCLING

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INDEX

ABSTRACT

The development of a test bench for bicycles allows to carry out tests on the behavior and response to vibrations, allows to analyze the behavior of components and accessories and to characterize their influence in relation to perceived comfort.

The purpose of the construction of a test bench is to allow the execution of repeatable tests and in controlled conditions and environment, bypassing the typical problems related to outdoor tests.

The test bench also allows, through the use of a special dummy (HYBRID III dummy), to disconnect the analysis from the influence of the human variable, especially from the anthropometric and aptitude point of view from the biomechanical point of view, even if there is currently the limit of his inability to reproduce the pedal stroke.

First aim of this thesis is the main characteristics and necessary functionalities that the test bench must have in relation to different possible types (concept), choose the one that best suits the case in question, completely define its structure, create its components and assemble them, verifying their operation. All with the aim of overcoming the limits highlighted by the previous test bench.

External vibrations are reproduced through a random load function and defined through the parameters and equations corresponding to the road types defined by the ISO 8608: 2016 standard which characterizes them from the point of view of wavelength, RMS, displacement and spectral density.

Comfort is an important parameter correlated to bicycle usability. It depends deeply on vibrations and human perception. The most of vibrations is generated by the interaction between road and wheels. These vibrations are transmitted to the cyclist causing bad feelings at the bottom, the hands and the foot.

For this reason, it was decided to test the performance of a specific product (shockstop - produced by Redshift) for bicycle handlebars which allows to reduce the amplitude of accelerations transmitted to the hands through a shock-absorbing handlebar stem that can be adjusted in the intensity of damping. Through his analysis from the point of view of the behavior towards different types of roads, we will define his performances highlighting the main aspects.

1 BASIC THEORY OF VIBRATIONS

In the following chapter, the main concepts and fundamental theory of both mathematics and signal analysis will be summarized and will be adopted for the analysis and processing of the results from accelerometers.

1.1 FOURIER TRANSFORMATION

Fourier Transform is a linear function used to analyse a signal in the frequency domain. In this way a complex time-based signal can be converted into its constituent frequencies. The correlation between input signal and output vibration is described by differential equations, but if the analysis pass from time to the frequency domain, the correlation is studied throw algebraic equations. Fourier Transform as follow (1.1):

$$F\big(f(t)\big) = \int_{-\infty}^{+\infty} e^{-i\omega t}\, f(t)\, dt \tag{1.1}$$

The Inverse Fourier Transform allow to come back to time domain integrating a function in frequency domain and is described below (1.2):

$$f(t) = \frac{1}{2\pi}\int_{-\infty}^{+\infty} e^{i\omega t}\, F(\omega)\, dt \tag{1.2}$$

Fourier Transform must be used with continuous analytical functions. Discrete function like signals obtained from sampling or from sensors (discrete signals stored in array) must be used Fast Fourier Transform (*FFT*) or Discrete Fourier transformation (DFT).

1.2 PSD: POWER SPECTRAL DENSITY

The acquired signal is a time-based array, containing those variables that are studied. Fast Fourier Transform function is used to shift domain from time to frequency, obtaining a complex array composed by amplitude and phase.
The frequency domain of the array is from -f/2 to f/2 where f is the sampling frequency. Often it's enough to plot and study only the single-sided spectrum due to the symmetry of the problem but to do this in this case, amplitude has to be double.
Frequency is set from 0 to f/2 and to analyse different transformed signals, it's useful to compare these arrays in terms of modulus (called spectrum), that evidences distribution of signal content along frequency. This representation can also be used to find and analyse the problem of resonance.

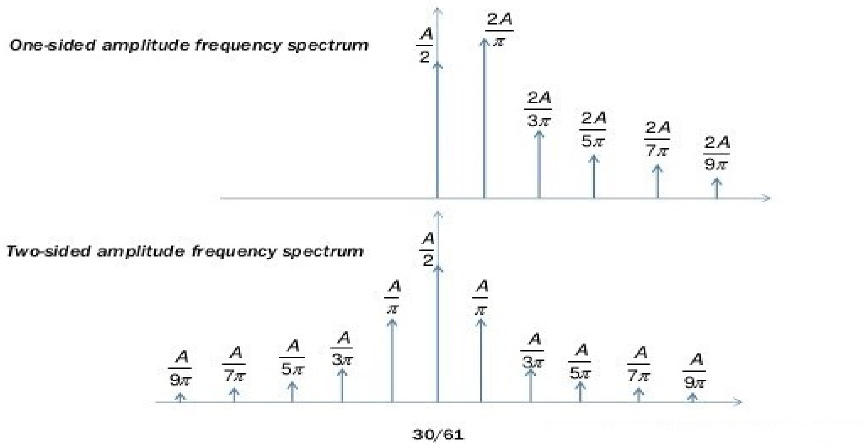

Figure 1.1: Spectrum of a casual periodical rectangular wave.
Differences from single and double-side spectrum.

Another way to analyse a vibration is to use the Power Spectral Density (*PSD*). The PSD value is $A^2/4$, where A is the amplitude of each frequency components. This parameter, describing the correlation between the signal and its own energy [1], it's one of the most used spectra. There are different way to calculated PSD of a given signal: in this thesis *Welch* method is used. In particular Hamming window and 50% overlap between contiguous sections are used to reduce the problem of leakage and to clean as much as possible the signal.
Moreover, after having obtained the PSD of a time-based array, RMS (Root Mean Squared) is calculated in a given range of frequencies in order to compare signals.[2] Another way to compare PSD of two arrays is to use the mean Error Index EI: this index is computed as the mean value of Error Index array, that is calculated as the difference between PSD of the two given signals.

$$EI = PSD_{SIGNAL\ 1} - PSD_{SIGNAL\ 2} \tag{1.3}$$

$$MEAN\ EI = \log_{10}(mean(|PSD_{SIGNAL\ 1} - PSD_{SIGNAL\ 2}|) * 10^4) \tag{1.4}$$

1.3 TRANSFER FUNCTON

To characterize the response of a mechanical system under external forces it is useful to define a Transfer function between input and output of the system. These outputs and inputs can be acceleration, speed or displacement arrays. Transfer Function is a complex function defined in frequency domain and it's the ratio between Fourier transform of output signal and Fourier transform of the input one and it describes the behaviour of the system.

This function can be plotted throw two diagrams. The first one represents the magnitude of output-input ratio in the frequency domain; the second diagram shows the phase in relation to frequency.

Phase represents the lateness of the output with respect to the input signal. Transfer function is defined by the following equation (1.5).

$$H(f) = \frac{Y(f)}{X(f)} \tag{1.5}$$

H(f) is the Transfer Function; X(f) is the Fast Fourier Transform of the input signal; Y(f) is the Fast Fourier Transform of the output signal.

Transfer Function of a system can be defined with a frequency sweep of sine input and measuring displacements (or accelerations) of input and output during the whole sweep. These signals must be transformed in frequency domain and their ratio gives the Transfer Function, according to equation *1.6.*

$$H(\omega) = \frac{F(B\sin(\omega t+\varphi f))}{F(A\sin(\omega t))} \tag{1.6}$$

$$f = \frac{2\pi}{\omega} \tag{1.7}$$

A and *B* are respectively amplitudes of input and output signals, ω is the angular frequency that is the changing variable of the sweep, and φ is the phase between input and output signals. To obtain Transfer Function in time frequency (in *Hertz*) *Equation 1.7* can be used.As said before output and input are complex arrays then Transfer function is a complex function with real and imaginary components. Equation 1.8 represent the function; but module and phase con be represented separatly with equations 1.9 and 1.10.

$$H(f) = R(f) + \boldsymbol{i}I(f) \tag{1.8}$$

$$|H(f)| = \sqrt{R^2(f) + I^2(f)} \tag{1.9}$$

$$\varphi\,(f) = \operatorname{atan}\left(\frac{I(f)}{R(f)}\right) \tag{1.10}$$

Transfer Function's Module gives information about the behaviour of the system reguarding the amplitude of the output respect to the input one for each frequency considered. Phase inform about the value of the lateness of the output signal respect to input one.

Fourier Transform and Transfer Function let to solve several complex easly thanks to an algebraic approach rather than a differential one. By Fourier Transforms, Transfer function can be calculated in order to describe the system dynamic behaviour. This flowchart is represented in the following figure 1.2.

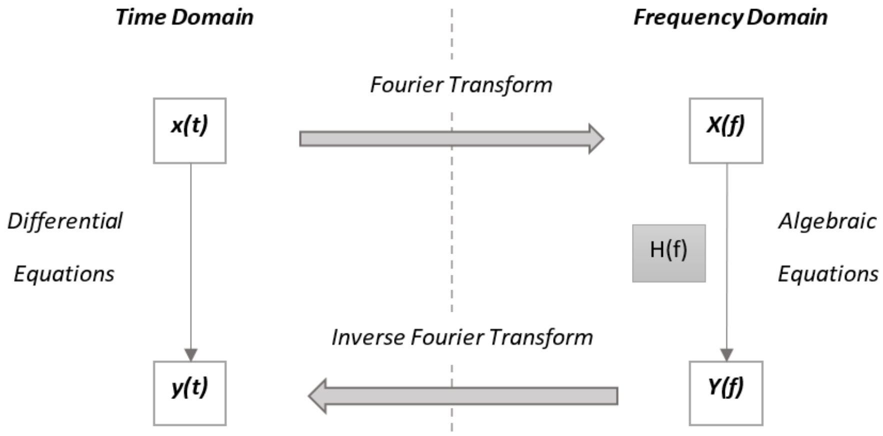

Figure 1.2: Scheme used to analyse the dynamic behaviour of a system.

2 MATERIALS AND INTRUMENTATIONS

The following chapter describes all the elements and equipment used (hardware and software) for carrying out the tests.

2.1 LIST OF MATERIALS AND INSTRUMENTATIONS

1 New Roller shaker

2 MTS® system test bench to reproduce road vibrations;

3 SOMAT EDAQLITE® acquisition system used with accelerometers and *HBM SoMat InField*

4 Accelerometers for the evaluation of Power Spectral Density (PSD) and transfer function (H)

5 Basement

6 Safety harness

7 Personal computer

8 Rhinoceros 5 ® - software 3D CAD for the design of some components ot the test bench;

9 Matlab

10 Microsoft Excel

11 Bike (Battaglin)

2.2 PREVIOUS ROLLER SHAKER AND BENCH

There is the possibility to use two different roller shaker. One is usable for bycicle (smaller) and another (widther) can be used for handbikes and wheelchairs.

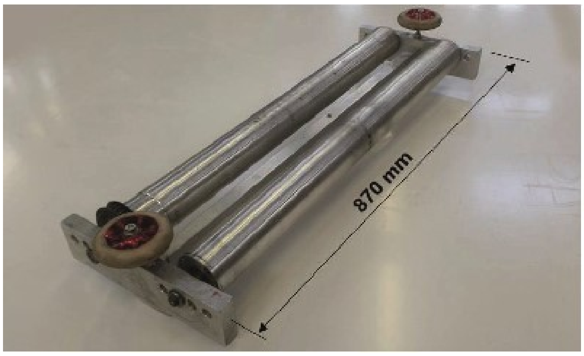

Figure 2.1: Wider roller shaker

Figure 2.2: Narrower roller shaker

2.3 MTS® SERVOHYDRAULIC CYLINDERS TO REPRODUCE ROAD VIBRATIONS

The MTS test bench used is composed by two main parts identificable as software and hardware:

Hardware: Hardware is composed by the following main system:

- Power alimentation panel (on/off electric current – set low and high level of pressure – safety button)
- hydraulic pump (power unit that provide to guarantee high pressure and oil flow)
- Hydraulic control panel (Fig 2.4)
- Two MTS® servo-hydraulic actuators (Fig2.5) The hydraulic actuators used in this study are made by *MTS System Corporation®*, with a maximum force of *14.7 kN*, frequency range 0-100 Hz, an effective area of *7.60 cm^2*, a static stroke of *209.6 mm* and a dynamic stroke of *203.2 mm.* Each actuator is guided by a *Moog®* valve for its control.
- MTS® servo-valves (Fig.2.3)

Fig.2.3 MTS® servo-valves

Fig.2.4 Control panel

Fig.2.5 Servo-hydraulic actuators

Software: Softwere part of the system is accessibile at the last level usign it’s interface. Using MTS Station Manager (Fig. 2.6) is possible to set all parameters to control each servo-hydraulic unit. This application allows to create the file set-up for carrying out the tests with a wide range of settings and use different control modes. Test’s parameters such as the type of function (sine curve, square) wave, random etc...) or the actuator control (force- displacement), PID (Fig. 4.6) and limits can be setup using the dialog window.

The interface of the software with main communication and control windows is showed in *Figure (2.6)* and it’s composed by 5 dialog windows.

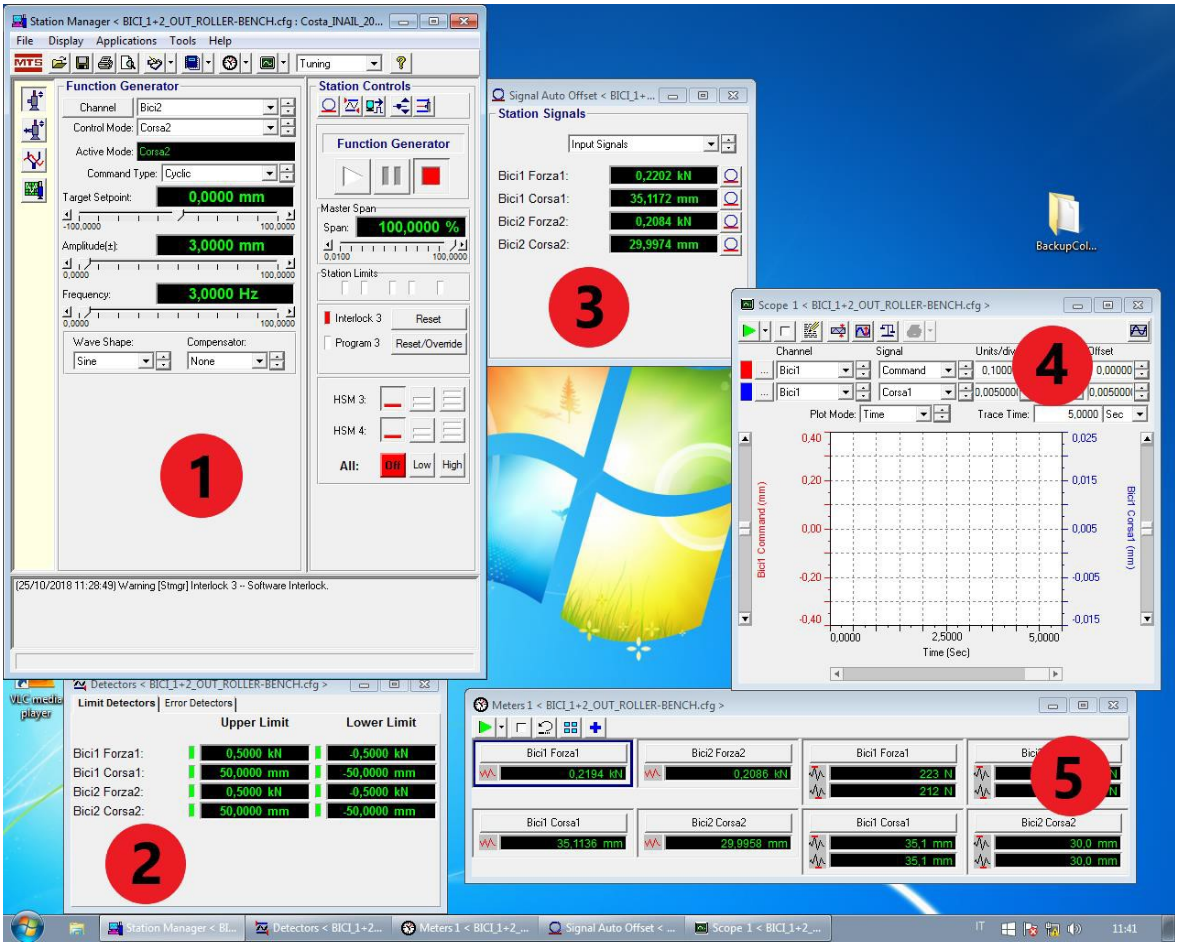

Figure2.6 :MTS Smart Station main interface.

1: *Command* window: to set the input desired as the type of function, channels, frequencies, displacement amplitude.

2: *Limits* window: to define upper and lower limits of cylinders displacement and force (for test and safety reasons)

3: *Offset* window: to set new zero points of stroke and force. For example for the definition of a zero position for the roller shaker.

4: *Scope* window: to plot different parameters as actuator force or stroke in real time; it's useful to compare applied force and output force to adjust the ability to follow the imput curve using PID control.

5: *Meters* window: to view instantaneous values and peaks of forces and strokes

Control modes: To allow the control of hydraulic actuators is used the *MTS Smart Station®*, that lets to use several different control modes. For the thesis two ways of control are used: *MTS Function Generator®* and *MTS MultiPurpose TestWare®*.

MTS Function Generator®: Allow the complete control ot the two servo-idraulic actuators (separately or coupled). With the function generator is possible to generate a digital function and control and correct the ability of the real sistem to follow the imposed signal (PID control). The feedback signal (force) is generated by the load cell placed at the end of the stem.

Functions tipe ar divided in three main groups:

- *Cyclic functions:* Sine, Square, ramp waves with constant amplitude and frequency;
- *Random function:* Random signal based on range of frequency and Root Mean Square;
- *Sweep function:* Sine waves with fixed amplitude and increasing frequency.

MTS MultiPurpose TestWare® allow control actuators using procedures that is a way to define the imput signals.
Using the *Profile Command* is possible to import a text and define a field of motion (stroke profile) that each actuators must follow (istant by istant).
This text file must be written with a syntax that explains which actuator to move, the wave shape, the type of control, the level in stroke to be reached, the frequency of changing between two levels.[3]

In the following figure (2.7), an example of *txt* file is shown: in the third row there is bicycle mean speed that is used to generate these displacements; below this, number of channels is presented. Here MTS software has to control two channels (the two actuators). After this, txt file starts to control the first hydraulic cylinder (S1) with a ramp function. Last information are about the level in stroke to be reached and the frequency of changing between two consecutive levels (50 Hz)

```
FileType=Phase
Date= 04-Dec-2019
Description= input profile 17 km/h
Channels=2

Channel(1)= S1
Shape=Ramp

Frequency Level1
Hz mm
1 -2.05E-01
50 -4.26E-01
50 -2.99E-01
50 -1.27E-01
50 1.68E-01
50 3.18E-01
50 3.43E-01
50 9.73E-02
50 -5.37E-01
50 -1.42E+00
50 -2.79E+00
50 -3.97E+00
50 -1.98E+00
50 3.30E+00
```

Figure 2.7: Example of txt file, written using a Matlab procedure to import a profile in MTS® system.

For each test is possible to export and save data measured instantaneously in *Meters* window (forces measured, strokes, and command profiles).

2.4 SOMAT EDAQLITE®

HBM SoMat eDAQlite® lets to acquire analog and digital signals and data from sensors with a sample rates until *100 kHz*[4]
The device is composed by different acquisition modules called layers (Each layer has four pins). There is the possibility to connect bridges strain gauges (acronym BRG) and other sensor, for example accelerometers or potentiometers (acronym HLS). In this study they were used accelerometers.

Figure 2.8: Datalogger SoMat eDAQlite® [5]

SoMat Test Control Environment® (*TCE*) is used to configure this system. The software lets to choose and setup the channel of each device, depending on its the properties. It's possible for example to set the calibration constant of accelerometers.
The link between the PC and the devices was made with an Ethernet wire. The maximum sampling frequencies is of 100 kHz

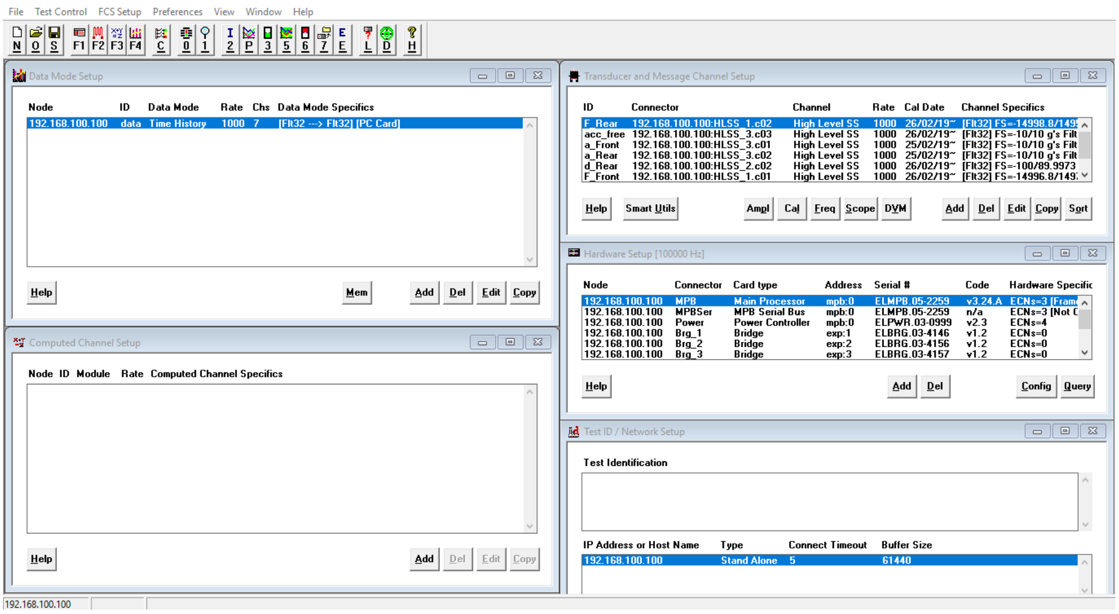

Figure 2.9: TCE® (test control environment) main interface

Data coming from the accelerometers and acquired with datalogger are stored in the system internal memory in *.sie* format. This file must be opened with *HBM SoMat InField®* software, that allow to plot and export an *ASCII* text file processable by other softwar for further calculation.

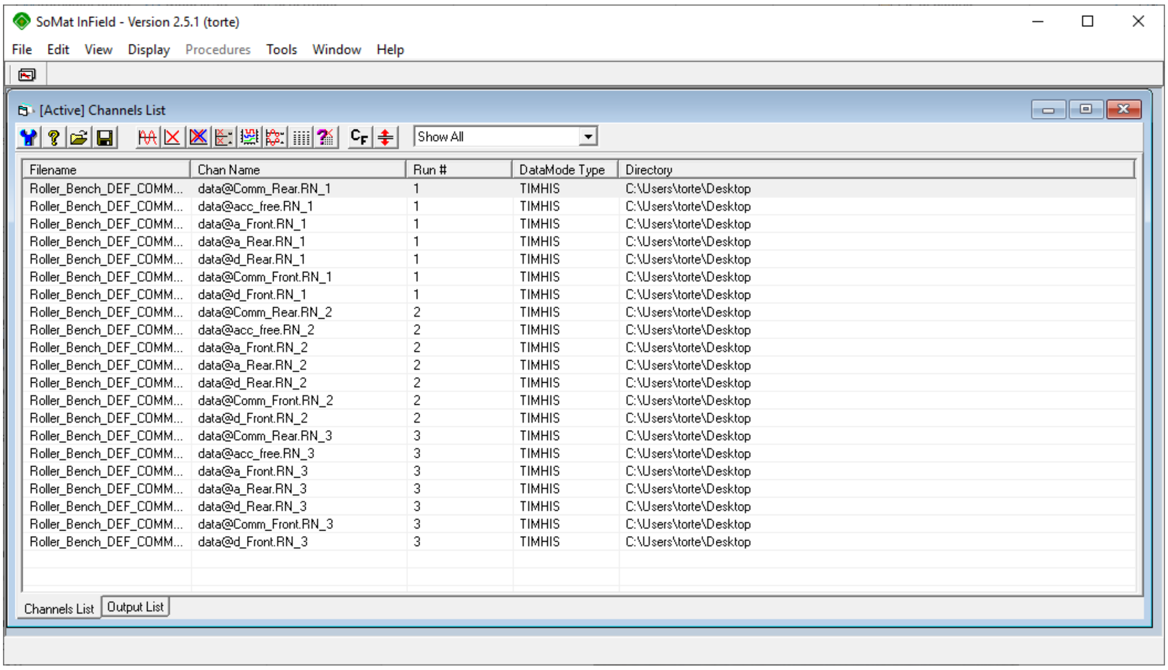

Figure 2.10: HBM SoMat InField® main interface

2.5 ACCELEROMETERS

Acceleration were measured by two uni-axial piezoelectric accelerometers (Fig.8-9) model SoMat HSL 1100 (+/-50 g full scale, 0,3 15000 Hz bandpass).

Figure 2.11: Uni axial acelerometer

Figure 2.12: Uni axial accelerometer with cables

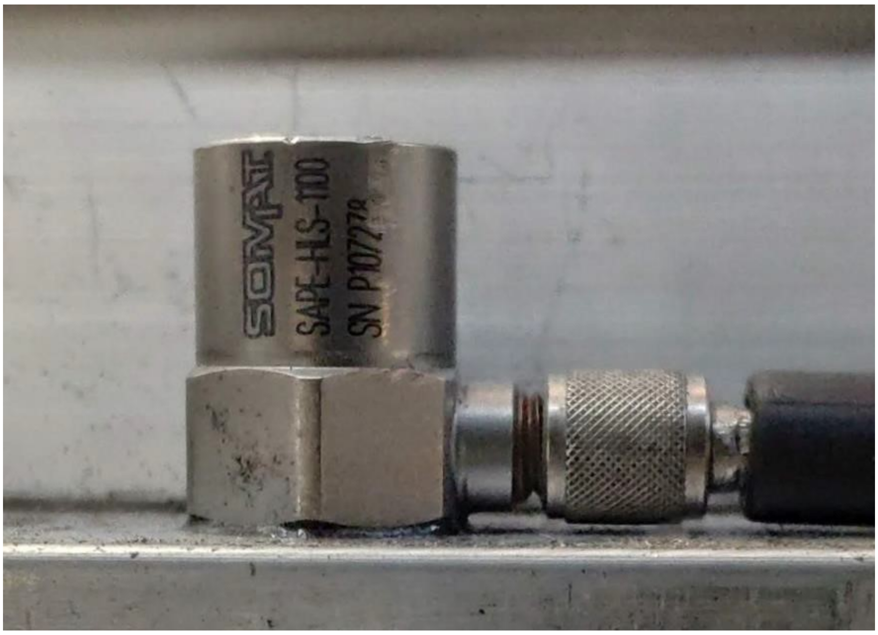

Fig 2.13 Detail of Uni-axial piezoelectric accelerometers

For this application *SoMat® Sensors Series IEPE* (Integrated Electronics Piezoelectric) uniaxial accelerometers are used, with a sensitivity of about $100\ mV/g$ (each device has a specific calibration constant) and with a maximum acceleration range of $\pm 100\ g$. Each of them is connected to an *HLS* port on the datalogger and is configured in *TCE®* software using the standard ICP Accelerometer setup with the specific calibration constant.

2.6 BICYCLE

The bike used is a racing bike produced by the Battaglin workshops in Marostica.

Figure 214.: Battaglin® bicycle used for the tests

Main caratheristics:

Mass: 7,96 Kg

Wheelbase: 1m

Material: alluminium alloy Frame, seat stays carbon fiber, Carbon fiber wheels

Wheel: Total diameter: 0,67m – Carbon fiber (Campagnolo)

Tyre pressure set for tests with *Battaglin®* bicycle is 8.5 bar for the front one and 8.5 bar for the rear one.

2.7 SUPPORT TOOLS AND EQUIPMENTS

- Safety harness
- Personal computer
- Rhinoceros 5® - software 3D CAD for the design of some components ot the test bench;
- Matlab
- Microsoft Excel

3 DEVELOPMENT OF THE NEW TEST BENCH

The development of a test bench allows to carry out in vivo tests under controlled and repeatable conditions to analyze the effect of vibrations coming from road disconnections. Comfort is important in two macro fields of application:

Vibrations in comfort: For city bikes and tour bikes, comfort is an important aspect to consider and there are a lot of methods to reduce them. Vibrations are generated by the road and are transmitted to the cyclist and this, especially in people with wrist and spinal disorders are a problem.
Increasing the level of comfort is guided by studying and researching new materials, new shapes new production technologies and new approach.

Vibration in sport performance: In the last years the the trend in professional cycling is to use very rigid composite frames. This penalize the comfort so cyclists are looking for more comfort because they have the idea that with a more comfortable bike, they are in a better condition to use their maximum power for all the duration of the competition.

Main topics are:

- *Method used to generate vibrations*
- *Test setting, (oggetto dello studio) and acquisition*
- *Analysis*
- *Conclusion*

3.1 EVOLUTION OF THE PROJECT

Part of he work done in this thesis is to consider all the work done before. This is the starting point to understand and define how to proceed. In this way has been possible to define objective, improvements and projectual directions based on previous experience and necessity revealed during tests and after the data analysis.The stages of the bench's evolution are the following:

Stage 1: Definition of comfort*:* Since 2011 a series of projects in Padova University around bicycle comfort started, that try to measure variables that can index comfort of different bicycle components. The research fixed the attention on some particular engineering characteristics that are interesting in terms of comfort: stiffness of components, for static evaluations, and Energy Dissipation and Transfer Function, for dynamic and vibrations evaluations.
Transfer Function is the best index that can be considered in case of induced vibrations, because it lets to know what the behaviour of the system in exercise is and it includes structural and inertial properties. With this variable the dynamic behaviour is well dfined in term of imput and output with a clear correlation.
The research started from a work done by Prof. Nicola Petrone and Federico Giubilato that that allowed the definition of the magnitude of Transfer Function of wheels, saddles and the whole rear part of the bicycle. The results have been correlated with subjective evaluations given by cyclists during field tests.

In the *Figure 3.1* and *3.2* is represented the test benches made in this first study in *Campagnolo®* and Padova University laboratories.

Figure 3.1: Campagnolo® wheel test bench[9]

Figure 3.2: Padova University laboratory test bench

Figure 3.3: Padova University laboratory test bench

The comfort index has been extracted using a protocol applyed to Transfer Function magnitude able to define a comfort index.

To define the Transfer Function, a hydraulic cylinder was used to generate sine input vibrations at different frequencies and several accelerometers to measure the transmitted accelerations and displacements.

Stage 2: The first bench (Seat shaker). The second work was made by Francesco Trabacchin in *2013*[5], with the development of a vibration test bench for bicycles that let cyclists to test bicycle components subjected to vibrations in an indoor controlled environment.

The first test bench was composed by a whole bicycle frame with saddle, handlebar end pedal system, but without wheels, fixed on the wheel axes to a basement. In *Figure 3.4* this test bench is showed (Seat shaker).

Figure 3.4: Seat shaker - Test bench with dummy bottom

In this configuration the cyclist could pedal thanks to a cycle ergometer fixed behind the test bench with a chain coupled to the rear shift. Vibrations were generated by a hydraulic actuator fixed on the seat tube. In first instance this configuration let to calculate the Transfer Function of different saddles.

Vibrations waves were based on a random approach to try to reproduce road vibrations measured during field tests. Following this approach accelerations under the saddle were measured in different road field tests and for each test the Power Spectral Density was calculated.

With *MTS® Function Generator* several random functions were generated, with different parameters in terms of frequency and Root Mean Square and their PSD were compared with field tests. This allowed to correlate a specific type of road with a specific random setup. After the calibration of the test bench in this way, in-vivo tests were done, with cyclists pedalling on the testing system. Each cyclist gave a subjective evaluation that was compared with an index defined by the Transfer Function. In *Figure 3.5* a cyclist pedalling on the same test bench is showed.

Figure 3.5: Cyclist pedalling on Francesco Trabacchin's test bench seat shaker [5]

Stage 3: First bench improvement (Roller shaker). After this pilot project Francesco Trabacchin developed an improved test bench that let to test a whole bicycle with all its components and with a cyclist pedalling on it. The new project was made by two moving rollers for the rear wheel, and one fixed roller for the front wheel.

To guarantee the safety of the tester a safety structure was made, to sustain the cyclist while he's pedalling and to avoid the longitudinal shifting of the bicycle.

Figure 3.6: Francesco Trabacchin's improved test bench Roller shaker for Transfer Function definition [3]

This test bench created an environment more similar to an outdoor bike tour, letting the cyclist to feel free to move. To generate vibrations for the simulation the same procedure of the previous version was adopted, based on field tests and random input. In this configuration there is no possibility to set the friction of the rollers and they are free to rotate.
During these studies a correlation between a Comfort Index (based on the Transfer Function of components) and subjective evaluations was found [5].

Stage 4: Second bench improvement (Double shaker 1.0). [6] The next study around the test bench was made by Prof. Nicola Petrone, Francesco Trabacchin and Fausto A. Panizzolo in *2015* and presented at *ASME 2015*, International Design Engineering Technical Conference.

In this case the system to transmit vibrations from actuator to rollers was improved and it was duplicated to let also front rollers to vibrate, trying to get nearer to a real-like cycling simulation. This new system was composed by an articulated parallelogram that let the rollers to stay in horizontal position while vibrating and it has been designed by Mattia Scapinello [11]. This system is showed in *Figure 3.7.* and is part of the work of this thesis.

Figure 3.7: Test bench with front and rear rollers shaking made by Mattia Scapinello

Stage 5: Last bench – (Double roller shaker 2.0) (07/11/2020) last bench is shown below and it's Mattia Scapinello concept with improvement done by Enrico Girlanda accordin to Mattia Scapinello's prescriptions with an improvement focused on safety structure, platform, handrail.

LEGENDA:

1 - Safety structure
2 - Safety Harness
3 - Rear safety roll
4 - Platform
5 - Rear rolls
6 - Hydraulic actuator
7 - Handrail
8 - Front safety roll
9 - Rear rolls
10 - Bench base
11 - Bike
12 - Cyclist

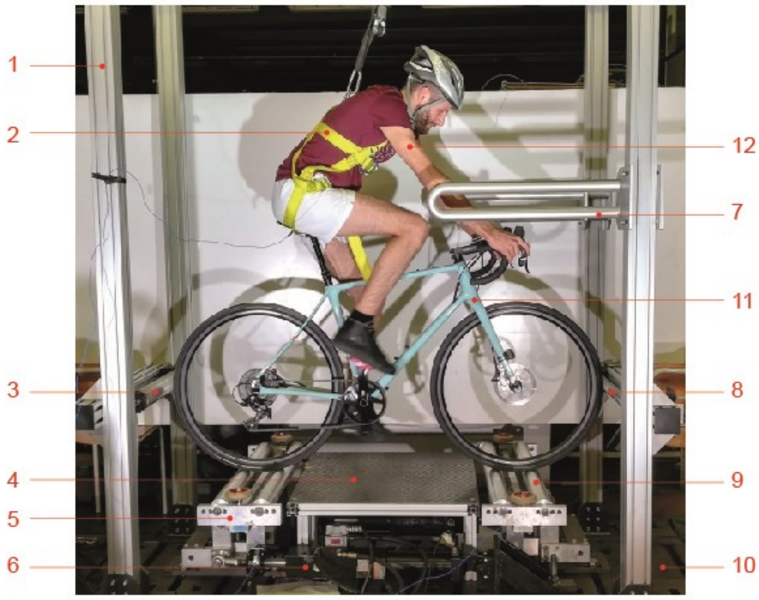

Figure 3.8: Test bench with front and rear rollers shaking improved By Enrico Girlanda

3.2 ASPECTS TO IMPROVE FOR DOUBLE ROLLER SHAKER BENCH

After a careful analysis of the work done and the results and comparing them with the real pedaling and stress conditions of a bike, some limits have become evident that must be overcome with the construction of a new test bench.

Features and limits:

- Stress/vibrations on a single wheel

- Field of motion of rollers shaker

- Safety

Prescriptions, limits and desired features: After the analysis and the observation of the result obtained a list of limitations and prescription but also desired features for the new system have been listed to guide the develope and the definition of the concept for the new test bench.

- *Heigh*t: the bench have to be as low as possible for safety reasons, in case of drop of the tester.

- *Safety*: improve general safety during the pedaling phase of the test

- *Weight*: weights of moving parts have to be limited for inertia reasons. During the motion the component's weight haven't to influence the load measured.

- *Low cost*: use equipment and materials mainly from of the laboratory like: alloy, steel profiles, bicycle rolls, bearings.

- *Stiffness*: at the same time the bench hasn't to deform itself under the load action.

- *Reduce set up and retooling time:* good control of the process, reducing the time of experimentation without having to repeat all times road tests.

- *Greater realism of pedaling and solicitation conditions:* Allow the simulation as soon as possible of the real conditions of the cycling.

- *Suitable for many light wheel vehicle:* Cosidering wheelbase variability for wheelchair ,bike,tandembike,trike bike,handbike,electric bike.

3.3 DEVELOPMENT AND PROJECT DIRECTIONS

Each development direction focus on a well defined need or possibility. A development direction is a way to conceiving o problem and focus on a specific aspect of the problem. define a perimeter of action and the boundary conditions that help to focus on the most important performance in order to follow that specific direction. It will be possible to identify the variables and quantities that act as a limit or opportunity for development.

For each direction a concept is developed and represent the "best as possible" solution for that need. After a deep and large thinking process have been defined five development directions explained as follow:

- ***Capitalization of the knowdlege***: To reduce time development and to use the experience accumulated by Francesco Trabacchin we can achive a better fitting with the real load and pedaling condition. This can be traduced with the develope af new roller shaker with the same desig. We have to ask if this direction impose severe limits or if the affot done allow a better realism and have significant impact.

- ***Exceed limits***: In order to exceed limits that reduce realism of the system in order of the way the load is applied, we can ask if there is a way to modify the system and improve the realism. This can be done identifying what are aspect or caratheristics that acts as a limit and doesn't allow to obtain the desired perfromance.

- ***Reduce the influence of the system***: Reduce inertia is the first way to follow. Inertia of the rolling system impose a reduction of applicable frequencies and introduce a resonance frequency. This can be a problem or a limit if must be conduced test that need frequencies that superate the phisical limits of the sistem. The ideal sistem have no inertia but in a real case the problem must be faced by asking ourselves in what conditions I can minimize the inertia by guaranteeing small deformations.

- ***Reduce complexity****:* Eleiminate each form of complexity can be a way to reduce the number of variables to consider ion the probleme and can have an high level of benefits compared to costs and in relation to the results obtainable. Create the condition to create a direct link (ideally the abscence of any system) between the actuators and the wheel coul be a solution.

- ***Maximize realism***: Analizing the way the contac from tyre and road occours, and inspiring by the tradmill has been idagate the possibility to blend the caratteristic of realism of the bike treadmill and the roller shaker. realism is an important aspect that reduces the distance between a test in the laboratory and one on the road and is also important because allow the biomechanical of pedaling that include aspect important like inertia due to pedaling, forces excanced,posture,micro movements,pedaling power.

3.4 CONCEPTS DEVELOPED

After a precise definition of the need and the desired result, In the following scheme, for each development dirtection is proposed a possible solution. The solution must be considered not the only but a possible one. Each solution, faithful to the design direction, try to resolve needs but in a different way. This obviously have advantages, disdvantages and specific caratheristics summarized below:

Capitalization of the knowdlege*:* limit of the field of motion, easy to reproduce,

Exceed Limits*:* New system, better field of motion, more realism,

Reduce the influence of the system*:* Reduce mass,avoid presence of rolls,

Reduce complexity*:* Direct link, no system, less realism, no pedaling allowed

Maximize realism*:* Reproduce ride condition, allow pedaling biomechanic, treadmill, cost

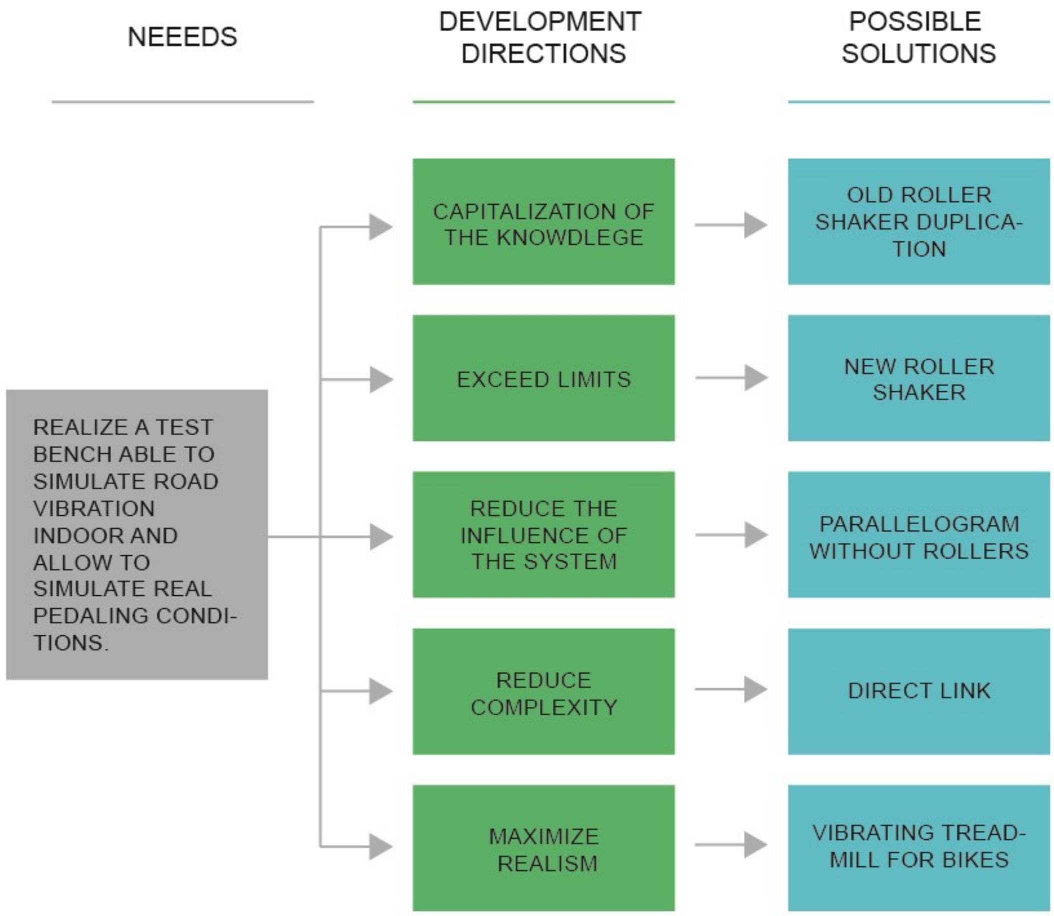

Figure 3.9: Development direction for the new test bench

3.4.1 IDEATION PHASE

After a deep analysis of each development direction the following concept are presented:

Concept 1: Old roller shaker duplication. With the intent to count on previous experience done by Francesco Trabacchin is possible to duplicate the configuration of the roller shaker. The limit to consider is the field of motion not ideal when displacement over 10 mm must be imposed. This magnitude of displacemente are not considerable vibrations.

Main components: Lever system – rollers

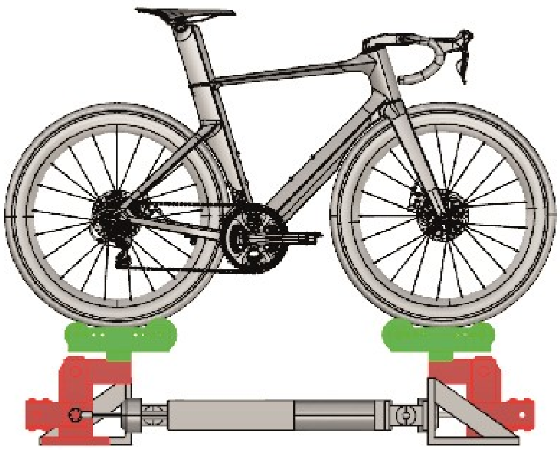

Figure 3.10: Lateral view for concept 1

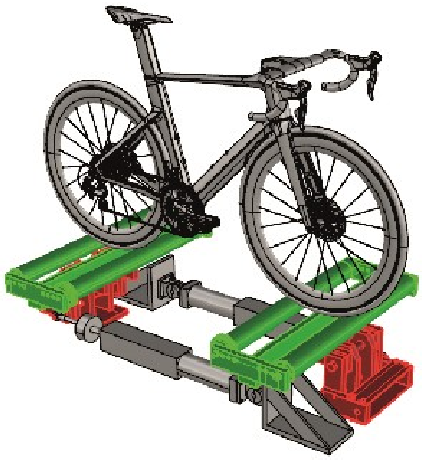

Figure 3.11: prospective view for concept 1

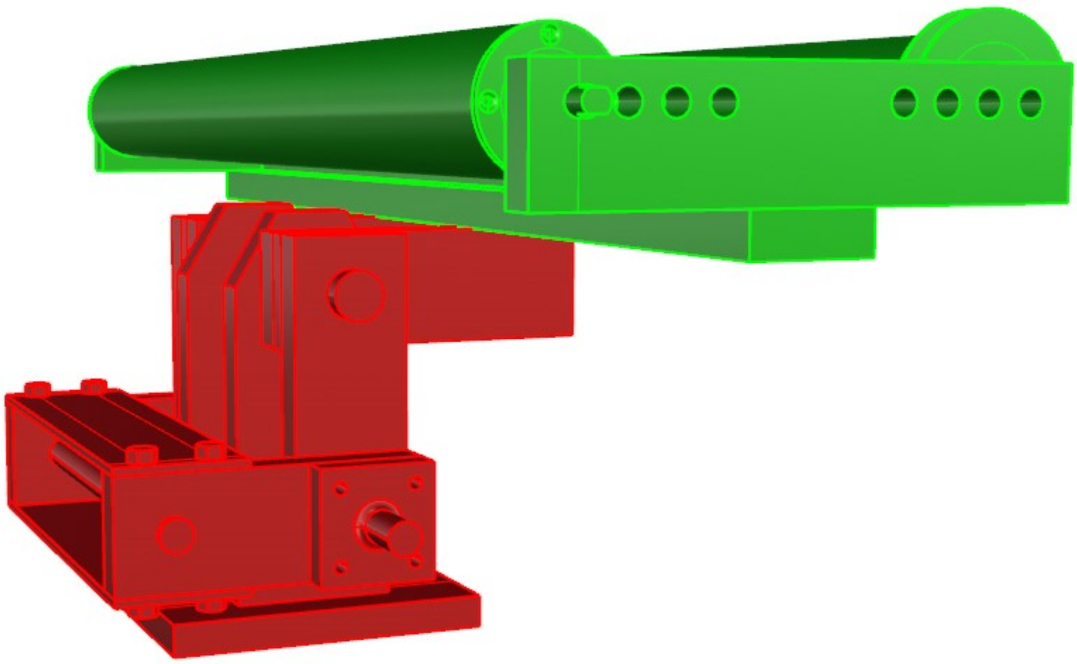

Figure 3.12: Roller shaker of concept 1

Concept 2: New roller shaker. Modify the old roller shaker in order to drastically reduce the horizontal component of the field of motion can be the first step to better reproduce real conditions. The idea is to adopt the articulated parallelogram to reduce the horizotal component of the field of motion.

Main components: Articulated parallelogram – roller system

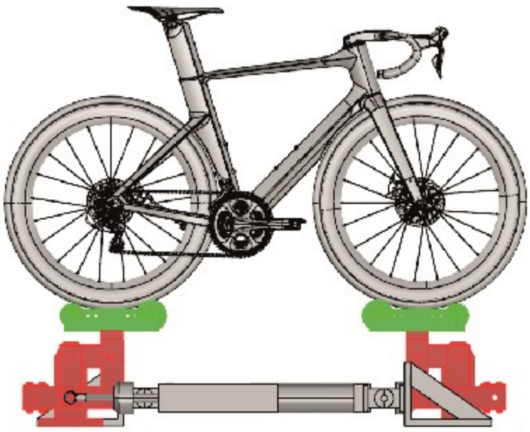

Figure 3.13: Lateral view for concept 2

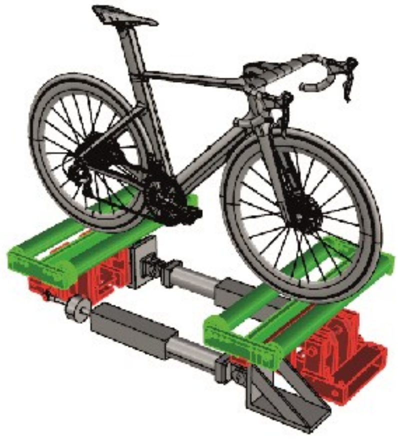

Figure 3.14: prospective view for concept 2

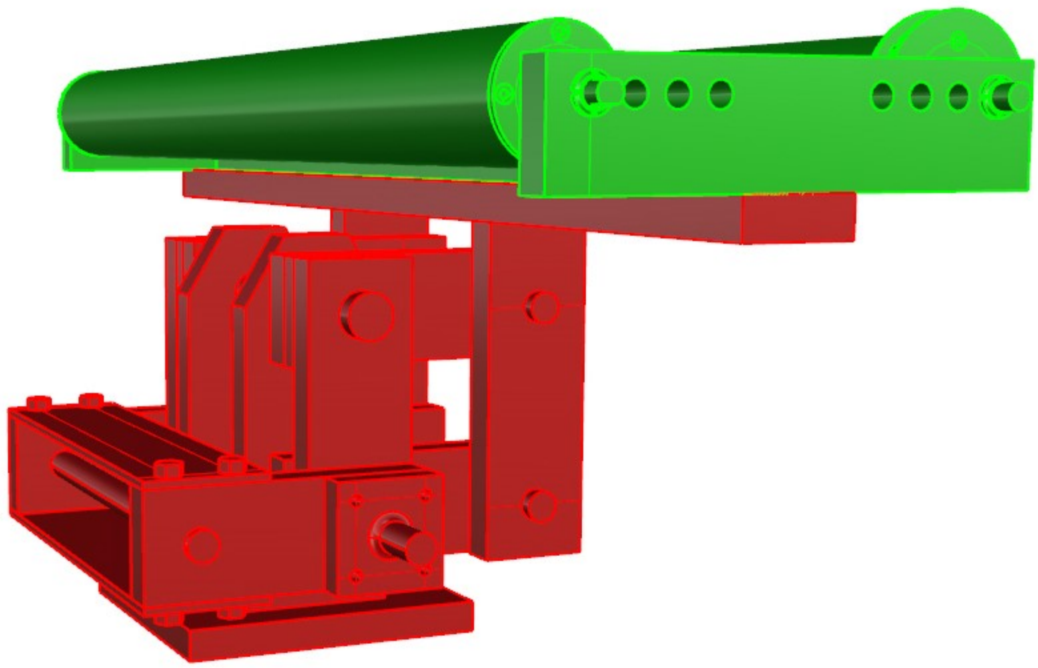

Figure 3.15: Roller shaker of concept 2

Concept 3: Parallelogram without rollers. Rollers and structure posed over the shaker have a certain mass and have a certain resonance frequency. This can be a limit if there is a necessity to conduct test campaign at high frequency. The idea is to realize a direct connection between the shaker ant the wheel. A certain gradeof realism in pedaling is lost but is possible to reduce inertia and resonance problem.

Main components: Articulated parallelogram – clamp connection

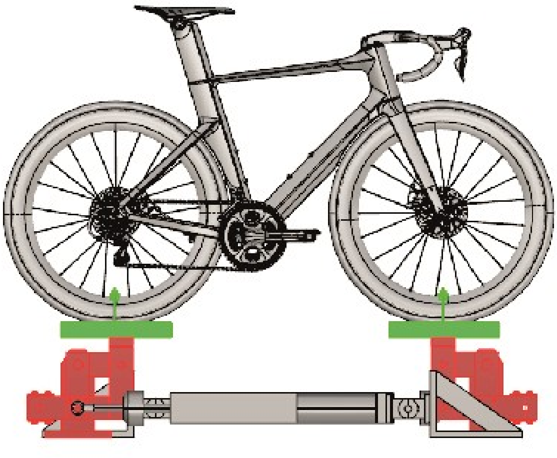

Figure 3.16: Lateral view for concept 3

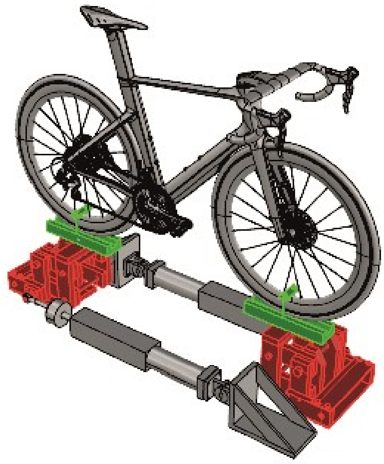

Figure 3.17: prospective view for concept 3

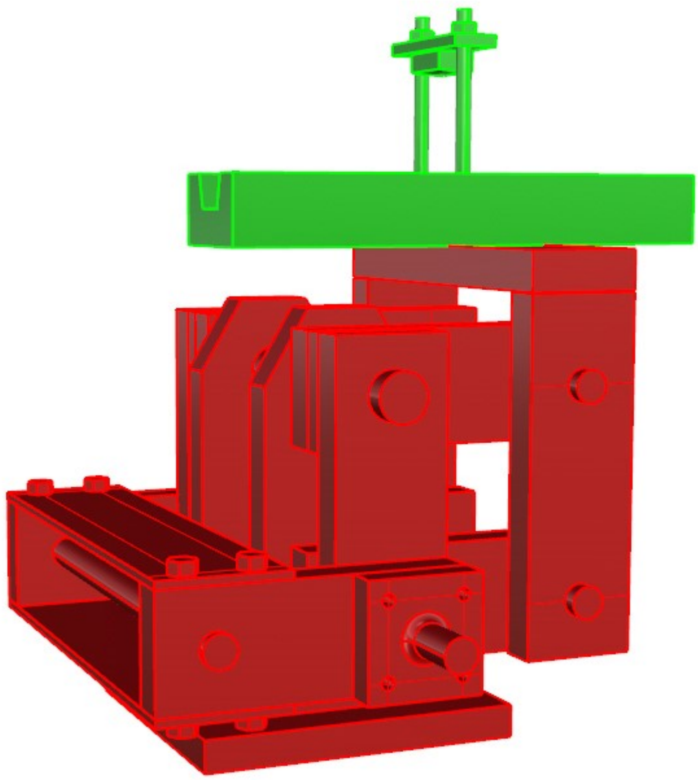

Figure 3.18: Roller shaker of concept 3

Concept 4: Direct link without roller shaker. The direct connection between the actuators and the wheel allow to further reduce inertia and improve the realism in the in the manner in which the load is transmitted. This not allow the biomechanic pedaling and wheel are fixed.

Main components: Clamp connection –linear guides

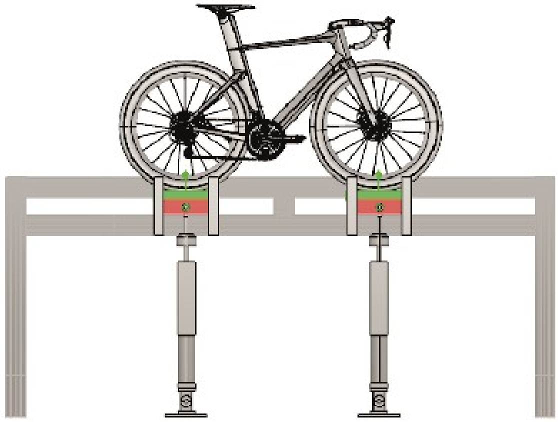

Figure 3.19: Lateral view for concept 4

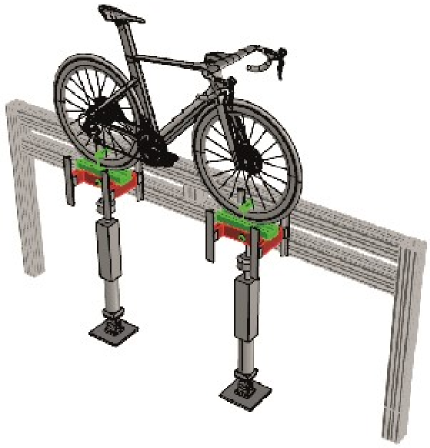

Figure 3.20: prospective view for concept 4

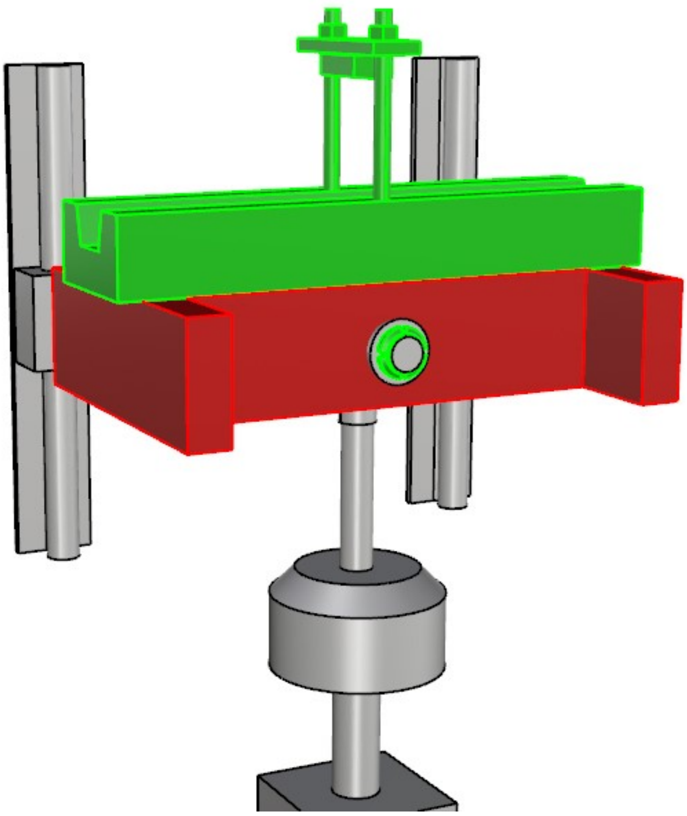

Figure 3.21: Roller shaker of concept 4

Concept 5: Vibrating treadmill for bikes. In order to reproduce at best the real pedaling condition the concept is based on a treadmill

Main components: Conveyor belt – sledges

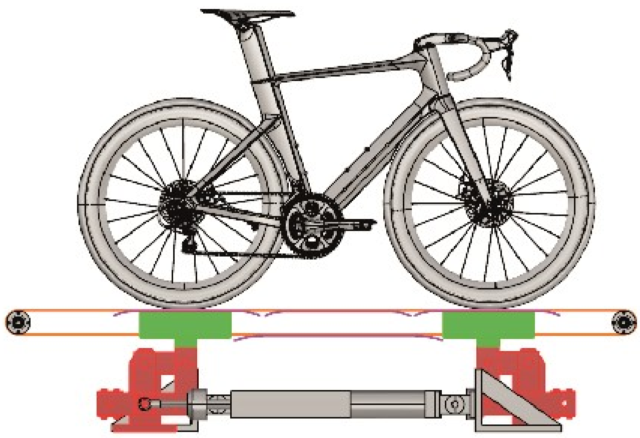

Figure 3.22: Lateral view for concept 5

Figure 3.23: prospective view for concept 5

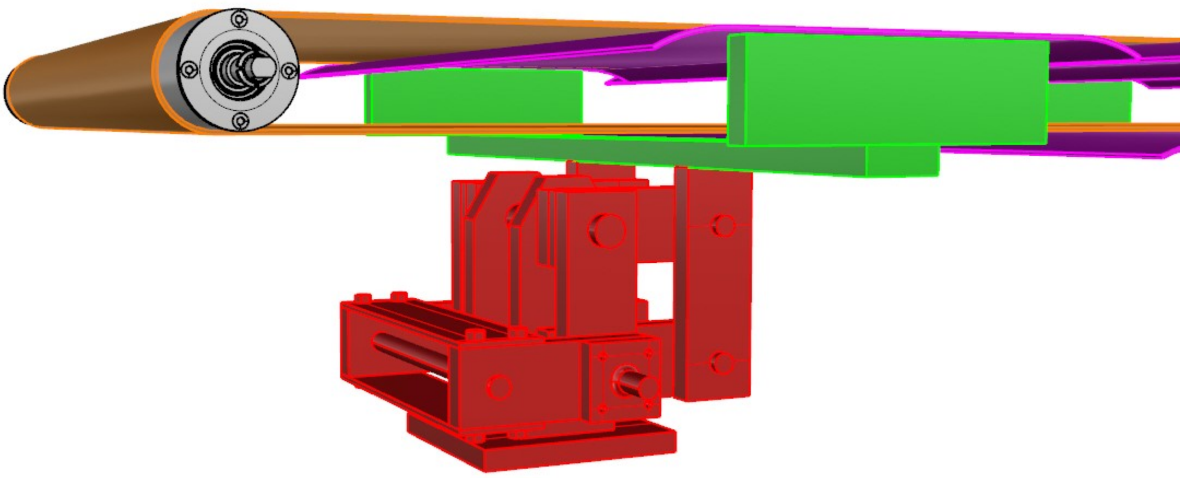

Figure 3.24: Roller shaker of concept 5

3.4.2 PRELIMINARY SIZING

The preliminary design will focus on concept number 5 being the most complex of all and being the configuration that has the greatest number of unknowns. For all the others, the general design does not present particular criticalities that would make their realization doubtful.

Concept1: Old roller shaker duplication*:* Reproduce the work done by Francesco Trabacchin

Concept 2: New roller shaker*:* Validation of the work done by Francesco Trabacchin and improvement of some details

Concept 3: Parallelogram without rollers: Follow the guidelines of concept 2

Concept 4: Direct link without roller shaker: Design of wheel connections

Concept 5: Vibrating treadmill for bikes*:* Following are the steps for the preliminary design functional to the determination of the construction cost.

Max speed (Vmax) = 30Km/h =8,33 m/s
Motor roller diameter: 0,0375m
Omega rullo= V/r =8.33/0,0375= 222,1 rad/s
*Ng rullo motore =Omega*60/6,28 =2121 g/min*

*Friction force(slitta/nastro)= F*μ= 1000N*0,25=250 N*

*Motor roller torque = 250N*0,0375m = 9,375 Nm – Approximation: 10 Nm*
*Power to the roll = Motor roller torque * Omega rullo = 9,375N * 222,1 g/min = 2082,18 W*
Overall transmission efficiency (supposed): 0,85
Electric motor power: 2082/0,85=2449 w

Electric motor (g/ min) = 3000 g/min
R(transmission ratio= omega2/omega1)=2121g/min/3000 g/min=0,707
Omega electric motor=314 Rad/s

Electric engine torque =Electric engine power/Omega electric engine= 2449W/(314rad/s)=7.8 Nm
Service factor chosen (Fs)=1,1
P corrected =2694W
P (electric motor) > 2694 W

Aspect to indagate:
The unknown is the possibility of damping the possible wave produced by the shakers and mutually transmitted so that there is no mutual influence.

3.4.3 ADVANTAGES AND DISADVANTAGES

Each time a concept is developed the primary function or caratheristic maximize the performance that allow to obtain the desired result (ex minimize inertia or maximize realism) this can affect other performance that must be kept under control because are anyway desiderable.
After have been discuss the performance of single concept a way to define the best compromise and the winning one is proposed. In this way is possible to monitor the parameters that converge in a project. For each concept are defined critical issue, main caratheristics, advantages and disadvantages:

Concept 1: Old roller shaker duplication

Advantages	Disadvantages
- Simple configuration - Low economic investment - Low design effort - Soluzione collaudata - Previous experience and data available - Allow biomechanic of pedaling	- Motion field with rotation component Problem when using two shakers

Concept 2: New roller shaker (articulated parallellogram)

Advantages	Disadvantages
- Better control of field of motion - Allow biomechanic of pedaling - More safety - Low vertical development of the system	- Number of components - More complexity

Concept 3: Parallelogram without rollers

Advantages	Disadvantages
- Low inertia respect rolls configuration - More safety - Lower number of parts respect rolls configuration - Low vertical development of the system - Lower complexity respect rolls configuration	- Low realism - Do not allow biomechanic of pedaling

Concept 4: Direct link without roller shaker

Advantages	Disadvantages
-Very low inertia -Simple concept -Simple to realize -Low vertical development of the system -More Safety -Low number of parts to built -Good actuators control	- Do not allow biomechanic of pedaling - Vertical development of the system - Difficult to retooling - Usable only with dummy

Concept 5: Vibrating treadmill for bikes

Advantages	Disadvantages
- Great realism - Allow biomechanic of pedaling - Low pedaling difficult respect to rolls - Easy retooling operations - Very low necessityo to variate wheelbase	- Increasing complexity respect other solutions - Greater economic investment - Security with more critical issues in the event of loss of control - Influenza del nastro sulla trasmissione delle vibrazioni Influence of the belt on the transmission of vibrations

3.4.4 CONCEPT COMPARISON AND CHOICE

When is necessary make a choiche between different concept or projects is necessary to evaluate them considering a lot of aspect convergin on their realization. This is due to the fact that a solution can perfectly fit an objective but not reach enought performance in aspect that are equal or just less important.
The perfect product or the perfect machine dosen’t exist but is necessary, consdiering important aspect and caratheristics to create a not arbitrary (less than possible) method to evaluate each concept.

Logic and method: The logic behind the choice of one concept over another is to make as rational as possible the evaluation of the performances that contribute to the definition of its performance valued in relation to certain attributes that contribute to defining its functionality, usability, affordability.
The individual attributes do not necessarily have the same weight (for example, it is acceptable to consider that for the concept its difficulty of use may be a relative problem but and therefore weigh little but that cannot be said for safety) and a precise evaluation of each one must be performed according to a hierarchical criterion.

***Attributes**:* Are all the most important aspects converging to the functionalities for the bench. All of them are independent from the concept and are pre requisites.

Each attriubute is expressed in a way that allow increasing score to increasing performance.This is because better is the value and higher must be the score to allow a cumultaive addittive score

- ***Previous experience***: (amount of data, quality of data, test field comparison)

- ***Speed of development***: Estimated time for recieve ordered components, design the bench and for it's phisical realization (machinery and assembly) considering the maximum time possible of 3 months. Compiling a Gantt chart for each concept and regarding each development phase, and scheduling delivery times for components.

- ***Realism***: Considering the cinematic and dinamic of pedaling on a bike and considering the interaction between tire and wheel with the road.

- ***Cheapness of realization***: the inverse of cost realization Considering a maximum cost of 2000 Euros for materials and machinery operations, exercise costs lower is the cost and higher is the score. (score attributed S=1)

- ***Level of knowns***: (the inverse of the level of unknows) How many variables (cinematic and dinamic behaviour of the solution, need of accurate design and calculation) are necessary to obtain a MVP (minimun viable product) .Is the level of unknow indagable with simulation or software or by analitic equation or consideration? are time and resource involved invasive?

- ***Ease of use***: The level of simplicity with wich is possible to perform an in-vivo test considering the hardware part of the banch.

- ***No Limits***: No limits =1 means a lot of limits, No limits =10 means that there are no limits.

- ***Set up speed***: the speed with wich preliminar operation are possible like assembling parts, bicycle change, sensoring.

- ***Safety level in use***: The level of safety for the tester and for other operators during a test.

Attribute definition*:* They are global attributes because they directly define the performance of the concept. Each of them is defined according to local attributes defined below.

- *Previous experience:* Amount of data,quality of data,test field,validation
- *Speed of development:* Design, recieve parts, realization
- *Realism:* Pedaling biomechanic,tire and wheel interaction,pedaling effort
- *Cheapness of realization:* Design,materials,machinery,assembly,exercise cost
- *Level of knowns:* Design, material,effectiveness, solicitation realism
- *Ease of use:* Pedaling, setting, monitoring
- *No Limits:* technological, logistical, regulatory, usability, mechanical, physical
- *Set up speed:* Bike, cyclist, parts
- *Safety level in use:* Low height, dangerous parts, not risk of falling

Performance evaluation*:* To evaluate performance and compare different concepts, the concept of the weighted average with respect to the defined attributes is used. It means creating a hierarchy of priorities through weights in order to give greater or lesser importance to certain performance.

Performance*:* caluclated as weighted average without any restrtiction in the case some Si are equal to zero. This calculation rewards high score in attributes with hight weight.

$$Performance\ (P_i) = \frac{\sum_1^n W_{ij} \cdot S_{ij}}{\sum_1^n maxW_{ij} \cdot maxS_{ij}} \tag{3.1}$$

Enhanced performance: It's a way to rewards concept with high score in performance with low Weight (Wij).

$$Enhanced\ Performance\ (EP_i) = \frac{\sum_1^n W_{ij} \cdot S_{ij}}{\sum_1^n maxW_{ij} \cdot maxS_{ij}} \cdot \prod_1^n S_{ij} \tag{3.2}$$

$$Restricted\ Performance\ (RSP_i) = \frac{\sum_1^n W_{ij} \cdot S_{ij}}{\sum_1^n maxW_{ij} \cdot maxS_{ij}} . \prod_1^n S_{ij} = 0\ if\ one\ S_{ij} = 0 \tag{3.3}$$

Restricted performance: is the Enhanced performance in case one score Sij is equal to zero. A Sij=0 can be an high limit able to exclude the concept. This because is assumable that each attribute is important and fondamental and is not possible to do without any attribute. If we consider the performance the concept with an Sij=0 and not the restricted performance, is considerable resonable the fact to let partecipate to the competition and the fact to have a score equal to zero. The restricted performance sai it must be excluded.

Legenda:

$i = 1 \dots n\ (numero\ concept)$; $J = 1 \dots 9\ (numero\ attributi\ globali)$; $k = indice\ locale\ attributo$

$s_{ij} = f(\alpha_{ijk}, LW_{ijk}, LS_{ijk}) = score\ attributed$ to the concept i for the attribute j. Values from 0 to 10 – Max Si =10 means is not possible to do better at the state of art.

$w_{ij} = weight\ attributed\ to\ the\ concept\ i\ and\ relative$ at the global attribute j. It reflect the importance of a determined performance respect to other. Max Wi= 10 means maximum priroity to a specific attribute.

$\alpha_{ijk} = local\ attributes\ that\ define\ the\ global\ Sij\ score$

$LW_{ijk} = local\ weight\ for\ the\ local\ \alpha_{ijk} attribute$

$LS_{ijk} = local\ score\ for\ the\ local\ weight\ LW_{ijk}\ and\ the\ local\ attribute\ \alpha_{ijk}$

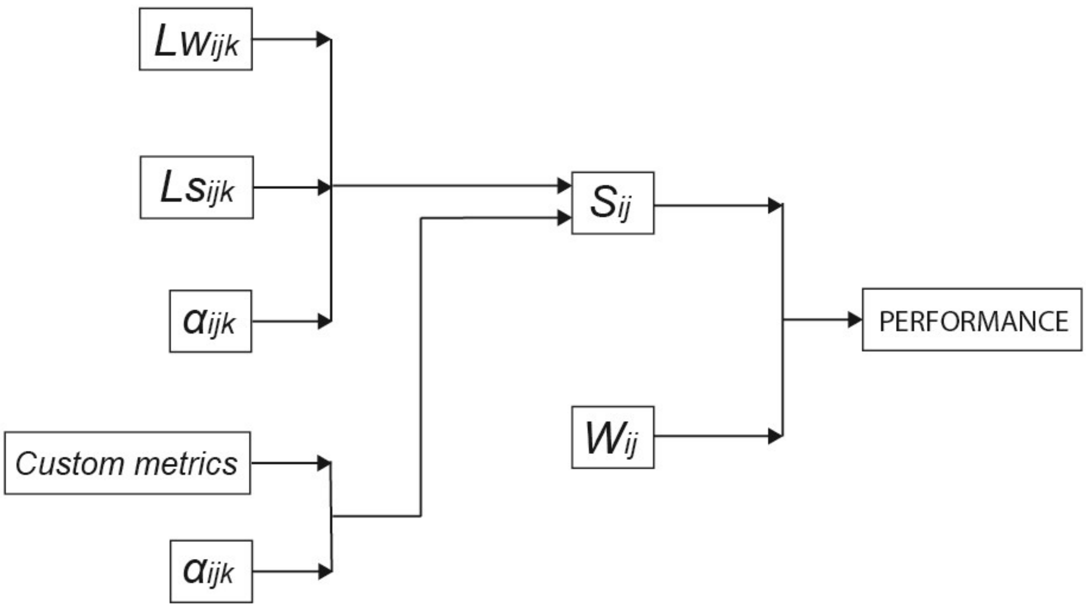

Figure 3.25: Scheme relative to score, attribute and performance evaluation

Assumptions:

- *ij (W)=ij (α)*
- Wij Must be the same for each concept. Custom Wij (different weight for the same perfromance in different concept) are not considerate in this case.
- Wij=Wheight of the single performance in the function variate from 0 to 10
- Sij= Score attributed to the single performance variate from 0 to 10

Closeness*:* Two or more concepts can have elements or operating principles in common. For example, if for concept 1 there are elements validated in relation to the attributes and concept two shares a part of them, they can be partially or totally inherited. This is if for concept 1 they have been validated.

Among the concepts a contiguity and closeness index (CCI) is defined which varies between 0 and 10 in relation to the various attributes. Each attribute will have its own score (a contiguity index must be defined… from 1 to 10) if it is greater than or equal to 6 then elements validated by the reference concept can be taken. The score is activated only if there is objective contiguity.
For this reason, not all mathematically possible combinations will be taken into consideration. The score indicates the "father" who owns the validated information and the "son" who inherits it.

Ex: CCI (1-2) / previous experience = 7

Concept 2 inherits from concept 1 in relation to the previous experience. The fitting is equal to 7/10 points

Procedure*:* to determine the score of each concept, proceed in the following way:

- For each attribute the relative tables that define it are compiled. This is done for each concept. Two examples are given below:

	1) PREVIOUS EXPERIENCE Wij (j=1)							
	αijk	LWijk(1-10)	LSijk(1-10)	LWijk*Lsijk	Max Lwijk	Max Lsijk	Max LWijk*Max Lsijk	Sij
K=1	Amount of data	5	6	30	5	10	50	6,5143
K=2	Quality of data	8	8	64	10	10	100	
K=3	Test field comparison	9	6	54	10	10	100	
K=4	Validation	10	8	80	10	10	100	

Figure 3.26: Scheme relative to single performance score for previous experience

	2) SPEED OF DEVELOPMENT (j=2)				
	αijk	Days	Sij(<90 days)	Sij (=90 days)	Sij (>90 days)
K=1	Design	15			
K=2	Recieve parts	15	5,0000	FALSO	FALSO
K=3	Realization	15			
	Somma giorni	45			
	max giorni	90			

Figure 3.27: Scheme relative to single performance score for speed of development

- For each concept the relative table is compiled which will report the different scores that will be evaluated

	CONCEPT 1: Old roller shaker duplication - (CAPITALIZATION OF THE KNOWDLEGE) (i=1)											
	Attributes	Wij (1-10)	Sij (1-10)	Media Sij	Wij*Sij	Max Wij	Max Sij	Max Wij*MaxSij	Performance (P)	Enhanced(P) restricted if =0)	Correction factors (Cfij)	Revalued (P)
j=1	1-Previous experience	7	6,5143		45,6000	10,0000	10,0000	100,0000			1,0000	
j=2	2-Speed of development	7	5,0000		35,0000	10,0000	10,0000	100,0000			1,0000	
J=3	3-Realism	10	6,5385		65,3846	10,0000	10,0000	100,0000			1,0000	
j=4	4-Cheapness of realization	6	2,6167		15,7000	10,0000	10,0000	100,0000			1,0000	
J=5	5-Level of knowns	7	7,3333	6,3262	51,3333	10,0000	10,0000	100,0000	0,5076	612,8418	1,0000	0,5076
J=6	6-Ease of use	8	6,3667		50,9333	10,0000	10,0000	100,0000			1,0000	
J=7	7-No Limits	10	6,3667		63,6667	10,0000	10,0000	100,0000			1,0000	
J=8	8-Set-up speed	6	8,2000		49,2000	10,0000	10,0000	100,0000			1,0000	
J=9	9-Safety level in use	10	8,0000		80,0000	10,0000	10,0000	100,0000			1,0000	

Figure 3.28: Scheme relative to single concept score

Scenario: we recognize that a link between some performances doesn't have to be fixed and we recognize the fact that the weight of certain attributes can be customized according to the concept. It means creating bonds and laws of interdependence. Bonds are created when you voluntarily make a decision or create a rule or when an interdependence is highlighted and you want to manage it.

Rules: A scenario is an IF condition that can be valid for some reason and as example only for some concept, for each concept or for some combination. It is a situation in which it is possible to define rules and interdependencies in an arbitrary way for well-defined purposes.

- Weight management: Not isoweight, not the same Wi for different concepts. It's a valutation that must be done carefully.

Rewarding rule: *This because is acceptable for a concept that is able to enhance a certain performance, give less weight to not competitive aspect linked to its realization* (Ex:the assumption is to premiate concept with high level of realism)

- *If realism is > = than 8*
  Then
  Cost weight Wij is < =3 (or cost weight is 0,8 cost weight attributed)

Penalizing rules: This because can be opportune to define an extra condition that affect the performance or the score if two attribute not reach the level of sufficiency established (Ex:the assumption is to penalize concept with both low level of cheapness and realism)

- *If cheapness of realization Score is <= 5 and if realism is <=5*
  then
  *Final Performance is 0,9*P*

- *If cheapness of ralization Score is < = than 5*
  then
  *Si =0.9*Si*

Revalued performance*:* among the defined attributes there may be some of greater importance because they allow a better convergence of purpose. For example, an economical but unrealistic solution is not as desired as a higher cost solution but also much more realistic. To take this into account, it is possible to act on the Wij weights or to adopt rewarding corrective coefficients if certain conditions occur. In the case in question, since realism is a very important aspect, it was decided that:

- If realism score is > = 7
  then

  Revalued performance is = 1.05 * P

$$Revalued\ Performance\ (RP_i) = \frac{\sum_1^n W_{ij} * S_{ij}}{\sum_1^n maxW_{ij} * maxS_{ij}} * \prod_1^n Cf_{ij} \tag{3.4}$$

Cfij = correction factors ; Cfij >1 (rewarding) ; Cfij <1 (penalizing)

Ties and interdipendence*:* Link between different performance. it is not an arbitrary but real bond or interdependence that can be found and formally defined. For exampre Design cost is linked with cheapness of realization with the design by hourly cost.

Winning concept

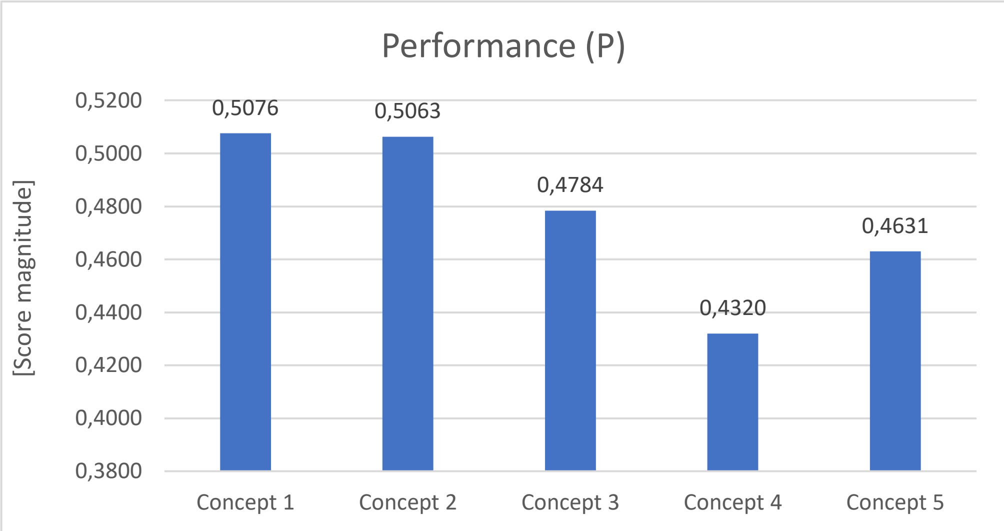

Figure 3.29: Performance P comparison

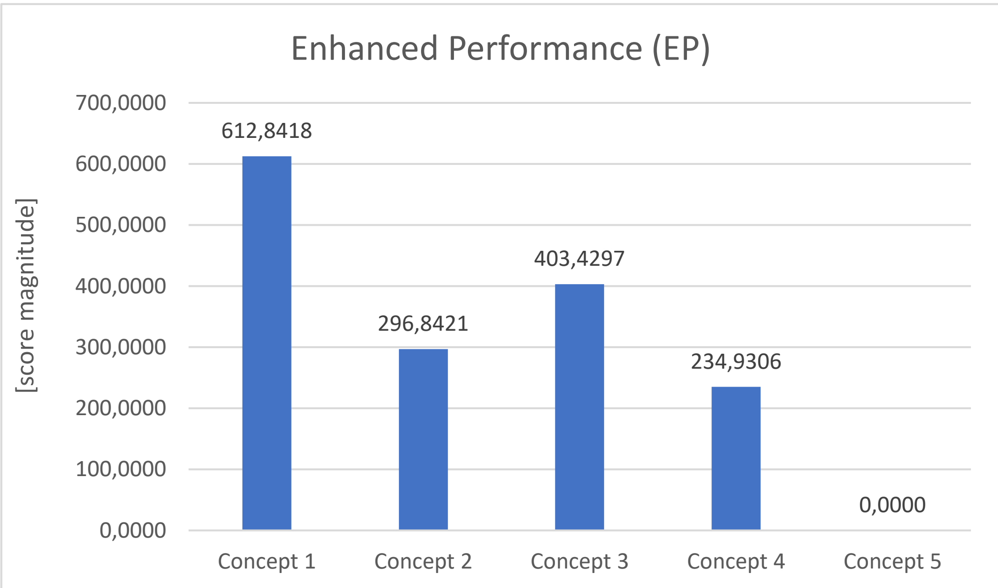

Figure 3.30: Enhanced performance EP comparison

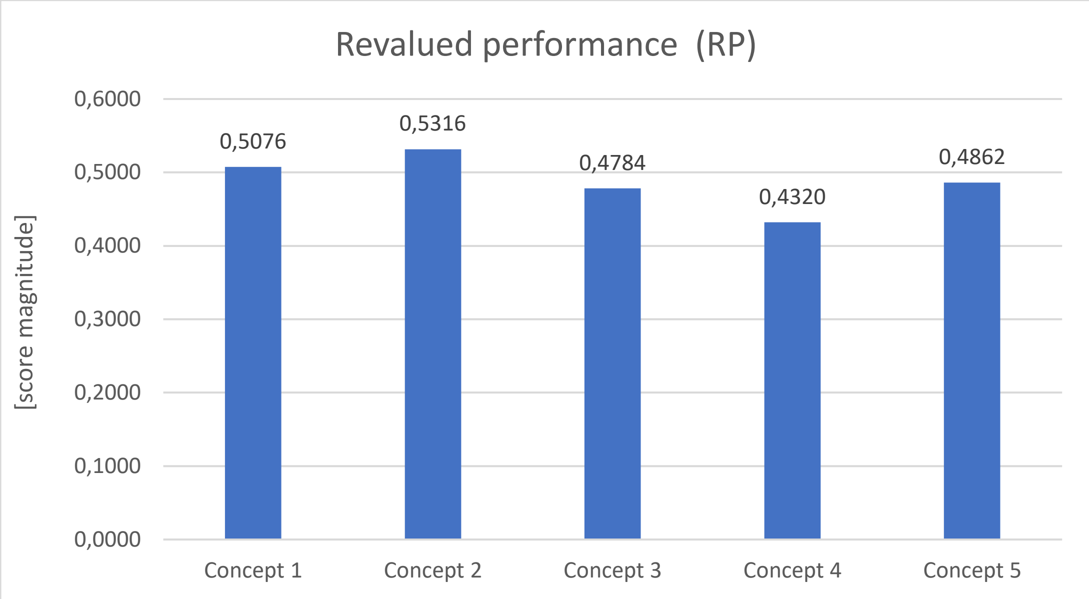

Figure 3.31: Revalued performance RP comparison

Explanation of the choice*:* By observing the graph showing the perfrormance P, concept number 1 reaches the highest rating but with a small detachment from concept number 2. Observing the enhanced performance (EP) it is instead observed that concept number 2 has a significant detachment. This despite an average score value that differs slightly. From the analysis of the various Sijs, it can be seen that this is due to the higher economy score of concept 1 compared to number 2.
However, given that Concept 2 achieves a Sij score with regard to realism> 7, its performance is rewarded by a multiplication factor of 1.05 being a very important aspect for reproducing indoor pedaling conditions outdoors. In this way, the advantage of concept 1 in terms of greater cost-effectiveness is lost as it is a less influential parameter than realism.
This is all the more reason for the fact that the concepts that exceed the foreseen budget receive a score equal to zero in the relative Sij and therefore are excluded as happens for the number 5 which represents the solution with the greatest the fixed limit.
Considering these aspects, **concept 2** will be developed.

3.4.5 DEVELOPMENT OF THE CHOSEN CONCEPT 2 (ARTICULATED PARALLELOGRAM)

The concept created consists of an articulated quadrilateral to which the roller system is anchored. The cylinders can be connected at the rear or side in order to favor compactness. The rollers can be brought together and, if necessary, it is possible to design them with reduced width.

The various views are shown below:

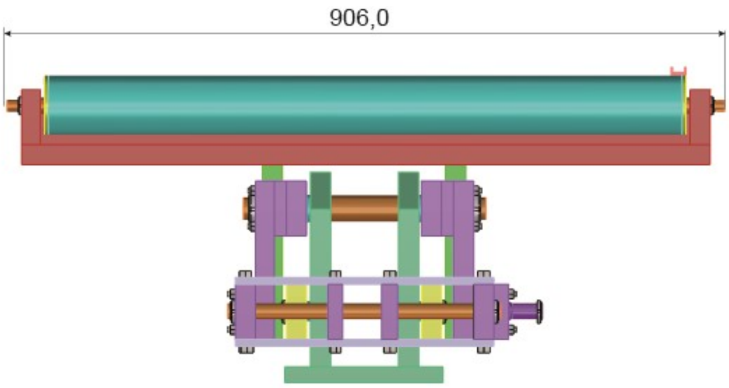

Figure 3.32: Assembly front view

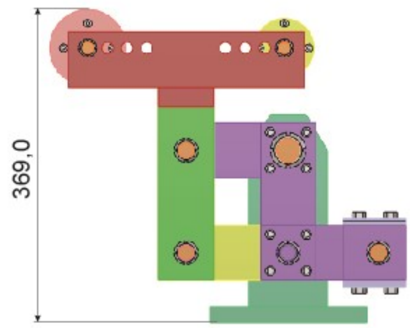

Figure 3.33: Assembly lateral view

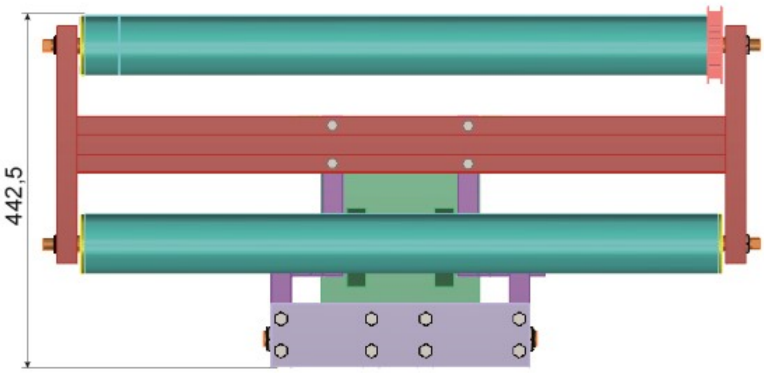

Figure 3.34: Assembly top view

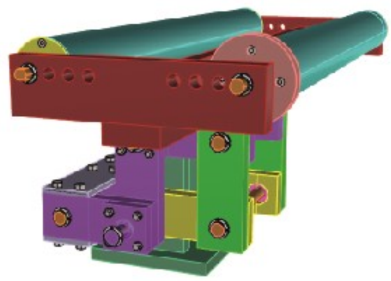

Figure 3.35: Assembly prospective view

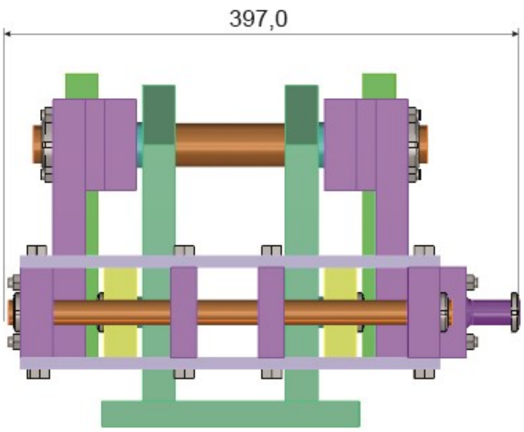

Figure 3.36: Parallelogram front view

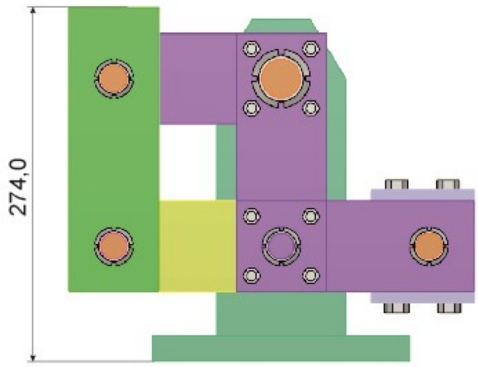

Figure 3.37: Parallelogram lateral view

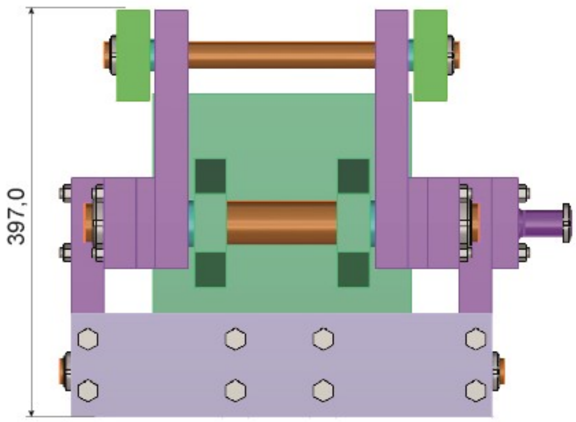

Figure 3.38: Parallelogram top view

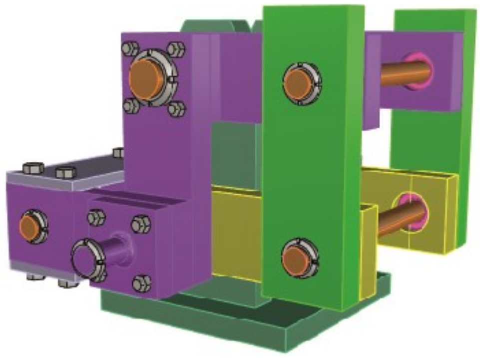

Figure 3.39: Parallelogram prospective view

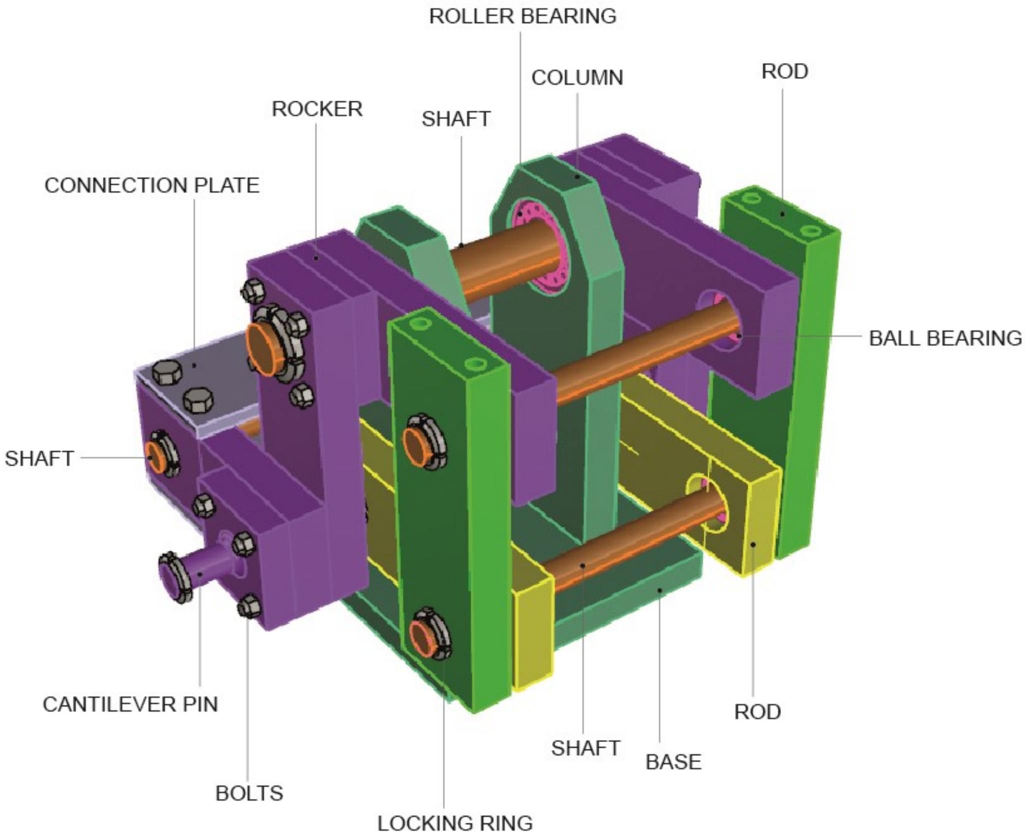

Figure 3.40: Parallelogram components

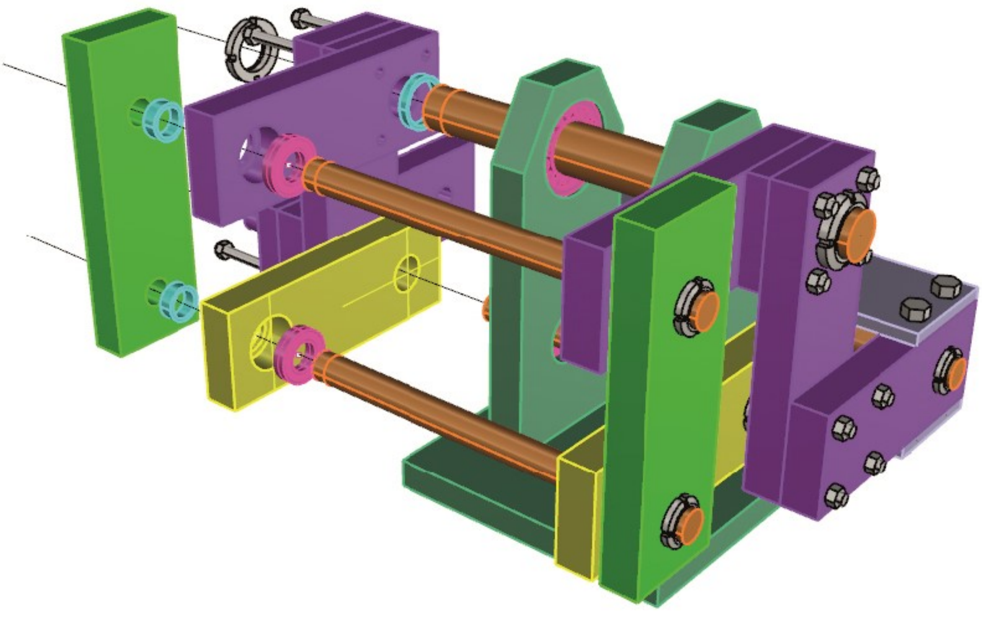

Figure 3.41: Parallelogram semi-explosed view

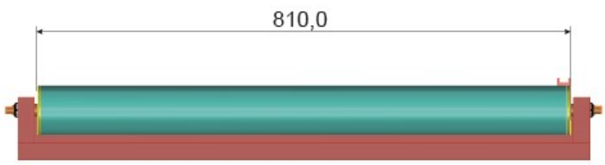

Figure 3.42: Rollers front view

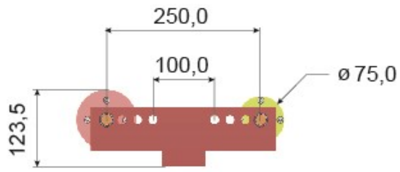

Figure 3.43: Rollers lateral view

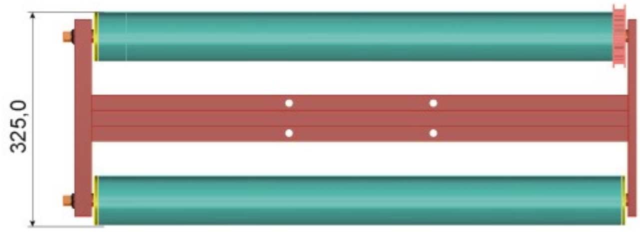

Figure 3.44: Rollers top view

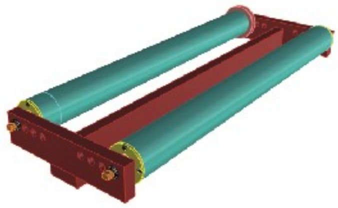

Figure 3.45: Rollers prospective view

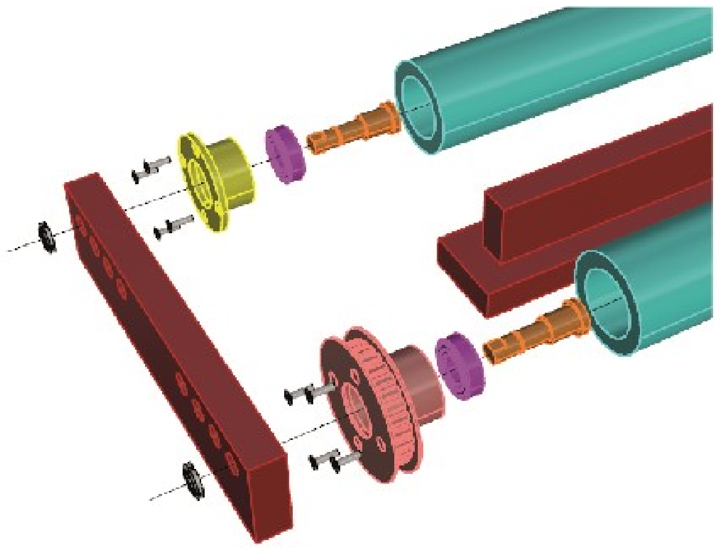

Figure 3.46: Rollers semi-explosed view

Overall system:

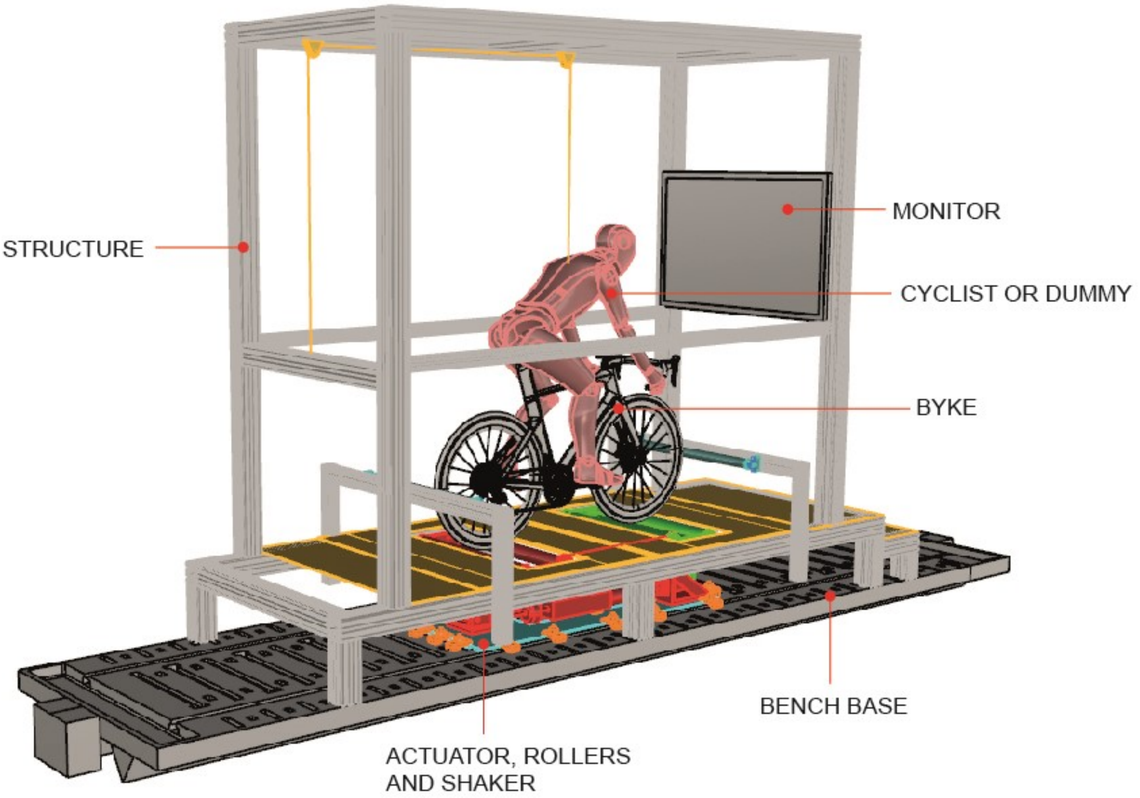

Figure 3.47:Overall system view

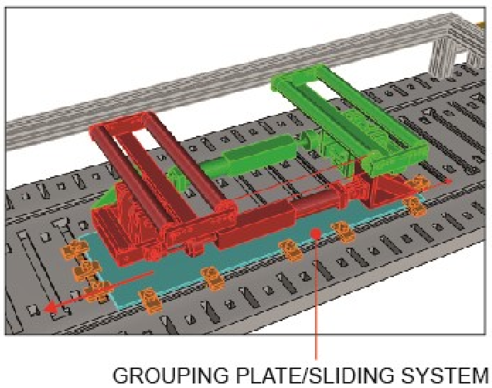

Figure 3.48: Sliding system

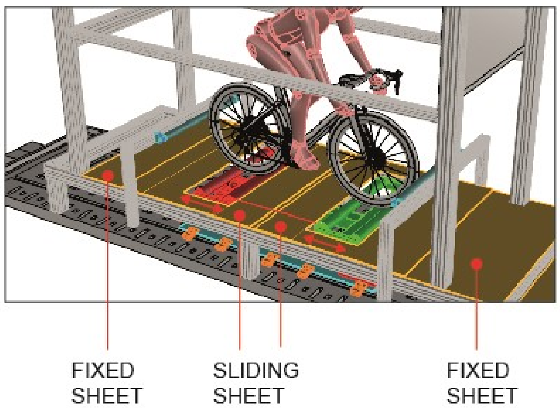

Figure 3.49:Sliding sheets

Main caratheristics: The fulcrum of the system is represented by the actuator cylinders coupled to the articulated parallelogram equipped with rollers. To allow easy modification of the wheelbase, one of the two groups is fixed on a plate that can slide over the base. This speeds up and facilitates operations in case of tests with different means. A structure around the vibrating system grants the safety anchor. A set of socrrevoli plates fixed to the floor allow to obtain an excellent flatness that facilitates getting on and off the bike. A monitor can allow a simulation of a possible route.

Kinematic representation: rocker and parallelogram compared to highlight the improvements obtained.

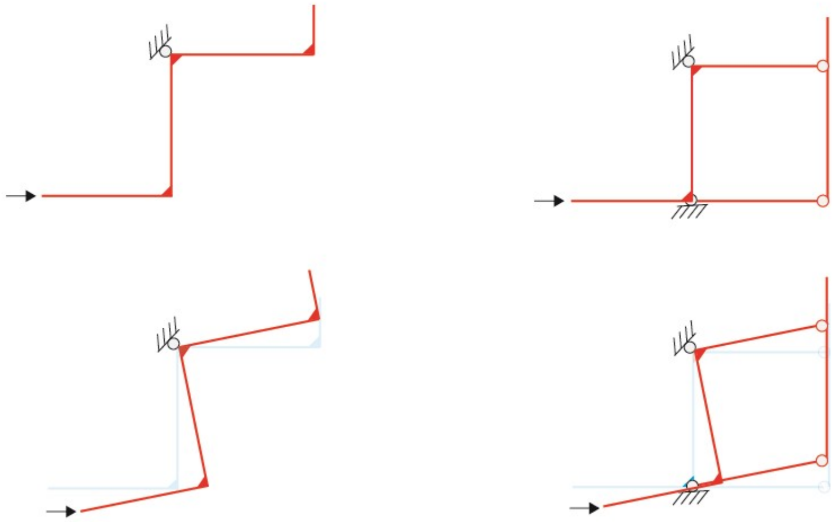

Figure 3.50: Averall schematic representation of rocker and parallelogram

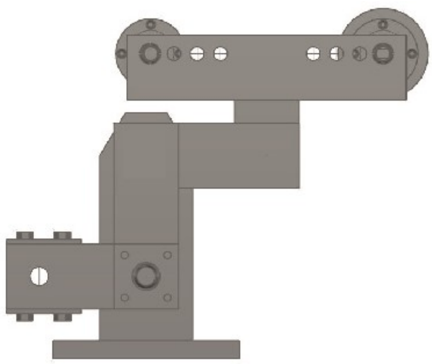

Figure 3.51: Concept 1 in zero position

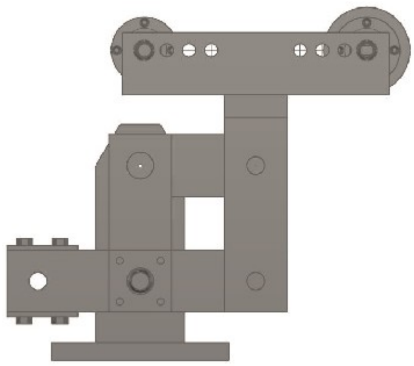

Figure 3.52: Concept 2 in zero positio

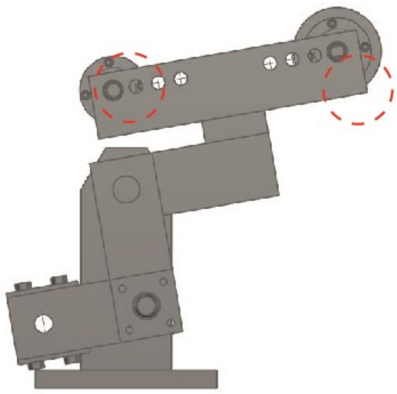

Figure 3.53: Concept 1 at maximum extension

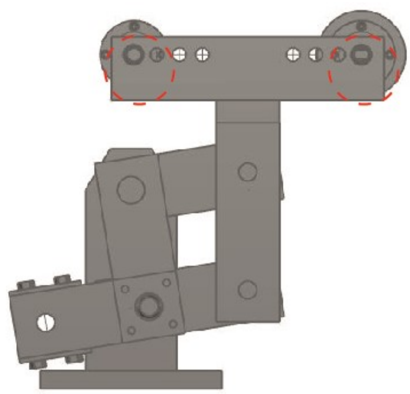

Figure 3.54: Concept 2 at maximum extension

The excursion range of the roller system is +23 mm and -23 mm respect zero position (rocket parallel to base)

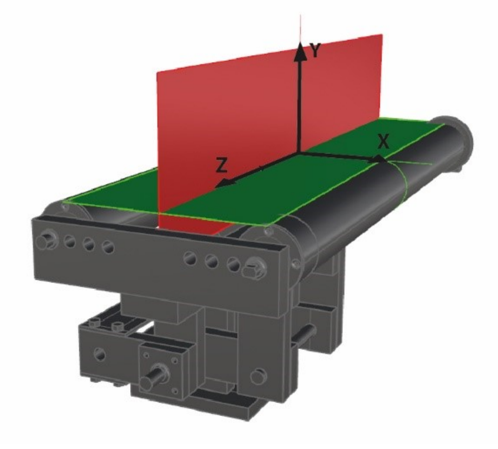

Figure 3.55: Definition of the reference system

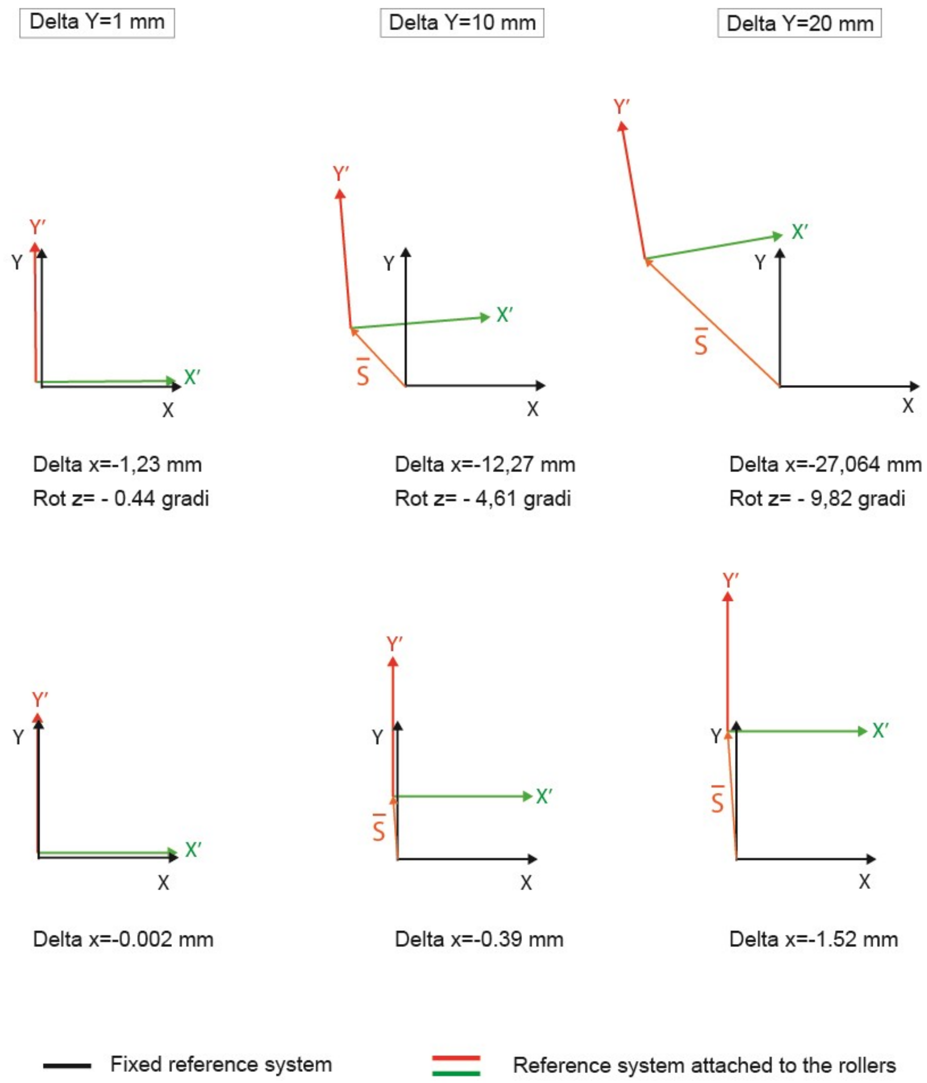

Figure 3.56: Comparison of the fields of motion

It is possible to note that even for displacements of the order of 1 mm the deviation in x is of the same order of magnitude. This may have little influence in the case of small movements but can be influential in the case of using two shakers and for shifts of the order of 10mm that can simulate, for example, walking on cobblestones. For this condition, the influence of the rotation of the system is also highlighted.

Main improvements obtaineble:

- Greater solicitation realism due to two shaker (front and rear wheel)
- Possibility of imposing displacements in major y direction
- Less influence of the horizontal displacement component
- No angular deviation (rotation)

Parts:

MATERIALS BASE LIST					
Num.	Description	Material	Quantity	Mass	Code/designation
01	Profile 25X70	Alluminum 6082	5m	4,73 kg/m	-
02	Steel drawn bar D=20mm	Steel C40	2,5m	2,5Kg/m	-
03	Steel drawn bar D=30mm	Steel C40	1m	5,5kg/m	-
04	Steel plate	-	-	-	-
05	Hollow cylinders D= 75 d=50	Alluminum 6082	3,5m	6,62 kg/m	-
06	Single row spherical roller bearings - skf	Steel	20	0,038 Kg/Pz.	61904
07	Single row cylindrical roller bearings	Steel	4	0,2 Kg/Pz.	NUP 206EC
08	Locking nuts M20x1	Steel	20	0,064Kg	KM tipo DIN 981
09	Locking nuts M30x1,5	Steel	4	0,12Kg	KM tipo DIN 981
10	Uniball M20 internal thread	-	2	0,49 kg/Pz.	DIN ISO 12240-4 serie K
11	Screw M10 x40	Steel	50	-	-
12	Nuts	Steel	40	-	-
13	Grooved profiles Bosh Rexroth	Alluminum	-	-	-

Table 3.57 Table of materials used:

3.4.6 POWER BALANCE DURING PEDALING

While pedaling the atlethe must produce a power able to win resistence due to many factors. An energy and power balance, symplified in some aspects is shown below:

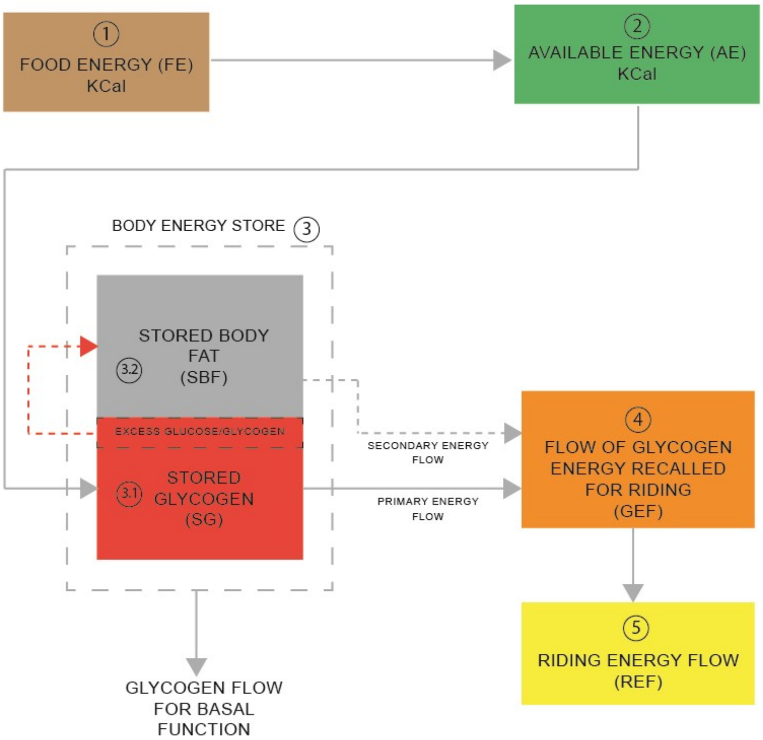

Figure 3.58: Illustrative scheme of the conversion and metabolic flows of energy

- 1 FOOD ENERGY: energy from food in terms of calories available before digestion or caloric intake
- 2 AVAILABLE ENERGY: energy available after digestion of the food introduced
- 3 BODY ENERGY STORE: total accumulation of energy that the human body has available in the form of fats and glycogen to complete the vital and physical functions in the form of movement
- 3.1 STORED GLYCOGEN: energy stored in the form of glycogen in the muscles and liver. The excess amount is stored in the form of fat in case of exceeding caloric intake respect energy requirement. Glycogen is a primary energy resource used.
- 3.2 STORED BODY FAT: body fat available as a form of secondary energy once glycogen has run out or used in the event of a caloric intake below the energy requirement
- 4 FLOW OF GLYCOGEN: energy in the form of flow recalled in case of physical effort. In term of flow is a power
- 5 RIDING ENERGY FLOW: net mechanical energy made available by the conversion of the recalled energy (glycogen) that the human body makes available for the athletic gesture as a whole.

$$Available\ energy\ (AE) = \eta_{trasformation} \cdot Food\ energy\ (FE) \tag{3.5}$$

$\eta_{trasformation}$ = metabolic, biochemical and energy conversion efficiency between food and energy available for the human body

$$Riding\ energy\ (RE) = \eta_{conversion} \cdot\ Glicogen\ energy\ (GE) \tag{3.6}$$

$\eta_{conversion}$ = metabolic efficiency between chemical and mechanical energy. Biochemical processes, muscles, articulation

Below is a useful diagram for determining metabolic power:

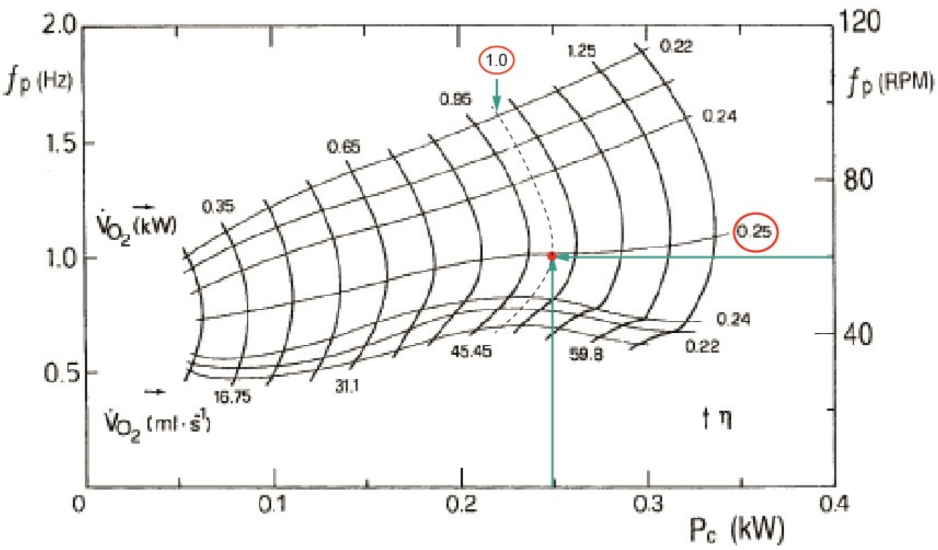

Figure 3.59: illustrative scheme of the conversion and metabolic flows of energy Pedaling frequency (in Hz on left ordinate,in RPM on the right ordinate) as a function of power mechanics developed in cycling. In addition, lines of iso-metabolic power to allow the assignment of a data fficiency value at any point on the graph)

From the graph we go back to the metabolic power (GEF) which depends on many factors but which can be determined in the following way which however depends on multiple factors:

- Trace the vertical line of the PC power (To the pedals) eg 0.3 Kw
- The pedaling frequency is identified as 60 Rpm
- The efficiency curve (η conversion) = 0.25 is identified
- The metabolic isopotence curve is found. In this case 1.0 Kw

The pedaling on a test bench, while taking place with excellent realism, does not reflect in its entirety all the elements that concur during the real case. The diagram below shows the main quantities that intervene in the case of road racing and in the case of use of the test bench:

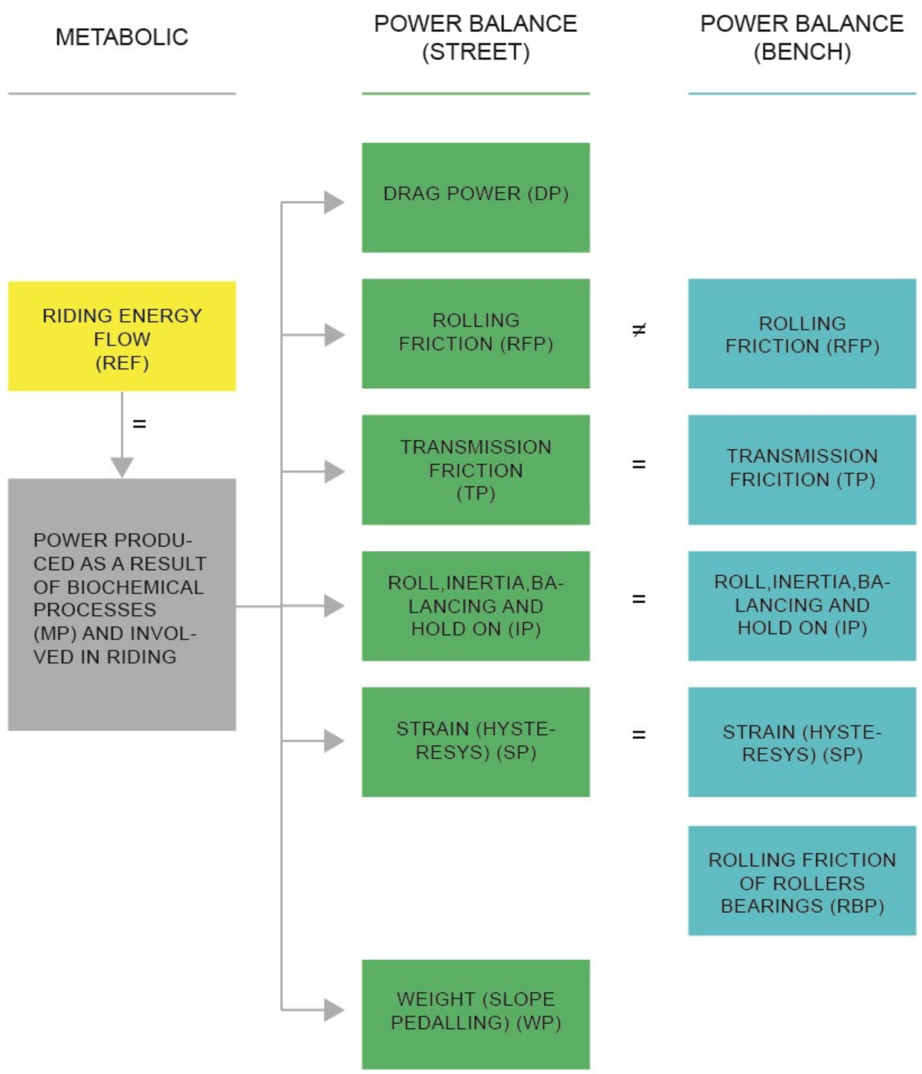

Figure 3.60: Power balance (street and roller bench 2.0)

Metabolic power balance (street) MP = DP+RFP+TP+IP+SP+(WP) (3.7)

Metabolic power balance (bench) MP = RFP+TP+IP+SP+RBP+(WP) (3.8)

Each contribution is described below with the reference equations and relative assumption for the calculation of the corrispondent power:

- ***Drag (DP)***: Is the power dissipated due to aerodynamic loss.

$$R = \frac{1}{2} * \rho * A * C_x * V^2 \tag{3.9}$$

$$\rho = air\ density\ ; A = frontal\ surface\ ;\ C_x = drag\ coefficient;\ V = speed$$

Valori tipici dell'Area Frontale **m** in m^2
0,5 statura superiore a 1,90 e peso 85 kg
0,475 statura tra 1,85 e 1,90 peso 80 kg
0,45 statura tra 1,85 e 1,80 e peso 75 kg
0,425 statura tra 1,80 e 1,75 e peso 70 kg
0,4 statura tra 1,75 e 1,70 e peso 65 kg
0,375 statura tra 1,70 e 1,65 e peso 60 kg
0,35 statura inferiore a 1,65 e peso 55 kg

Figure 3.61: Tipical frontal area

Composizione corporea
+0,025 ogni 5 kg in più del peso indicato
-0,025 ogni 5 kg in meno del peso indicato
Posizione in bicicletta
+0,05 se si usa una posizione molto alta in bici
+0,025 se si usa una posizione alta in bici
-0,025 se si usa una posizione bassa in bici
-0,05 se si usa una posizione molto bassa in bici

Figure 3.62: Frontal area increments

Valore tipico del coefficiente aerodinamico
Ciclista medio **0,800**
Fattori che influnzano il coefficiente aerodinamico
-0,05 in caso di ruote aerodinamiche (a razze o lenticolari)
-0,05 se utilizziamo una prolunga del tipo "Spinaci" o simile (posizione a braccia chiuse)
-0,012 se utilizziamo un abbigliamento aerodinamico (casco, body ..)
+0,012 se utilizziamo un abbigliamento largo, "svolazzante"
-0,012 se si usa una bici con accorgimenti aerodinamici
+0,037 se si usa una posizione molto alta e larga in bici
+0,025 se si usa una posizione molto alta in bici
+0,012 se si usa una posizione alta in bici
-0,012 se si usa una posizione bassa in bici
-0,025 se si usa una posizione molto bassa in bici

Figure 3.63: Aerodinamic coefficient

- ***Rolling friction (RFP)***: Is the power due to rolling friction between wheel and asphalt. In this estimation we assume that the contact between wheel and aluminium rolls does not dissipate the same power than the contact between wheel and asphalt. The roughness of the asphalt is higher than of the rollers and variate with the street surface condition. The contact between rolls and tires is not variable and generate a different footprint (respect a plane condition) and take place in four point instead of two.

$$Rv = \frac{Fn}{R} * N_v \qquad (3.10)$$

$$F_n = normal\ force\ ;\ N_v = rolling\ friction\ coefficient\ (tyre/street)$$

$$N_v = f(street, tyre\ type.pressure, wheel\ diameter, load);\ supposed = 0.003$$

- ***Transmission friction (TP)***: Amount of power wasted due to the transmission of power from pedals to rear hub. Is mainly due to drive chain, toothed crowns and pinions. Common value of the efficiency is around 98%. It depend on the transmission ratio used.

- ***Roll, inertia, balancing and hold on (IP)***: The biomechanics of pedaling reveals that during the cyclist it dissipates power through the movement of the body to be able to pedal, balances itself through the arms and performs compensatory micro movements to be able to push on the pedals. It is estimated as %(TP(transmission friction power)+DP(drag power)+RFP(rolling friction power)) + %WP

- ***Strain (SP)****:* The bicycle frame deforms under the action of pedaling due to loads. this deformation work in the elastic field over time is associated with a dissipated power (power loss due to hysteresis). It is grouped with IP power

- ***Rolling fricion of bearing's rollers (RBP)***: The rollers rotate thanks to bearings that rotating under load generate a resistant torque. The power dissipated at a given load depends on the rotation speed. Skf simplified equation is used for a first estimation.

$$Mv = 0{,}5 * \mu * F * d \tag{3.11}$$

$$\mu = Rolling\ friction\ coefficient;\ d = bearing\ hole\ diameter;\ F = force\ on\ rolls$$

$$\mu = 0{,}0015$$

- ***Weight (WP)***: Is the amount of power spent due to the parallel (to the road) component of the weight that must be win to proceed at a given speed. In this work is not a contemplate contribute because is not possible with the actual bench.

NOTE: consideration are valid for similar transmission ratio

Exluding the case of power spent to uphill pedaling that is a case not possible with the actual bench, to maximize realism the brake system to compensate the power balance must be able to dissipate the difference between drag power(absent in the banch) and the power disspiate during the pedaling due to rolling friction in the roller bearings.

BSD= Brake system dissipation (power)

DP= Drag power

RBP= Rolling friction of rollers bearing power

RFP= Rolling friction power (street or bench)

$$BSD = DP - RBP + RFP_{STREET} - RFP_{BENCH} \tag{3.12}$$

Simulation of the resistant effort: The table above report the power that must be compensated due to abscence of drag force. Considering the presence of rolling friction due to bearings in rollers, for each speed from ten to thirty Km/h has been calculated the relative power to dissipate. To maximize realism durind pedaling must be considered a sistem able to dissipate the power indicate in the table for each speed. A power meter can be used to confront results and assumption.

Using figure. 3.54 can be estimated metabolic power involved during the test (blue column)

				Power (Watt)											
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Speed (Km/h)	Speed (m/s)	ω_{rolls} (rad/s)	Round/min (rolls)	Metabolic power	Roll power + strain (IP)	Power (PC = REF) (Watt)	Transmission loss power (TP)	Drag power (Watt)(DP)	Roll power (street) (RFP)	Roll power (rolls)(RFP)	Roller bearing power of rolls (RBP)	Brake power	Brake torque Nm	Normal force (brake)	Regulation displacement (metri)
10	2,78	79,37	758,27		1,55	32,56	0,61	8,23	22,17	22,17	1,22	7,01	0,09	5,890	0,015
11	3,06	87,30	834,09		1,80	37,85	0,71	10,95	24,38	24,38	1,34	9,61	0,11	7,342	0,018
12	3,33	95,24	909,92		2,08	43,72	0,82	14,22	26,60	26,60	1,46	12,76	0,13	8,932	0,022
13	3,61	103,17	985,74		2,39	50,23	0,94	18,08	28,82	28,82	1,58	16,50	0,16	10,661	0,027
14	3,89	111,11	1061,57		2,73	57,42	1,07	22,58	31,03	31,03	1,71	20,88	0,19	12,527	0,031
15	4,17	119,05	1137,40		3,11	65,36	1,22	27,78	33,25	33,25	1,83	25,95	0,22	14,532	0,036
16	4,44	126,98	1213,22		3,53	74,09	1,38	33,71	35,47	35,47	1,95	31,76	0,25	16,675	0,042
17	4,72	134,92	1289,05		3,98	83,67	1,56	40,44	37,68	37,68	2,07	38,37	0,28	18,957	0,047
18	5,00	142,86	1364,88		4,48	94,14	1,76	48,00	39,90	39,90	2,19	45,81	0,32	21,377	0,053
19	5,28	150,79	1440,70		5,03	105,57	1,97	56,45	42,12	42,12	2,31	54,14	0,36	23,935	0,060
20	5,56	158,73	1516,53		5,62	118,00	2,20	65,84	44,33	44,33	2,44	63,41	0,40	26,631	0,067
21	5,83	166,67	1592,36		6,26	131,49	2,46	76,22	46,55	46,55	2,56	73,66	0,44	29,466	0,074
22	6,11	174,60	1668,18		6,96	146,09	2,73	87,64	48,77	48,77	2,68	84,96	0,49	32,438	0,081
23	6,39	182,54	1744,01		7,71	161,85	3,02	100,14	50,98	50,98	2,80	97,34	0,53	35,549	0,089
24	6,67	190,48	1819,84		8,52	178,83	3,34	113,78	53,20	53,20	2,92	110,85	0,58	38,799	0,097
25	6,94	198,41	1895,66		9,38	197,08	3,68	128,60	55,42	55,42	3,05	125,56	0,63	42,187	0,105
26	7,22	206,35	1971,49		10,32	216,65	4,05	144,66	57,63	57,63	3,17	141,49	0,69	45,712	0,114
27	7,50	214,29	2047,32		11,31	237,60	4,44	162,00	59,85	59,85	3,29	158,71	0,74	49,377	0,123
28	7,78	222,22	2123,14		12,38	259,98	4,85	180,67	62,07	62,07	3,41	177,26	0,80	53,179	0,133
29	8,06	230,16	2198,97		13,52	283,83	5,30	200,73	64,28	64,28	3,53	197,20	0,86	57,120	0,143
30	8,33	238,10	2274,80		14,72	309,22	5,77	222,22	66,50	66,50	3,65	218,57	0,92	61,199	0,153

Table 3.64 Table of estimated powers.

Power balance mechanism:

Power dissipation mechanism: To follow will be presented two brake systems able to dissipate the power necessary to balance the affort (road vs bench) and increase realism

Brake – CONCEPT1

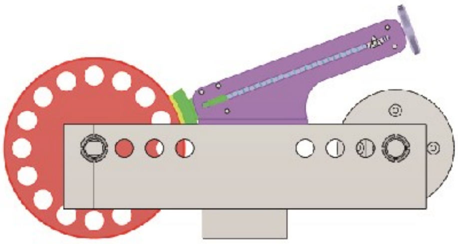

Figure 3.65: Lateral wiev of concept 1

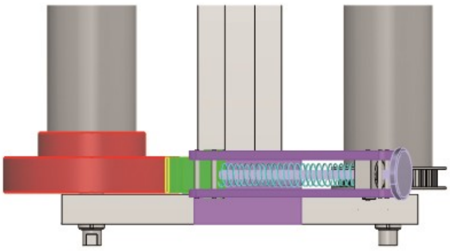

Figure 3.66: Top wiev of concept 1

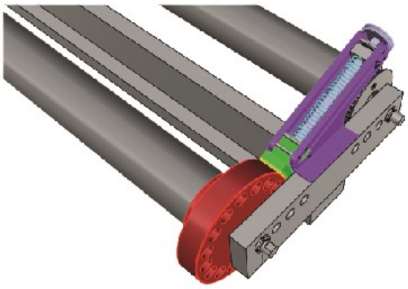

Figure 3.67: Anterior prospective wiev of concept 1

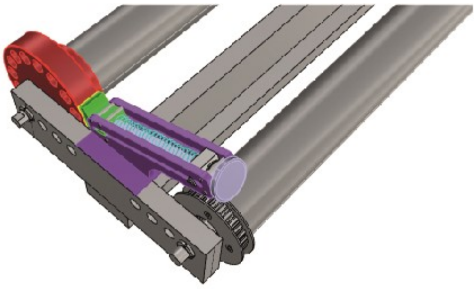

Figure 3.68: Posterior prospective wiev of concept 1

Brake – CONCEPT2

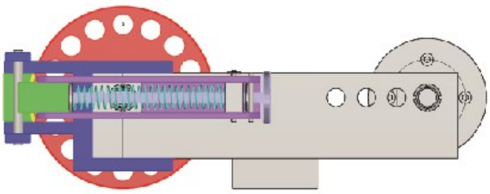

Figure 3.69: teral wiev of concept 2

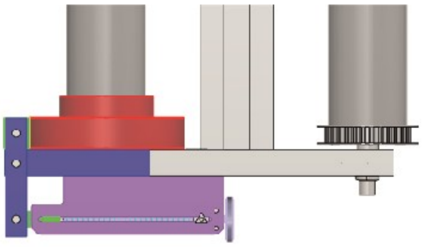

Figure 3.70: Top wiev of concept 2

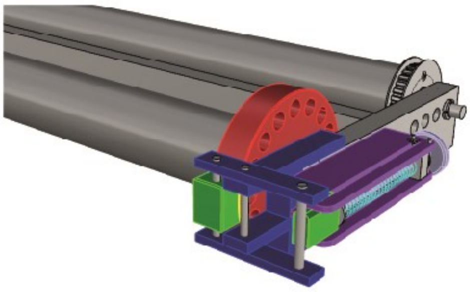

Figure 3.71: Anterior prospective wiev of concept2

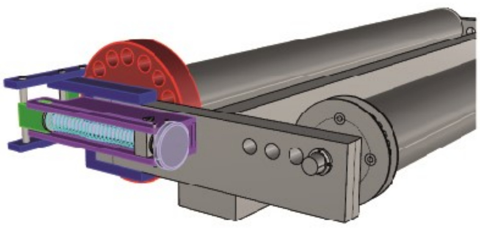

Figure 3.72: Posterior prospective wiev of concept 2

4 INDOOR TESTS

In the following chapter, the behavior of a shock-absorbed stem used to reduce vibrations to the handlebar will be studied and and compared with a rigid stem.

Main topics are:

- *Research question (Hyp redshift reduce A rms)*
- *Materials*
- *Method used - MatMethod used to generate vibrations*
- *Results (grafici)*
- *Discussion*

4.1 RESEARCH QUESTION

The different types of roads defined (A, B, C) by iso 8608 are characterized by the RMS of accelerations and increasing displacements according to a specific variation of PSD in relation to Wave lenght. The question to be answered is whether the Redshift shockstop shock absorber stem, is able to reduce the acceleration value (RMS value) compared to those characteristics of the different types of roads for a rigid stem.

4.2 MATERIALS

Test bench: Double roller shaker 2.0 with thiner rolls:

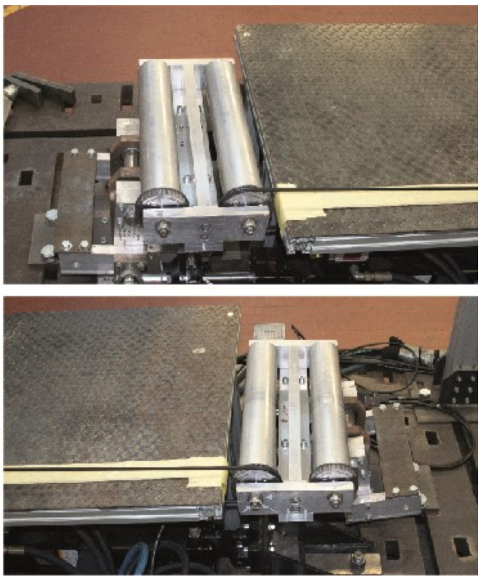

Figure 4.1: Test bench roller shaker 2.0

Figure 4.2:Rollers detail

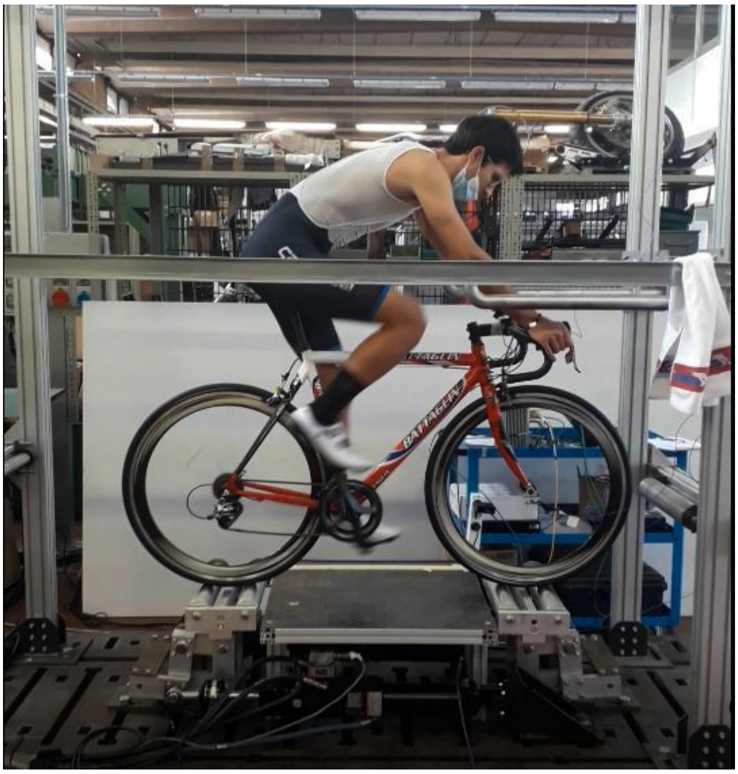

Figure 4.3: Riccardo Donato pedaling during test session

Rigid stem: Deda zero 100 alluminum rigid stem for bike,figure(4.5)

Suspension stems: ***Redshift shockstop*** fig(4.4) In line with the purpose for which the test bench was created, it was decided to test a component for the reduction of vibrations transmitted to the handlebar (Bike Stem Redshift Shockstop). It consists of a stem to replace the rigid one. The stem can rotate with respect to the fixing element to the steering head. its free rotation is hindered by the presence of elastomers which are compressed during rotation and stress. The stiffness is regulated and through the use of special elastomers supplied. It is a reversible stem with 6 degree inclination and it’s alluminum made.

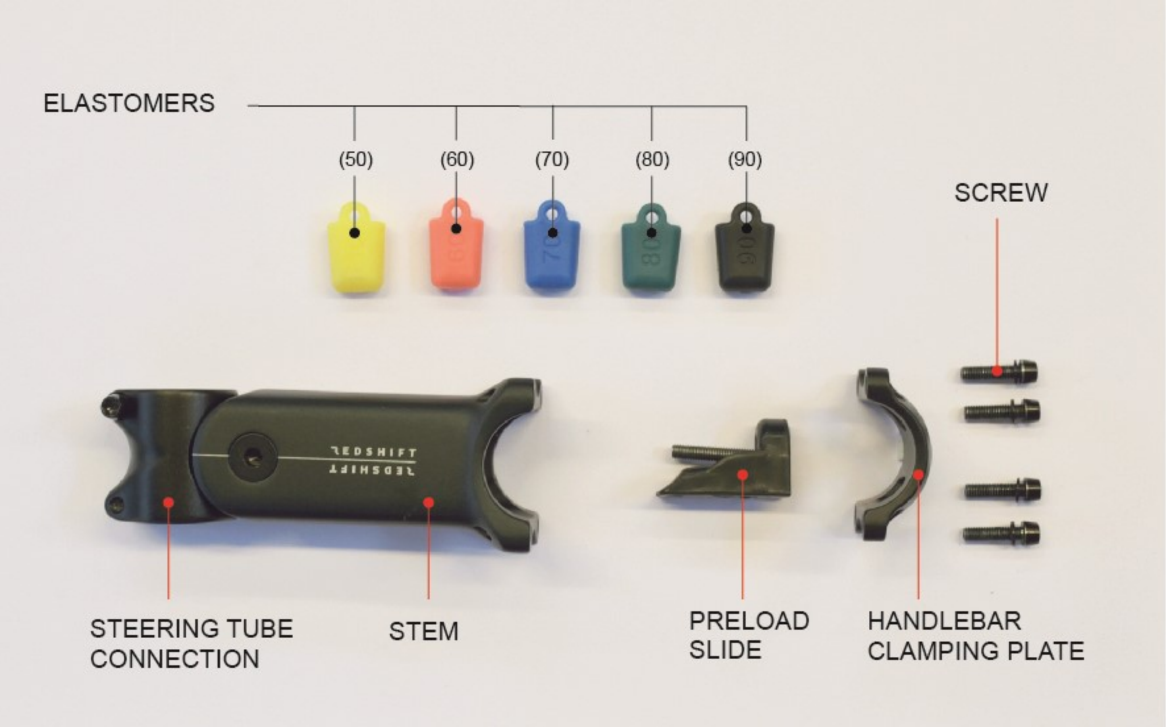

Figure 4.4: Redshift shockstop main components

Figure 4.5: Deda rigid stem

Figure 4.6: Redshift shockstop suspension stem

Figure 4.7: Suspension stem with elastomers

Figure 4.8: Suspension stem rought components

Sensors: Uni axial accelerometers for Rolls and Wheel hub and tri-axial accelerometers for stem:

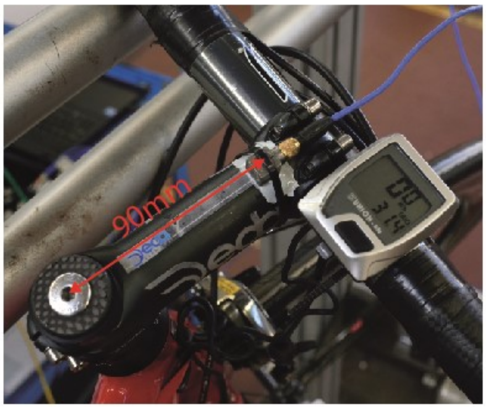

Figure 4.9: Tri-axial accelerometer on Rigid stem

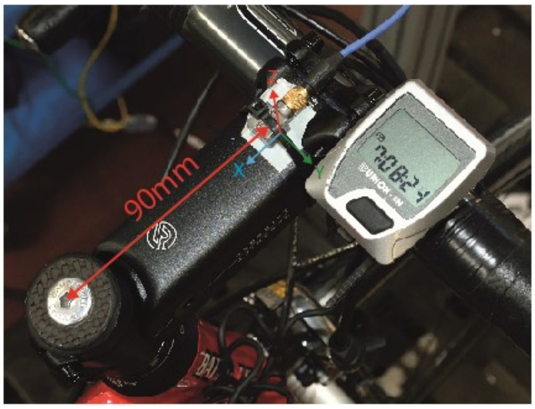

Figure 4.10: Tri-axial accelerometer on Suspension stem

Figure 4.11: Hub acelerometer

Figure 4.12: Roller accelerometer

4.3 METHODS [8]

The method that will be used to carry out the comparative test is based on Random generated approach (figure 4.14) and is divided into the following phases:

- Generation of a procedure to reproduce the roads of typology A, B, C through the test bench

- Sensing by accelerometers on the wheel hub, stem and on the rollers

- Signal acquisition using a rigid stem and two different postures (Posture1 and Posture2)

- Acquisition using three different settings for the suspension stem (SOFT, MEDIUM, HARD) using Posture1 and Posture2

4.3.1 INPUT SIGNAL GENERATION

In the first part of this chapter, the two methods used to generate real vibrations on the indoor test bench are presented. These methods were used and introduced by Enrico Girlanda [5] in order to reproduce real external road profiles in an indoor environment. The first method is the Inverse Transfer function approach, a method that starts from the outdoor tests data and the calculation of bicycle Transfer Function to generate input displacements for both actuators. The second method is the Random Generator approach and it is based on the application of a random displacement generator. The idea is to generate random displacements for the two actuators, giving as input bicycle speed and road roughness. These two methods are reassumed in the following diagram:

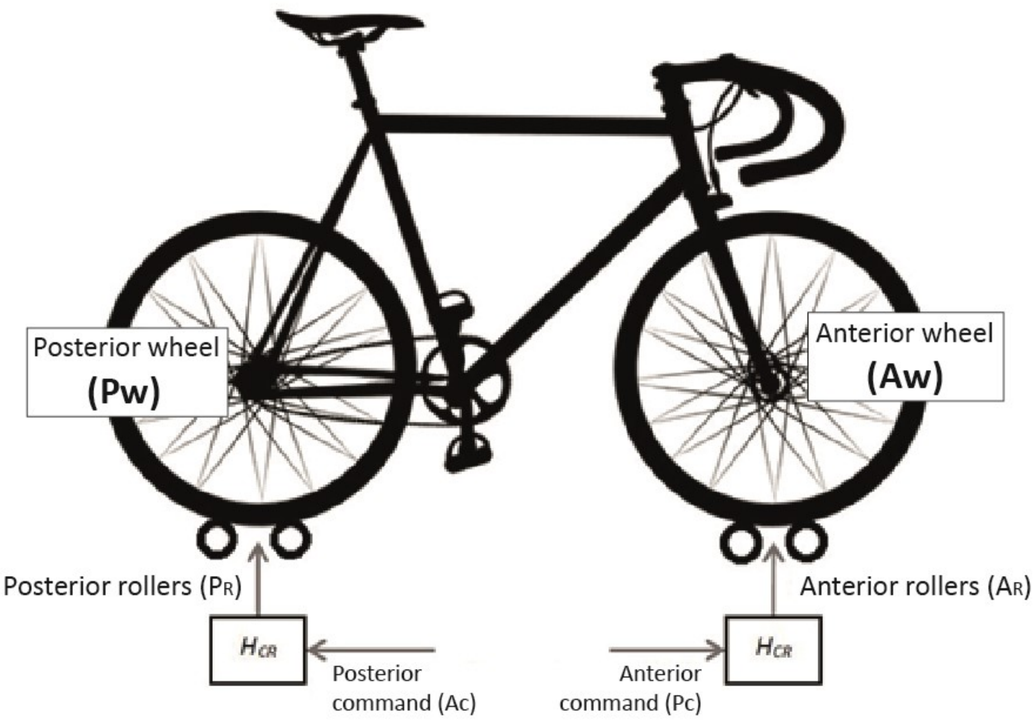

Figure 4.13 Reference scheme for main quantities

The general nomenclature is reported below:

- *Transfer Function Command-Rollers (H_{CR}):* Transfer Function between actuators' input command and acceleration measured on rollers' frame;
- *Transfer Function Rollers-Wheel (H_{RW}):* Transfer Function between acceleration measured on - rollers' frame and the one measured on bicycle wheel axis;
- *Transfer Function Command-Wheel (H_{CW}):* Transfer Function between actuators input command and acceleration measured on bicycle wheel axis.
- *$a_{WF}(t)$:* Wheel axis acceleration measured during Field Tests in time domain;
- *$A_{WF}(f)$:* Wheel axis acceleration measured during Field Tests after Fast Fourier Transform, in frequency domain;
- *$a_C(t)$:* Actuator Command acceleration during frequency sweep, for Transfer Function definition;
- *$A_C(t)$:* Actuator Command acceleration during frequency sweep, for Transfer Function definition after Fast Fourier Transform, in frequency domain;
- *$a_W(t)$:* Wheel axis acceleration during frequency sweep, for Transfer Function definition;
- *$A_W(t)$:* Wheel axis acceleration during frequency sweep, for Transfer Function definition after Fast Fourier Transform, in frequency domain;
- *$H_{CW}(f)$:* Transfer Function between actuator Command and Wheel axis, defined during frequency sweep;
- *iTF:* inverse Transfer Function, operation to define $A_{CV}(f)$ from $A_{WF}(f)$;
- *RGa:* Random Generator algorithm;
- *D_{-2}:* Double integration to define displacements from accelerations;
- *$A_{CV}(f)$:* Actuator Command Virtual acceleration in frequency domain;
- *$a_{CV}(t)$:* Actuator Command Virtual acceleration after Inverse Fast Fourier Transform, in time domain;
- *$z_C(t)$:* Actuator Command displacement in time domain;
- *$a_{WI}(t)$:* Wheel axis acceleration measured during Indoor Tests in time domain;
- *PSD_{Welch}:* Operation to obtain PSD from a time domain signal;
- *$PSD_{WIF}(f)$:* Wheel axis acceleration PSD measured during Indoor Tests with inverse Transfer Function approach, in frequency domain;
- *$PSD_{WIR}(f)$:* Wheel axis acceleration PSD measured during Indoor Tests with Random Approach, in frequency domain;
- *$PSD_{WF}(f)$:* Wheel axis acceleration PSD measured during Field Tests, in frequency domain.
- *EI:* Error Index between Field test PSD and inverse Transfer Function Approach PSD;
- *DI:* Deviation Index between Field test PSD and Random Approach method PSD.

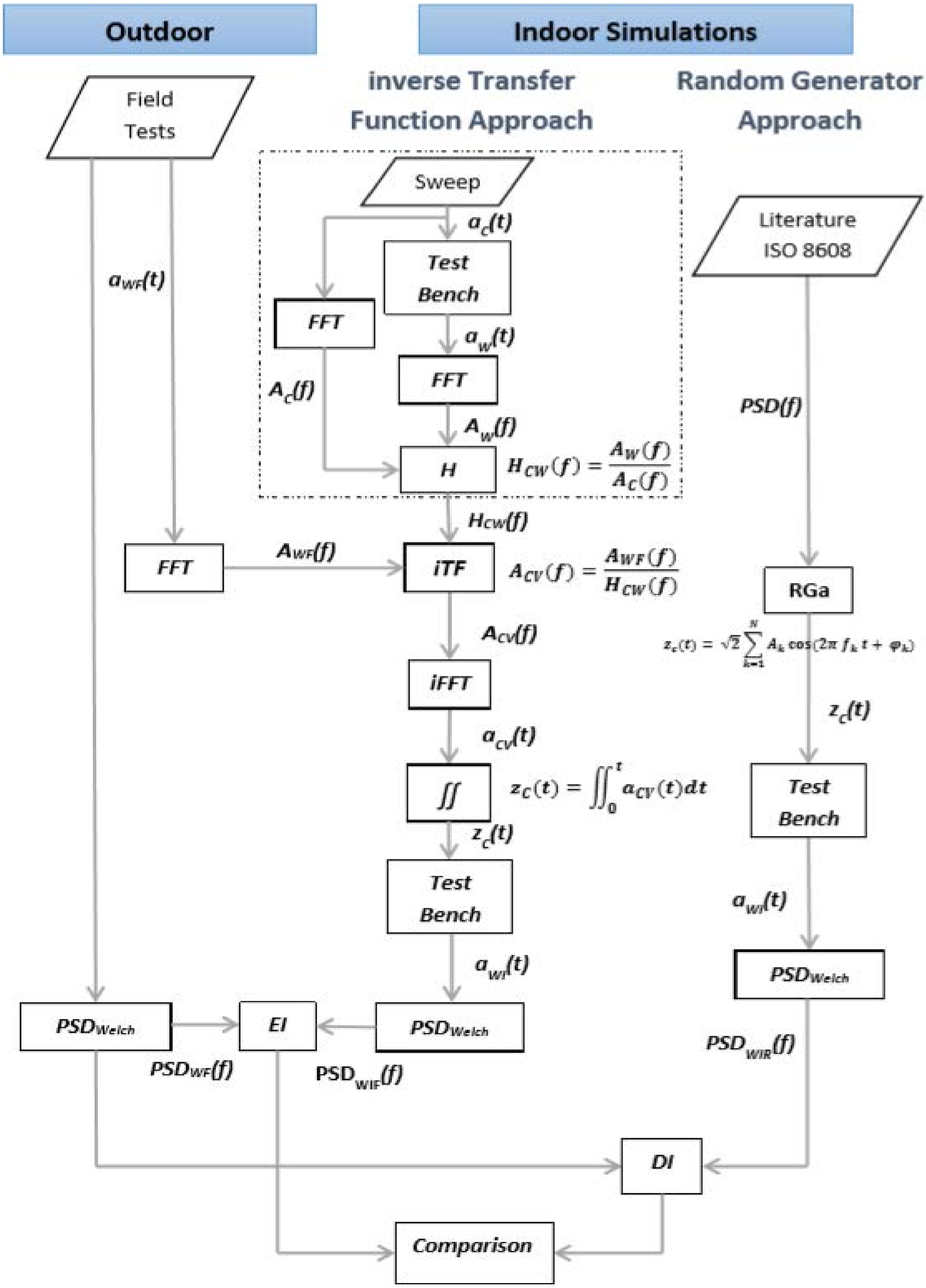

Figure 4.14: Flowchart of the methods realized by Enrico Girlanda

Random approach is a method that generates random displacement profiles of the type of road given. This approach starts from the knowledge of *ISO 8608* [8] that explains in terms of roughness the different types of road profiles. This approach needs two variables: bicycle speed and the type (roughness) of road. The scale defined goes from *A* (Smooth road) to *H* (Rough road).

Each road profile is represented by Power Spectral Density (PSD) of road displacement. An estimate of the degree of roughness of road can be made by the index $G_d(n_0)$ and it represents a band between two lines in the logscale diagram. In the following diagram there are PSD values for different road profile in terms of space coordinates.

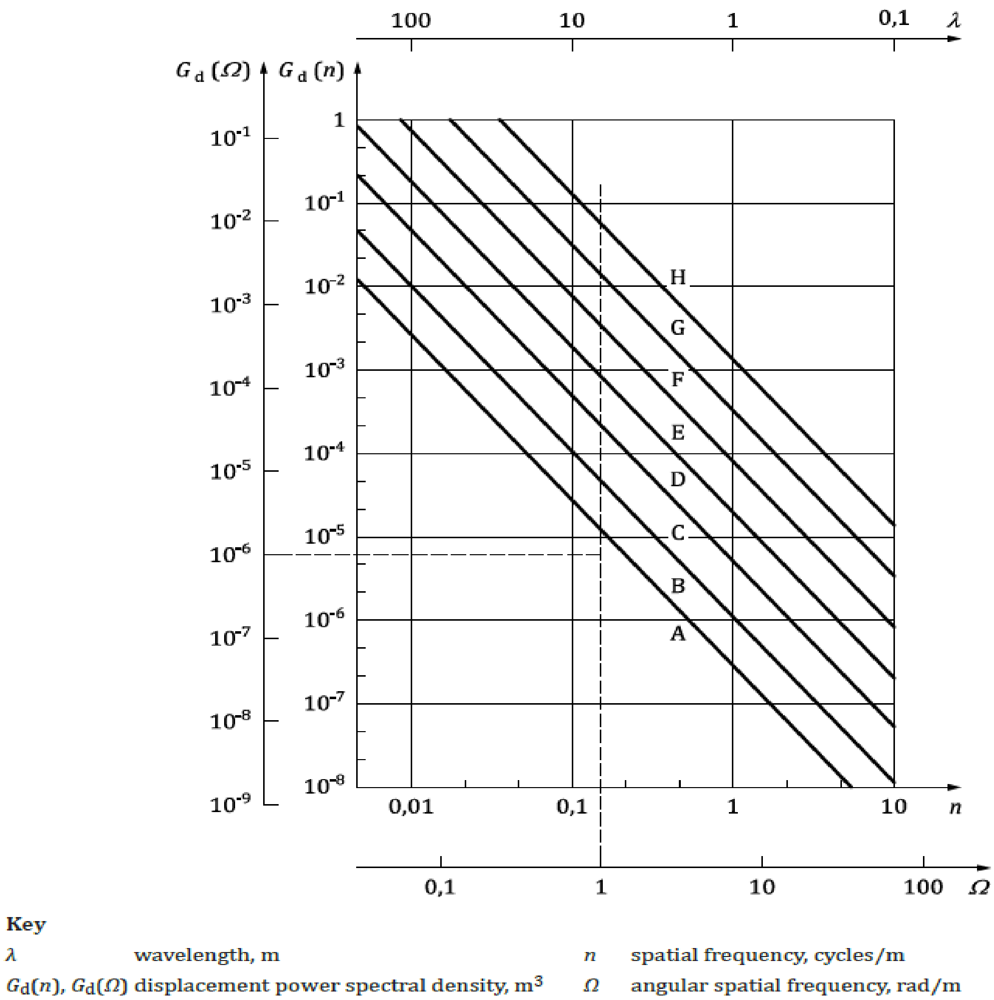

Figure 4.15: Road profile PSD diagram.

In *ISO 8608:2016*, equations are used to define a random signal basing on the PSD of displacement of each road. The following one gives a method to define Root Mean Square of road profile. The idea is to define G_d in order to obtain the real road profile RMS.

$$\int_{n_{in}}^{n_{fin}} G_d(n)\, dn = RMS^2 \tag{4.1}$$

ROAD CLASS	$G_d(n_0)\ [10^{-6}\ m]$	DESCRIPTION
A	16	Airport runways and superhighways
B	64	Normal pavements
C	256	Unpaved roads and damaged pavements
D	1 024	Rough unpaved roads
E	4 094	Enduro tracks
F	16 384	Off-road tracks
G	65 536	Rough off-road tracks
H	262 144	Simulation purpose only

Table 4.16: Road Classes [13] – [14]

In this equation n_{in} is the lowest spatial frequency considered and n_{fin} is the highest one. These values are respectively $0.011\,\frac{cycles}{m}$ and $2.83\,\frac{cycles}{m}$, according to literature data [8].

Using *ISO 8608*, a random displacement can be generated maintaining constant the RMS: in the following equation amplitude depends on the road profile type while frequency changes from a maximum to a minimum, in the previous defined range. All these equations are used considering a constant PSD velocity [2].

$$z(x) = \sqrt{2} \sum_{k=1}^{N} A_k \cos(2\pi\, n_k\, x + \varphi_k) \tag{4.2}$$

In this equation: $z(x)$ is the ground vertical displacement, x is the bike horizontal displacement on the road; $A_k = \sqrt{G_d(n_k)\,\Delta n}$ where Δn is the space frequency increment, φ_k is the signal phase, a number in the range of $[0, 2\pi]$, N depends on space frequency increment and n goes between n_{in} and n_{fin}.

To generate a displacement reproducible in indoor tests, displacement profile has to be changed from space to time domain. The relation between these two signals is correlated to this equation:

$$f = v * n \tag{4.3}$$

In this expression f is the time frequency [Hz], v is bicycle speed [m/s] and n is the space frequency. The equation that can be used to generate a time-based signal is:

$$z(t) = \sqrt{2}\sum_{k=1}^{N} A_k \cos(2\pi\, f_k\, t + \varphi_k) \tag{4.4}$$

where $A_k = \sqrt{G_d(f_k)\,\Delta f}$ and $\Delta f = \Delta n * v$.

```
clearvars

A = 16E-6;
B = 64E-6;
C = 256E-6;
D = 1024E-6;
E = 4094E-6;
F = 16384E-6;
G = 65536E-6;
H = 262144E-6;

SR = 100;   % Insert the sample rate of the signal (it should be the same of MTS system)
% NB: MTS sample rate is defined as a complete cycle and not just a ramp
% you have to divide it by two
v_kmh = 19;     % Insert the speed in km/h
v = v_kmh/3.6;  % Calculates the speed in m/s
df = 0.01;      % time frequency interval
n_min = 0.2;      % From ISO 8608 (0,011)
n_max = 2.83;
f_in = v*n_min;   % Initial time frequency depending on the speed
f_fin = v*n_max;   % Final time frequency depending on the speed
f = [f_in:df:f_fin];        % Frequency range
F = length(f);
n0 = 0.1;       % From ISO 8608
G_d_n0 = D;     % Insert the type of road you want to simulate (from A to H)
t = [0:1/SR:20];        % Generate time vector in seconds
Ak = sqrt(G_d_n0*v*n0^2./f.^2.*df);
for k=1:F, j=1:length(t);
    z(k,j) = sqrt(2)*Ak(k)*cos(2*pi*f(k)*t(j)+rand*2*pi);
end
disp = sum(z,1);
```

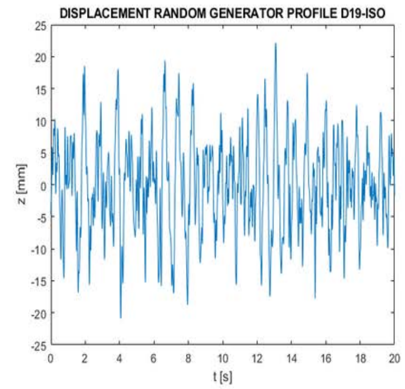

Figure 4.17: Matlab script used to generate command displacements according to ISO 8608; on the right there is the time-based displacement simulating road profile D at the speed of 19 km/h.

Time duration of each run is set to *20* seconds because it is considered enough to let the tester to get familiar with vibrations.

Due to the fact that this method generates different random profiles at each run, for front and rear wheels the same profile has to be used. However this profile has to be shifted due to the wheels distance and the bicycle speed. Using the following formula, phase between the anterior and the posterior wheels displacement can be calculated:

$$p = \frac{d}{v} \tag{4.5}$$

Where p is the phase between front and rear signals, d is wheels distance [m] and v is the bicycle speed [m/s]. For Fantic bike, wheel axes distance is 1,24 metres (in the configuration without loads on the bicycle).

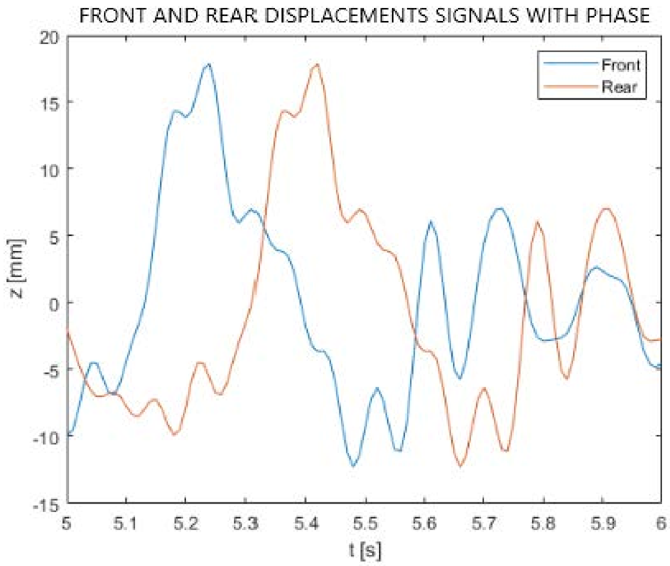

Figure 4.18: Example of road profile D at the speed of 19 km/h.

Displacement profiles defined in this way are also called virtual Displacement Signals and they represent input signals used for the simulation. This method will be used in chapters 5, 6, 7 and 8.

Below it is reported the nomenclature used for the main quantities used for test session:

a_{fw} = Acquired front wheel (Hub) acceleration

a_{st} = Acquired stem acceleretion (Z direction)

a_{fr} = Acquired front rolls acceleration

a_{fw}^{RMS} = Calculated root mean square of front Wheel (Hub) acceleration

a_{st}^{RMS} = Calculated root mean square of stem acceleration

a_{fr}^{RMS} = Calculated root mean square of rolls acceleration

F2 = Acquired force (cylinder 2)

C2 = Acquired stroke (cylinder 2)

$F2^{RMS}$ = Calculated root mean square of force for cylinder 2

$C2^{RMS}$ = Calculated root mean square of stroke for cylinder 2

$H_{CW}(f)$= Transfer Function between actuator Command and Wheel axis, defined during frequency sweep;

$H_{cs}(f)$ = Transfer Function between actuator Command and stem, defined during frequency sweep;

$H_{rw}(f)$ = Transfer Function between rolls and wheel hub

$H_{ws}(f)$ = Transfer Function between wheel hub and stem

$H_{rs}(f)$ = Transfer Function between rolls and stem

$TI(rw) = \frac{a_{fw}^{RMS}}{a_{fr}^{RMS}}$ = Transmission index between rollers and front wheel

$TI(ws) = \frac{a_{st}^{RMS}}{a_{fw}^{RMS}}$ = Transmission index between front wheel and stem

$TI(rs) = \frac{a_{st}^{RMS}}{a_{fr}^{RMS}}$ = Transmission index between rollerer and stem (global transmission index)

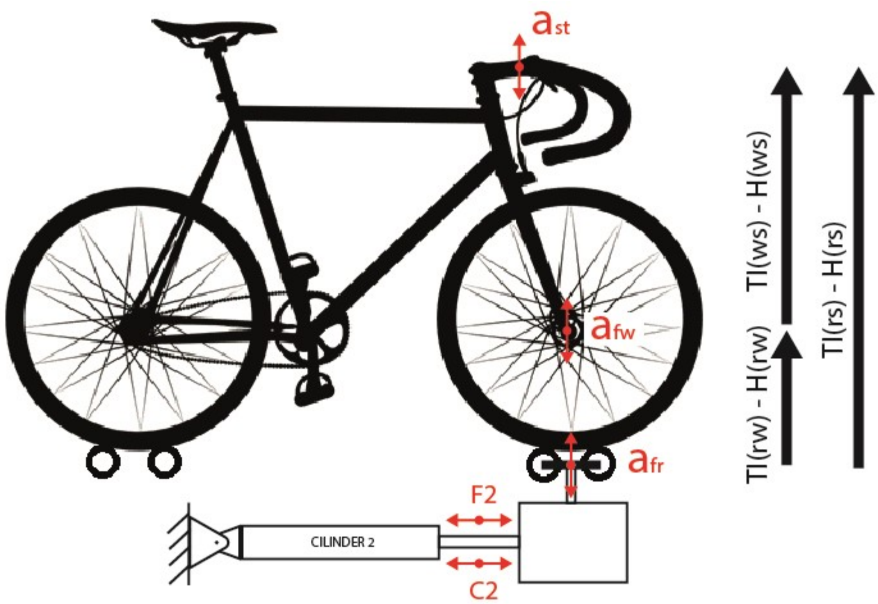

Figure 4.19: Reference scheme for main quantities invovled during test session.

Acquired channels during test sessions:

- Stem: X-Y-Z acelerations
- Hub: Z aceleration (wheel)
- Rolls: Z acceleration
- Load cells: load (N)
- Cylinders: displacement (mm)

4.3.2 INDOOR TEST SESSION WITH RANDOM GENERATOR APPROACH

During the procedure only the types of roads A, B, C were investigated. This is to be able to reproduce levels of severity consistent with the possibility of pedaling on the test bench with a racing bike. For roads with characteristics higher than C, the permissible travel limits for the test bench are exceeded and in fact it is difficult to perform the in-vivo tests.

The method used to produce the random displacement is the random generator approach of (Fig 4.14) according to prescription of ISO 8608

Road type	*Description*	*ZRMS (mm) (displacement)*	*Zmin (mm)*	*Zmax (mm)*	*a RMS (g) (m/s^2)*
A	Airport runways and super highways	0,840	-2,774	3,673	0,238
B	Normal pavements	1,705	-5,058	5,520	0,375
C	Unpaved roads and damaged roads	3,757	-12,432	12,682	0,732

Table 4.20: Main quantities for road A,B,C

Settings and configurations: For the execution of the test campaign, the procedure decribed below was used, relating to a speed of 24 Km/hour. According to equation (4.5) p= 0,15 s

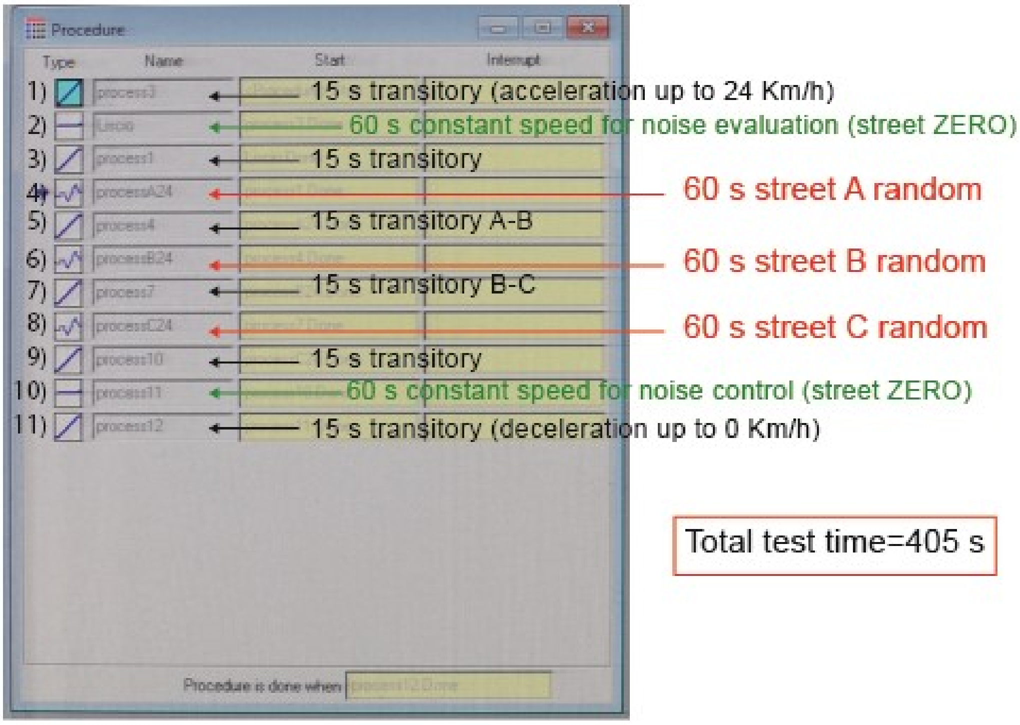

Figure 4.21: Procedure used during test sessions

Posture types: During the test, two typical postures (Figure 4.22 and 4.23) for the racing bike were investigated. Posture1 is typically used while pedaling at cruising speed while Posture2 is used while sprinting.

Figure 4.22: Posture1

Figure 4.23: Posture2

Suspension hardness: The stiffness of the stem is defined according to the following scheme. The average setting refers to the specifications given in the product manual. The setting rigid is referred to the rigid stem.

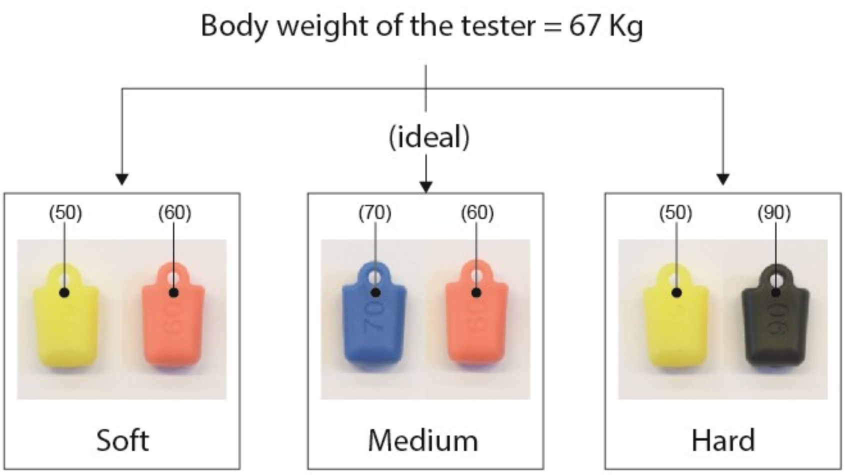

Figure 4.43: Elastomers combinations used during test session

Tester: the tester was Riccardo Donato, a sports cyclist. Height: 1,8 m ; Weight: 67 Kg

Tyre pressure: P= 8.5*10^5 Pa (verified before and after test session)

4.4 RESULTS

The diagram shown represents the variables to be studied and correlated each other and the channels acquired for accelerations, forces and displacements:

Independent variables: These are all the variables that it was decided to investigate and are essentially a choice. For the test we have chosen: **Road surface harshness** (Zero,A,B,C) - **Stem Stiffness** (SOFT,MEDIUM,HARD,RIGID) - **Posture** (HAND POSTURE 1,HAND POSTURE2)

Dependent variables: The dependent variables include the RMS values of the accelerations and the quantities derived from them such as the PSD, the transfer function, the indices based on the relationships between the accelerations and the numerical integrations.

Acquired channels: Cylinder force (F2,F1) – Cylinder dysplacement (S1,S2) – Accelerations (Rolls(Z) ,Hub(Z), Stem(X,Y,Z))

The following **3way within** system represent the variables to study:

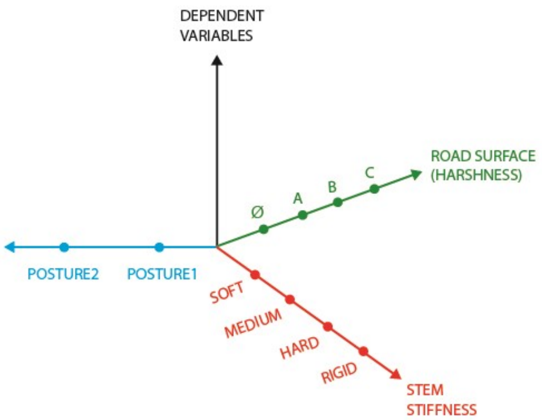

Figure 4.25: Scheme of research quantities and involved variables

The 3 Way within study is dividen in two **2way within**

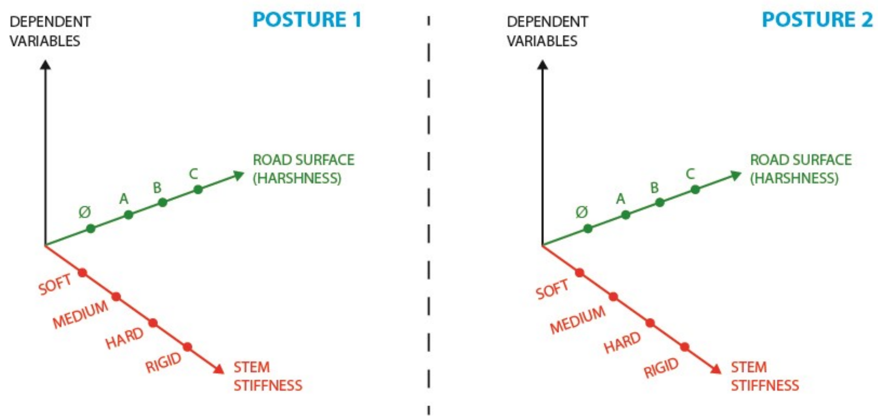

Figure 4.26: Subdivision of the research quantites

4.4.1 Raw data analisys

Street type identification: Below are the data relating to a typical acquisition from which it is possible to identify the different types of roads (ZERO-A-B-C)

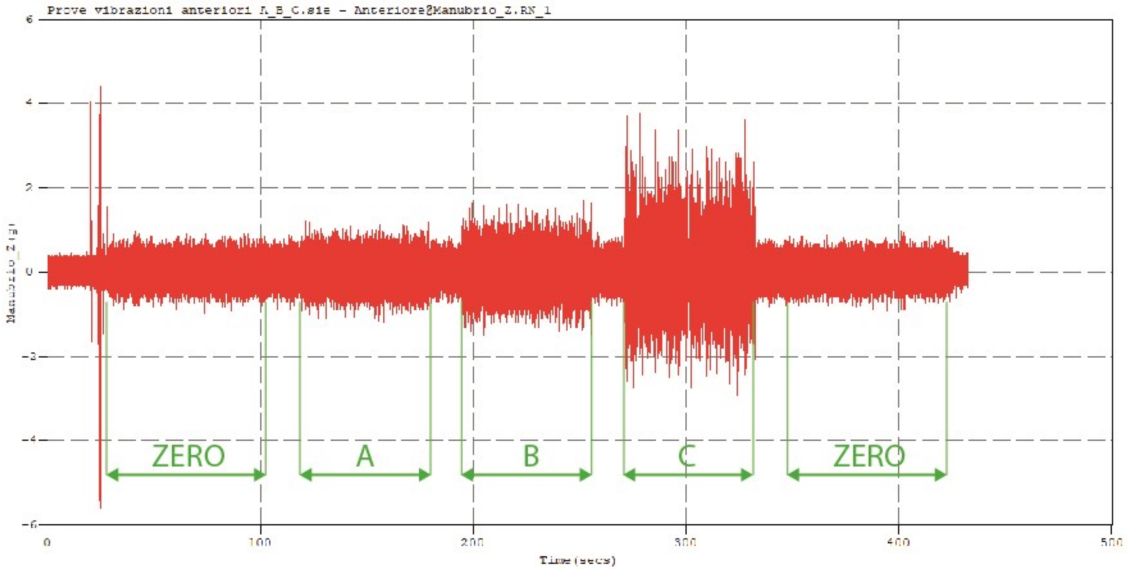

Figure 4.27: Raw data acquisition for road ZERO,A,B,C

Road ZERO signal: the analysis of the raw data from step 2) of procedure for stem and hub, allow to carry out some preliminary checks and considerations. Considering raw data related to the initial acquisition without load coming from actuators is possible to see a periodic behaviour.

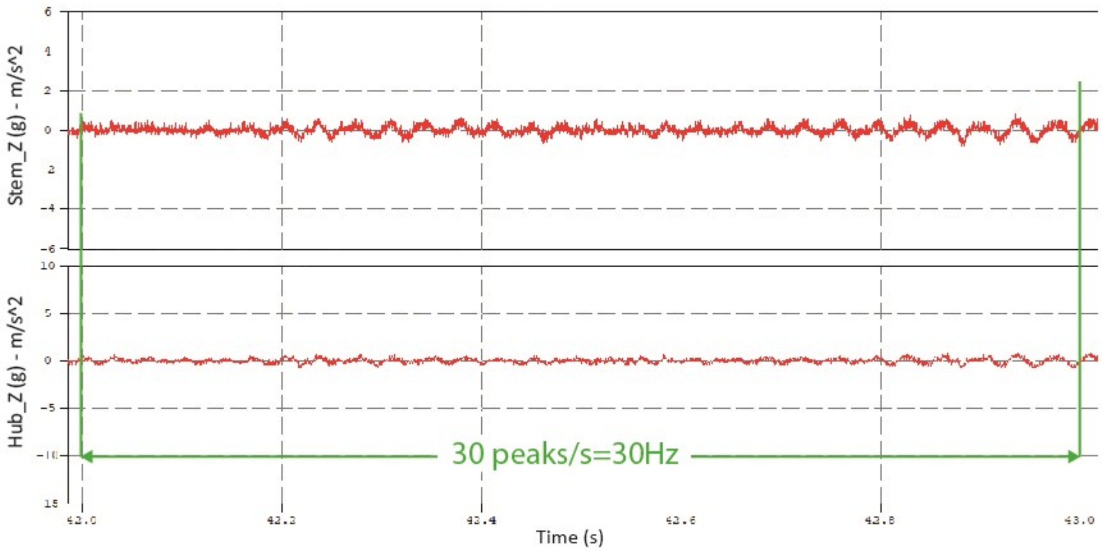

Figure 4.28: One second acquisition (raw data)

$$v = 24\ \frac{Km}{h}\ = 6.66\frac{m}{s}$$

$$r = cilynder\ radius = 0.0375m$$

$$v = \omega \cdot r\ \rightarrow \omega = \frac{v}{r} = \frac{6.66\frac{m}{s}}{0.0375\ m} = 190,3\ \frac{rad}{s}$$

$$\omega = 2 \cdot \pi \cdot n\ \rightarrow n = \frac{\omega}{2 \cdot \pi} = \frac{190.3\frac{rad}{s}}{2 \cdot \pi} = 30,30\ \frac{giri}{s} = 30,30\ Hz$$

The sinusoidal signa is compatible with the eccentricity of the cylinder due to the coupling of the bearings. The rollers' signal generated by their rotation without input vibrations will be quantified in terms of RMS acceleration and PSD. This for is done for the *24 km/h* pedaling speed.

Road type ZERO evaluation: Values of the acceleration (Rms) acquired during the second step of the procedure (60 seconds constant speeD). In this case the street type is identified as ZERO that means without cylinder solicitantion. In table below are reported for Rollers, Wheel hub and Stem the PSD peak and the relative frequency.

Posture 1

Road type ZERO			
	a RMS (g) m/s^2		
Stem type	*Rollers*	*Wheel hub*	*Stem*
SOFT	0,1424	0,4993	0,3012
MEDIUM	0,1534	0,7276	0,4269
HARD	0,1670	0,4183	0,2775
RIGID	0,1830	0,2537	0,2192

Table 4.29: Road type zero RMS acceleration for Rollers, Hub, Stem – Posture1

Road type ZERO						
Stem type	*Rollers*		*Wheel hub*		*Stem*	
	PSD peak(g^2/Hz)	*f (Hz)*	*PSD peak (g^2/Hz)*	*f (Hz)*	*PSD peak (g^2/Hz)*	*f (Hz)*
SOFT	3,90E-04	26	2,02E-01	26	6,14E-02	26
MEDIUM	1,32E-03	27,5	4,38E-01	27	1,35E-01	27,5
HARD	5,20E-04	28	1,60E-01	28	4,43E-02	28
RIGID	2,19E-04	27,5	2,64E-02	27,5	1,54E-02	28

Table 4.230: Psd peak for Rollers, Hub, Stem for road type ZERO (frequency range 0-50 Hz) – Posture1

Posture 2

Road type ZERO			
	a RMS (g) m/s^2		
Stem type	*Rollers*	*Wheel hub*	*Stem*
SOFT	0,1729	0,2794	0,2307
MEDIUM	0,1586	0,3716	0,2226
HARD	0,1844	0,3539	0,2149
RIGID	0,1561	0,2952	0,2122

Table 4.30: Road type zero RMS acceleration for Rollers, Hub, Stem – Posture2

Road type ZERO						
Stem type	*Rollers*		*Wheel hub*		*Stem*	
	PSD peak(g^2/Hz)	*f (Hz)*	*PSD peak (g^2/Hz)*	*f (Hz)*	*PSD peak (g^2/Hz)*	*f (Hz)*
SOFT	2,48E-04	28	5,67E-02	27	1,39E-02	27
MEDIUM	3,62E-04	27,5	8,15E-02	27,5	2,19E-02	27,5
HARD	2,32E-04	28	5,45E-02	28	4,43E-02	28
RIGID	1,44E-04	28	3,87E-02	28	1,42E-02	28

Table 4.31: Psd peak for Rollers, Hub, Stem for road type ZERO (frequency range 0-50 Hz) – Posture2

PSD diagram relative to road ZERO for rollers:

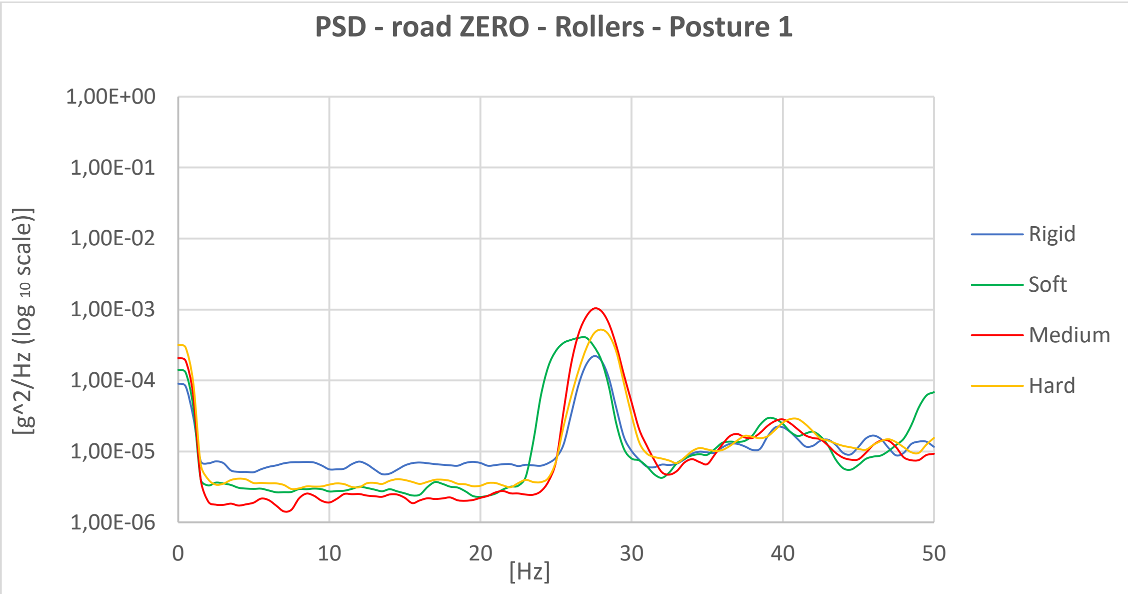

Figure 4.32: Psd peak for road type zero (frequency range 0-50 Hz) – Posture1

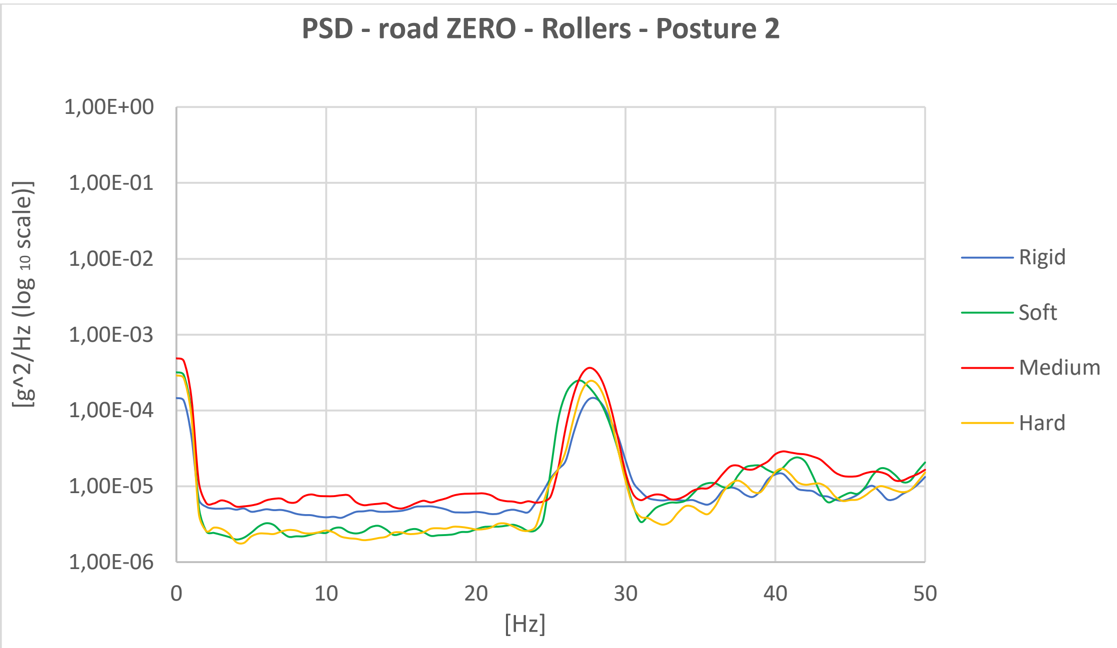

Figure 4.33: Psd peak for road type zero (frequency range 0-50 Hz) – Posture2

Transfer function Relative for road ZERO: For both postures it is possible to note that the peak of the transfer function is obtained for the configuration of the stem set in the medium configuration (ideal in relation of the body weight of the tester). The minimum of the peak is obtained for the rigid stem. This fact is contrary to what we would expect.

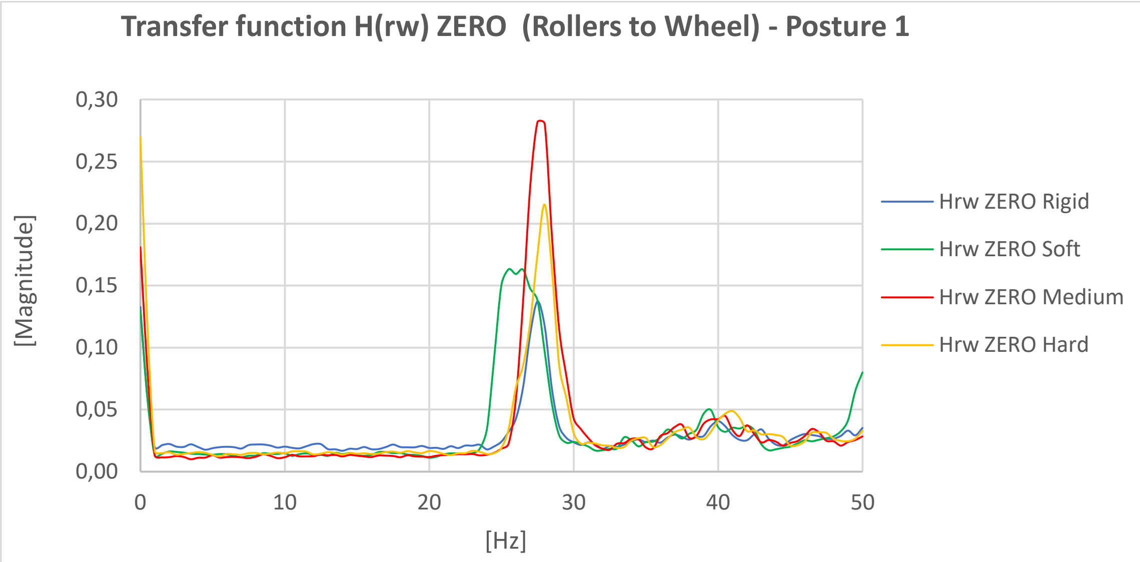

Figure 4.34: Transfer function for road type zero (frequency range 0-50 Hz) – Posture1

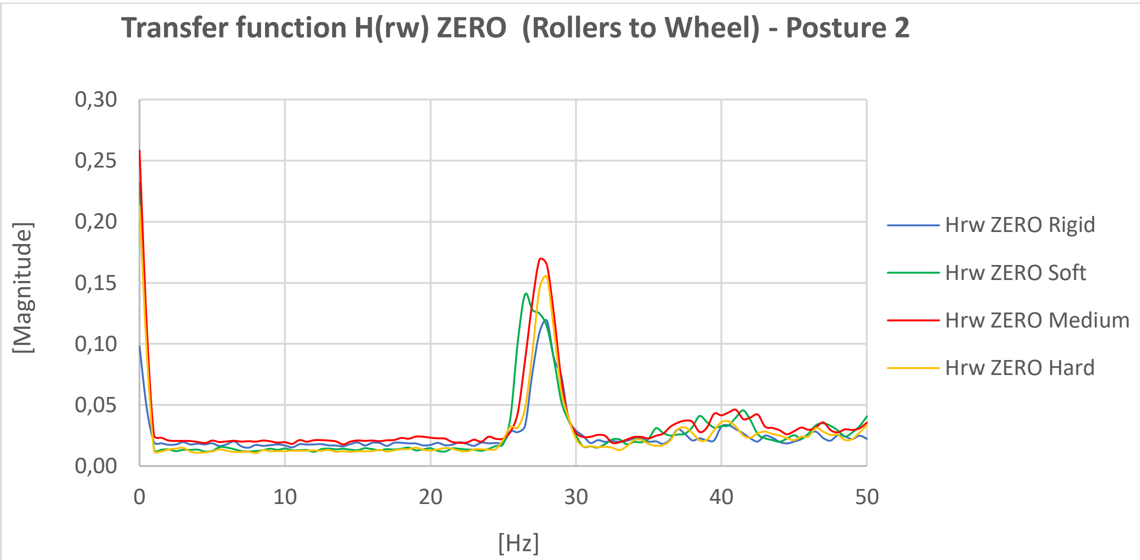

Figure 4.35: Figure 4.34: Transfer function for road type zero (frequency range 0-50 Hz) – Posture2

From the analysis of the transfer function it is evident that for both Posture1 and Posture2 the peak value is recorded around 27 Hz which is substantially equal to the pedaling frequency. The configuration of the rigid stem is the one that has the peak of greatest amplitude, while the one relating to the medium stem records the greatest value.

Pedaling frequency: By analyzing a longer period of time and equal to 30 seconds it is possible to verify the pedaling frequency mixed with arms frequency solicitation.

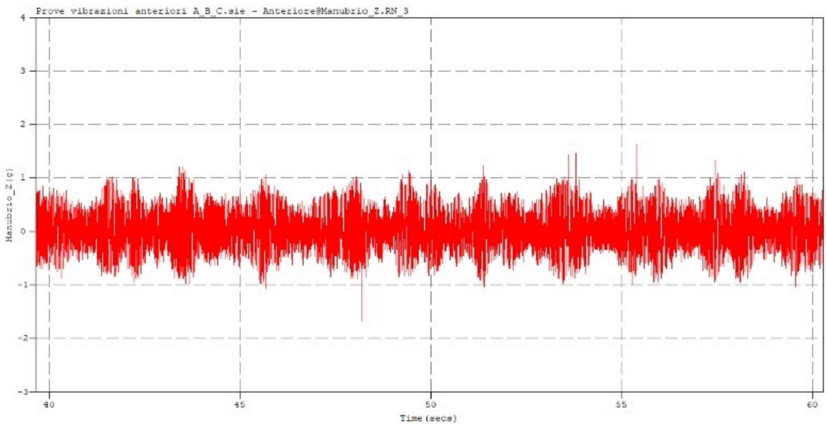

Figure 4.35: Thirty seconds acquisition for pedaling frequency evaluation

During the test, always carried out with the same transmission ratio, the speed was maintained with a frequency of 92 pedal strokes per minute equal to 1,538 Hz. This is equal to 30,8 peaks.

Acceleration comparison (bench and condition stability during the test): In line with the purpose of the analysis, some graphs relating to the acquisitions made to carry out some preliminary checks are presented below. This is to confirm the stability of the bench and of the main parapeters during test sessions

Force and displacement control: The first analysis to be carried out is to check the value of the main quantities such as acceleration, displacement and force.

POSTURE1							
		A		B		C	
		Value	Dev.Std.	Value	Dev.Std.	Value	Dev.Std.
SOFT	a RMS Rolls (g)	0,230	±0,230	0,370	±0,370	0,730	±0,730
	Force F2 (N)	134,470	±132,118	262,364	±261,111	574,809	±572,285
	S2 (mm)	0,844	±0,844	1,732	±1,729	3,788	±3,788
MEDIUM	a RMS Rolls (g	0,240	±0,240	0,372	±0,371	0,734	±0,856
	Force F2 (N)	129,913	±129,235	262,153	±261,158	574,427	±570,964
	S2 (mm)	0,818	±0,814	1,720	±1,720	3,741	±3,741
HARD	a RMS Rolls (g	0,238	±0,240	0,376	±0,375	0,734	±0,734
	Force F2 (N)	130,190	±130,120	262,234	±261,478	572,977	±571,378
	S2 (mm)	0,826	±0,826	1,702	±1,702	3,634	±3,364
RIGID	a RMS Rolls (g	0,246	±0,245	0,381	±0,380	0,728	±0,728
	Force F2 (N)	130,090	±130,001	261,830	±261,156	573,343	±574,247
	S2 (mm)	0,826	±0,826	1,715	±1,715	3,780	±3,780

Table 4.36: Main parameters evaluation and control table – Posture1

POSTURE1						
	A		B		C	
	Value	Dev.Std	Value	Dev.Std	Value	Dev.Std
Force F2 (N)	130,415	±0,712	262,161	±0,236	573,890	±0,867
Displacement S2 (mm)	0,828	±0,011	1,717	±0,012	3,736	±0,071
Acceleration a RMS (g)	0,238	±0,006	0,375	±0,005	0,732	±0,003

Table 4.37: Main parameters evaluation and control table – Posture1

POSTURE2							
		A		B		C	
		Value	Dev.Std.	Value	Dev.Std.	Value	Dev.Std.
SOFT	a RMS Rolls (g)	0,220	±0,220	0,364	±0,365	0,728	±0,728
	Force F2 (N)	129,650	±190,154	263,410	±262,985	576,79	±575,801
	S2 (mm)	0,837	±0,836	1,710	±1,710	3,779	±3,780
MEDIUM	a RMS Rolls (g	0,264	±0,264	0,381	±0,380	0,737	±0,737
	Force F2 (N)	129,530	±129,478	262,460	±261,917	574,050	±573,987
	S2 (mm)	0,838	±0,838	1,695	±1,684	3,737	±3,736
HARD	a RMS Rolls (g	0,224	±0,224	0,361	±0,360	0,719	±0,719
	Force F2 (N)	129,440	±129,852	262,780	±262,746	571,560	±570,989
	S2 (mm)	0,831	±0,831	1,700	±1,700	3,761	±3,761
RIGID	a RMS Rolls (g	0,243	±0,243	0,373	±0,373	0,732	±0,731
	Force F2 (N)	129,60	±130,691	261,880	±261,963	578,540	±578,235
	S2 (mm)	0,835	±0,835	1,702	±1,702	3,795	±3,795

Table 4.38: Main parameters evaluation and control table – Posture2

POSTURE2						
	A		B		C	
	Value	Dev.Std	Value	Dev.Std	Value	Dev.Std
Force F2 (N)	129,659	±0,245	262,635	±0,638	575,235	±3,069
Displacement S2 (mm)	0,835	±0,003	1,702	±0,006	3,768	±0,025
Acceleration a RMS (g)	0,238	±0,020	0,370	±0,009	0,729	±0,007

Table 4.39: Main parameters evaluation and control table – Posture2

There is a substantial constancy of the values, a symptom of the fact that the test conditions and the execution of the test itself have not changed and have not intervened in their modification.
Force are calculated without considering weight of bike and tester because load cell have been reset during the start position.
Force, displacement and acceleration for anterior rollers (2) relative to road A,B,Care represented vbelow

Graphical representation of main quantities to control the stability of the test:

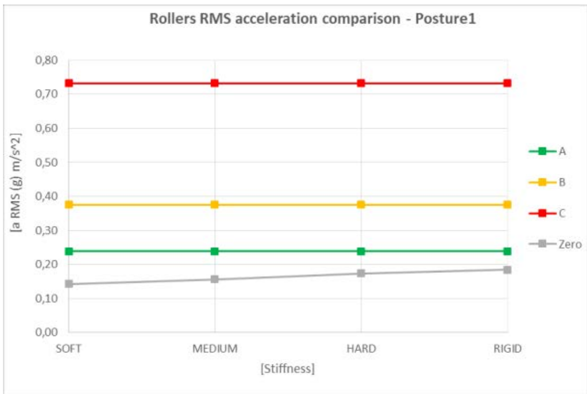

Figure 4.40: Roller acceleration comparison – Posture1

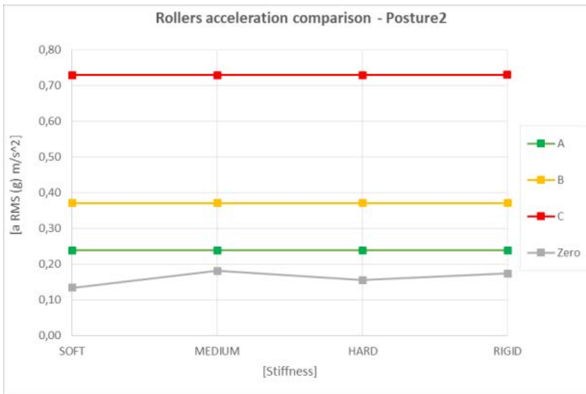

Figure 4.41: Roller acceleration comparison – Posture2

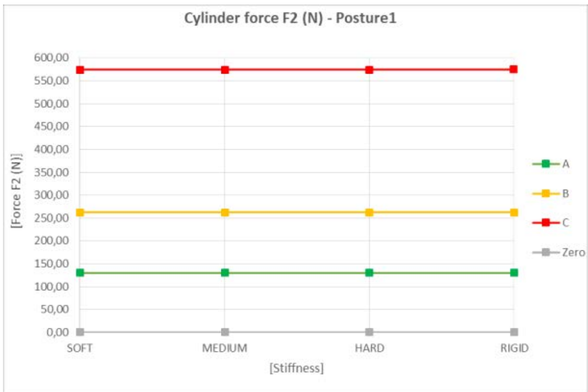

Figure 4.42: Cylinder force comparison – Posture1

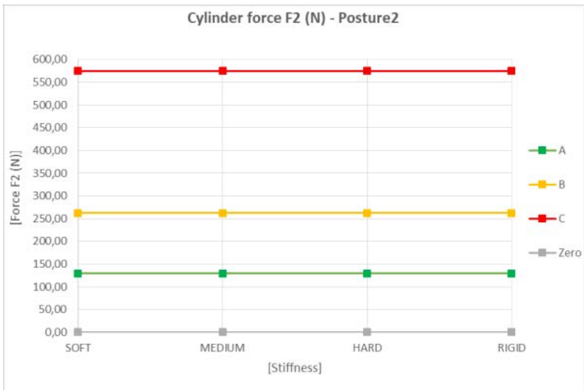

Figure 4.43: Cylinder force comparison – Posture2

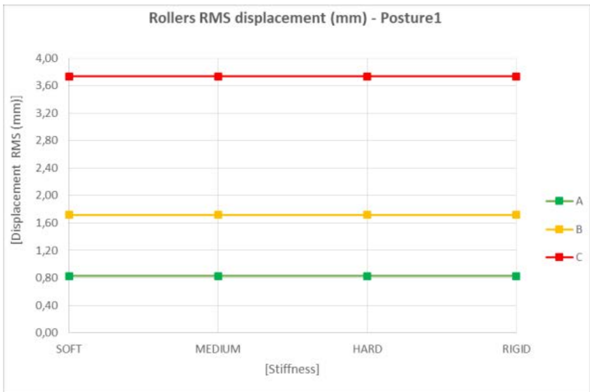

Figure 4.44: Displacement comparison – Posture1

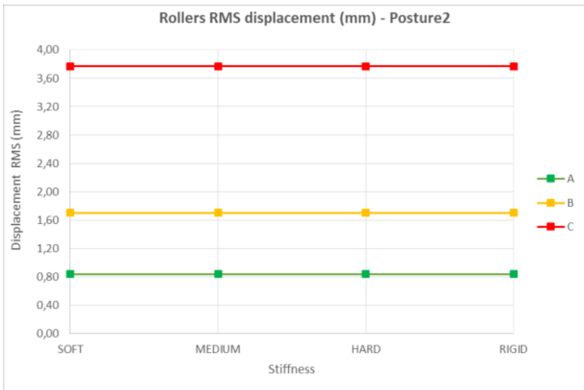

Figure 4.45: Displacement comparison – Posture1

Wheel hub and stem acceleration comparison for Posture1: Once the stability of the acceleration for the roller system (input) has been verified, the significant quantities to be represented for an initial analysis of the system's behavior are the RMS value of the acceleration of the hub and of the stem:

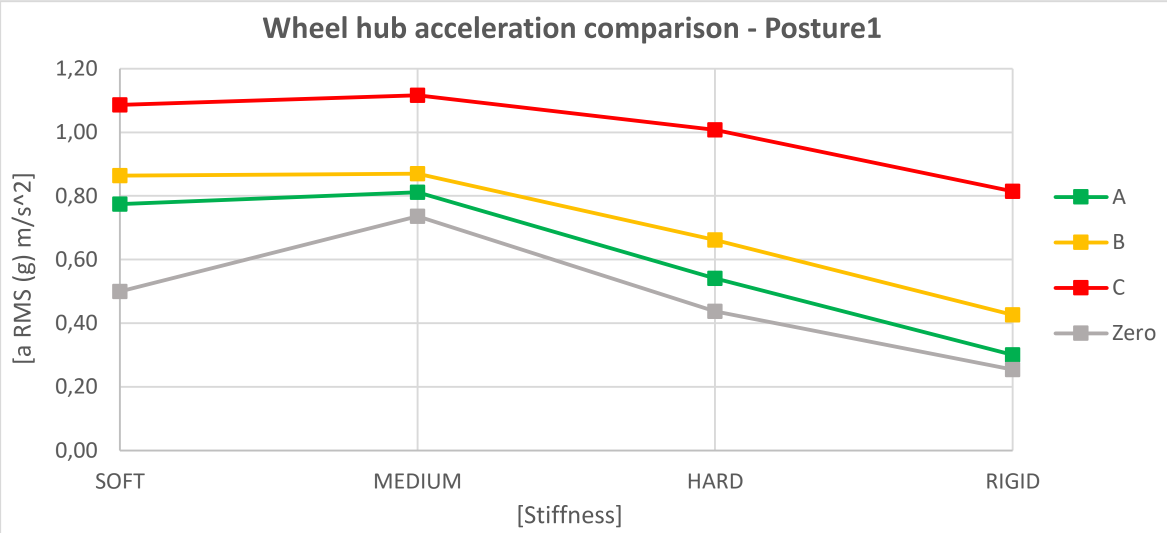

Figure 4.46: Wheel hub acceleration comparison for Posture1

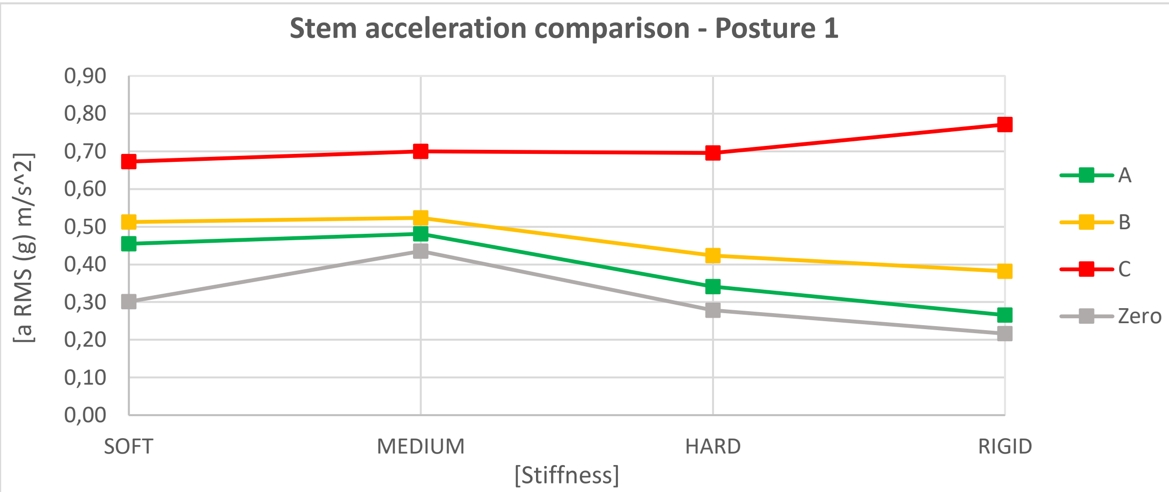

Figure 4.47: Stem acceleration comparison for Posture1

Analyzing the figure (above) it can be observed that for all types of roads (A, B, C) and for all configurations of the rod (SOFT, MEDIUM, HARD) the accelerations at the hub are greater than in the case of a rigid rod.

This means that for a given value of the acceleration imposed by the rollers, the introduction of the dampened rod generates a greater amplification of the acceleration at the hub than would happen with a rigid rod.

The graph (Below) highlights the global behavior by representing the acceleration of the stem. It is evident that for the type C road and for the SOFT, MEDIUM, HARD configurations, the damped rod reduces the average value of vibrations compared to the rigid case.

Rollers ,Wheel hub and stem acceleration comparison for Posture1:

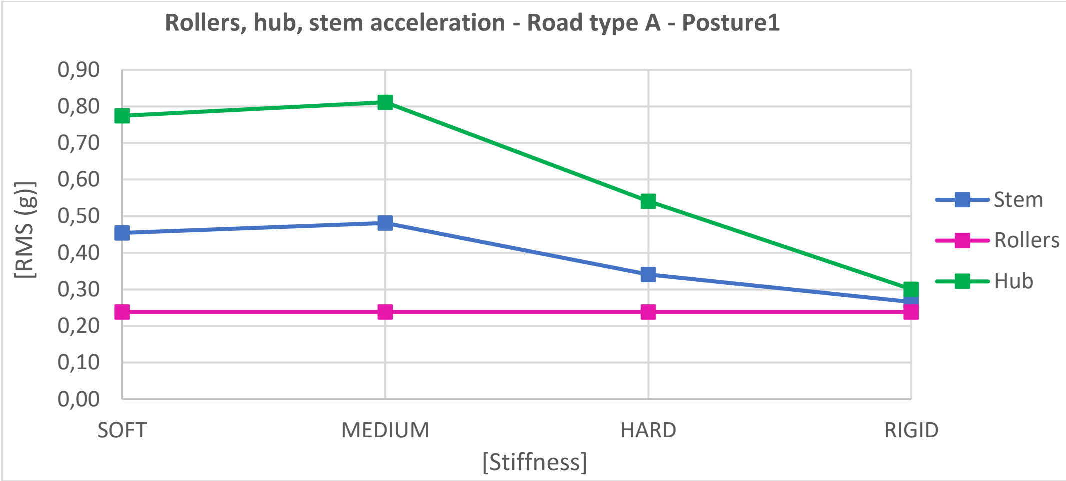

Figure 4.48: Rollers, hub and stem accelerations – Posture1 - Road A

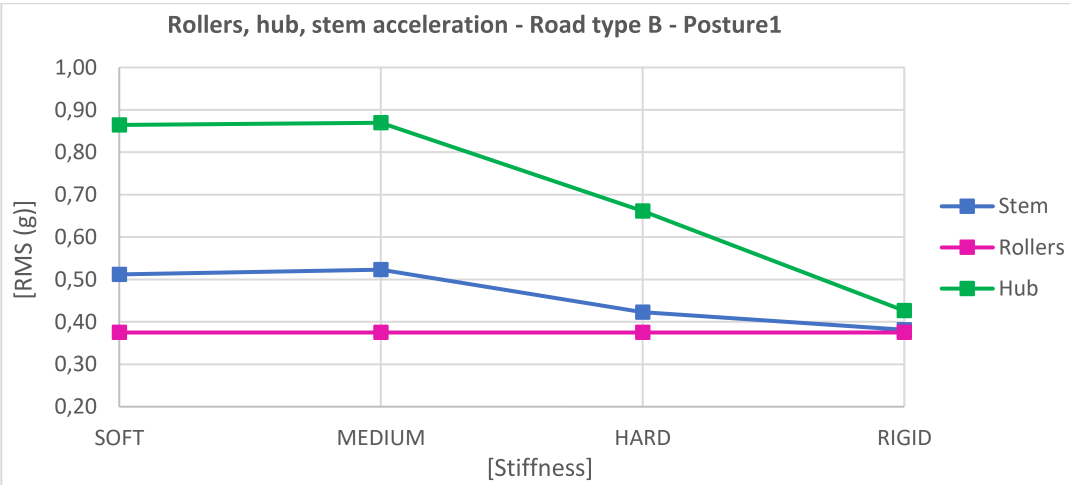

Figure 4.49: Rollers, hub and stem accelerations – Posture1 - Road B

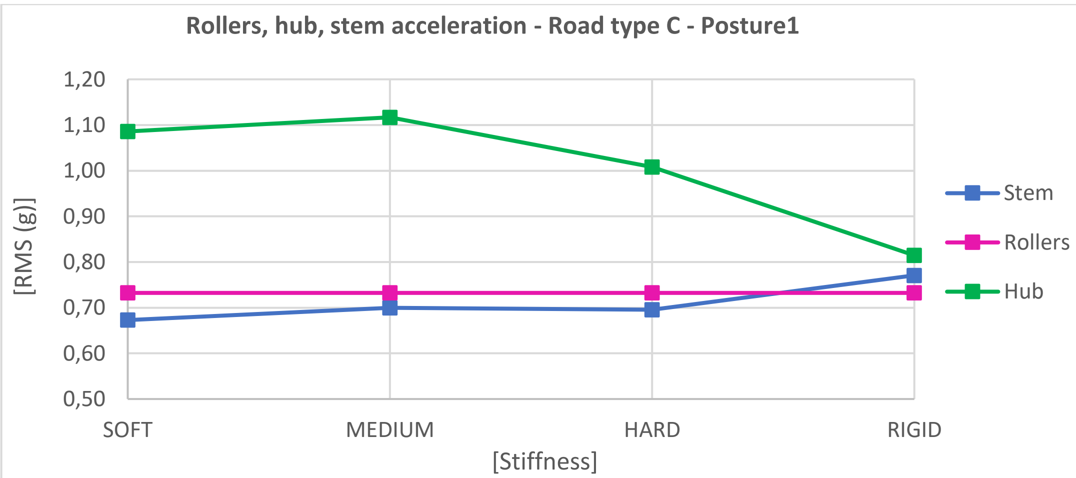

Figure 4.50: Rollers, hub and stem accelerations – Posture1 - Road C

The behavior of the bike equipped with different stems is evident by analyzing the graphs placed below. For a given value of the roller acceleration (at RMS) and characteristic of each type of road (A, B, C), there is an increase in the average acceleration of the hub which is subsequently damped by the rod.

For road A and B, the decrease in acceleration between hub and stem is significant if compared with the rigid case. Its reduction by the amortized stem is not sufficient to obtain average acceleration values (at RMS) of the stem lower than in the rigid case.

The reduction of accelerations compared to the rigid case is only evident for the type C road for the SOFT, MEDIUM, HARD configurations.

Wheel hub and stem acceleration comparison for Posture2:

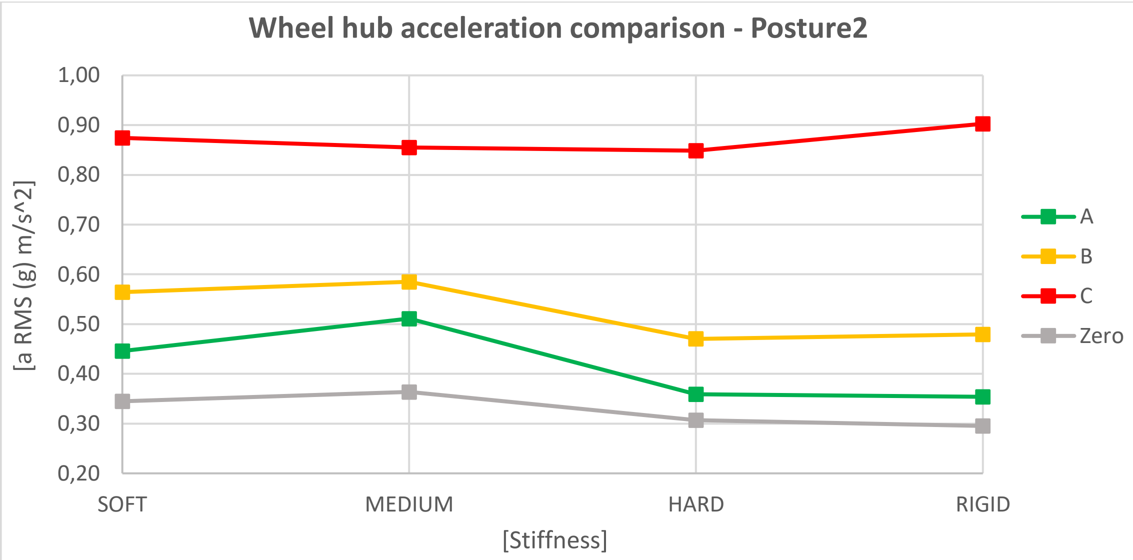

Figure 4.51: Wheel hub acceleration comparison for Posture2

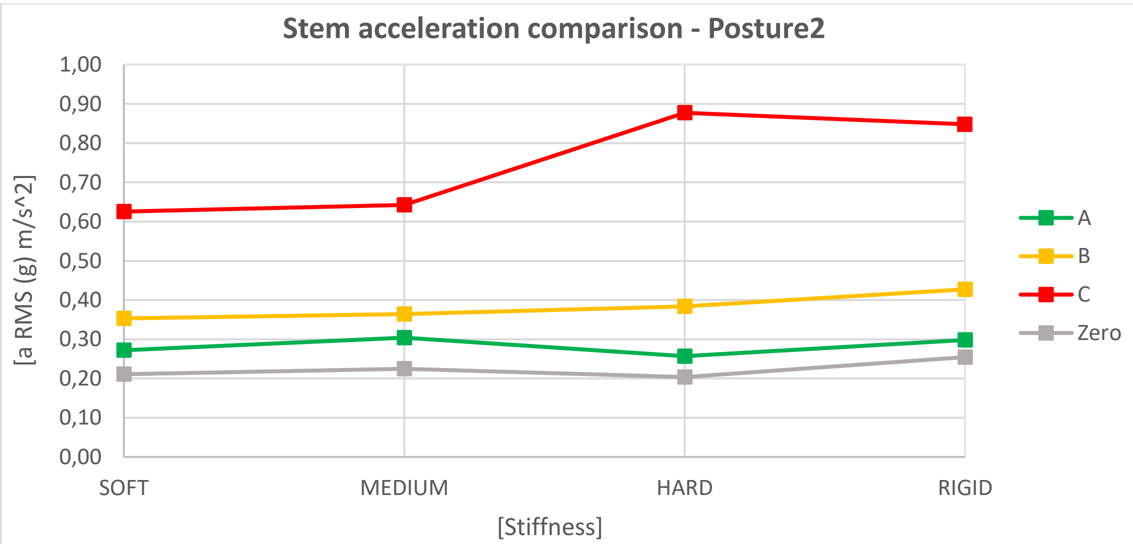

Figure 4.52: Stem acceleration comparison for Posture2

Rollers ,Wheel hub and stem acceleration comparison for Posture2:

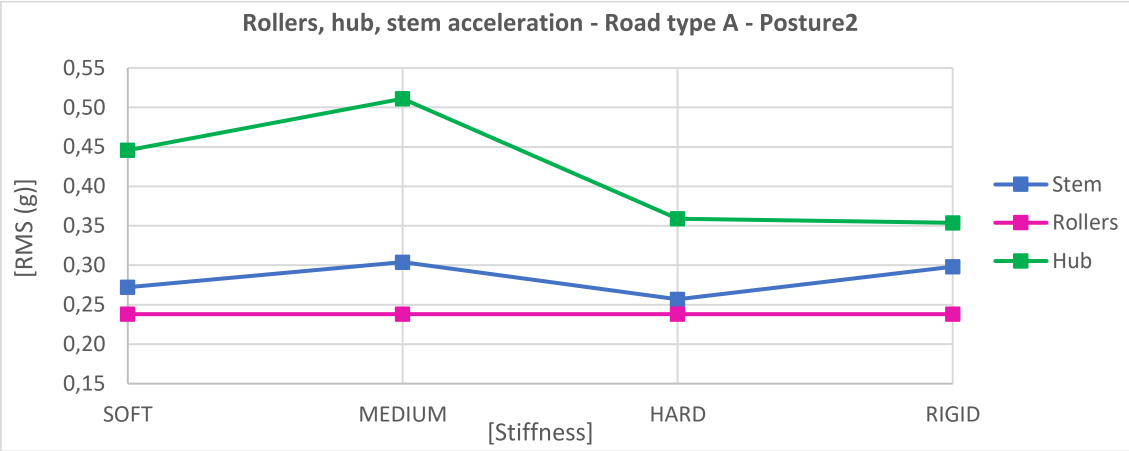

Figure 4.53: Rollers, hub and stem accelerations – Posture2 - Road A

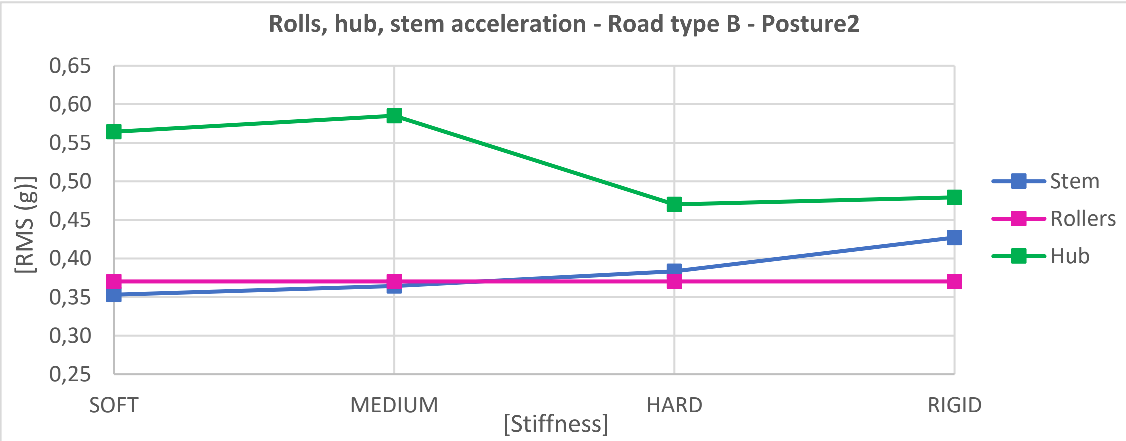

Figure 4.54: Rollers, hub and stem accelerations – Posture2 - Road B

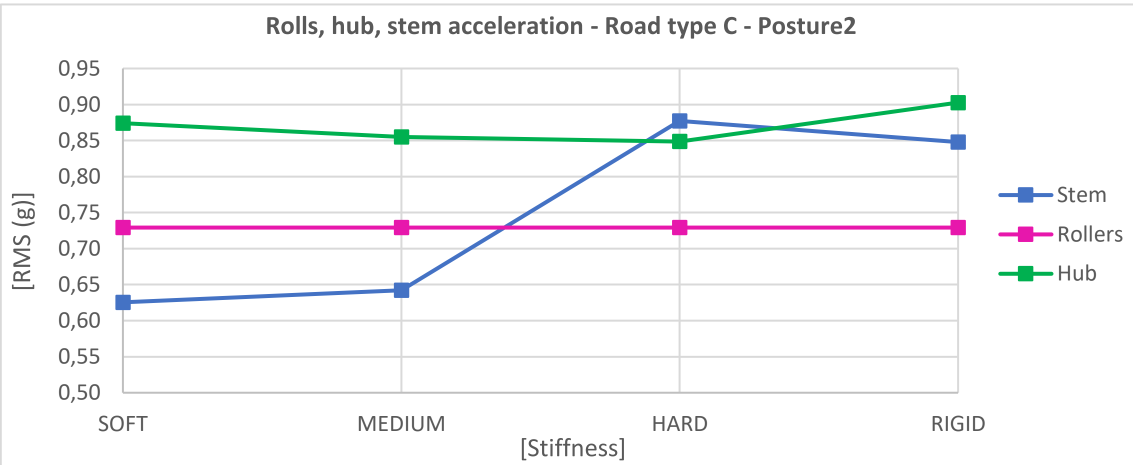

Figure 4.55: Rollers, hub and stem accelerations – Posture2 - Road C

For Posture2 the behavior is different from the previous case. For the roads type A and B the accelerations of the hub for the SOFT, MEDIUM and HARD configuration are greater than in the rigid case, while for the road type C they are lower. Overall, analyzing the behavior of the stem it is not possible to say the same. For the type A road, the SOFT and HARD configuration reduces the average acceleration value compared to the RIGID case, while for the MEDIUM setting it is equivalent to the RIGID case.
For the road type B, all accelerations for the suspension stem are lower than for the rigid one.
For the type C road, only the SOFT and MEDIUM configuration is able to reduce the average value of the accelerations compared to the RIGID case, while the HARD configuration presents a greater acceleration than the RIGID case.

Transmission index for Posture 1: TI_{rw}(roll-wheel) – TI_{ws}(wheel-stem) – TI_{rs}(rolls-stem)

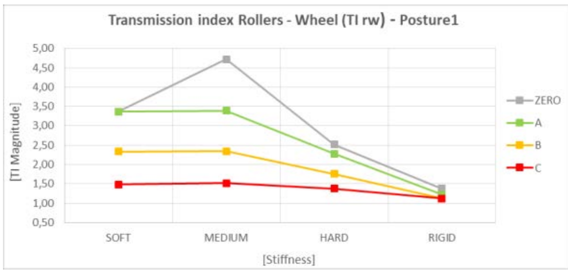

Figure 4.56: TI (rw) variation – Posture1

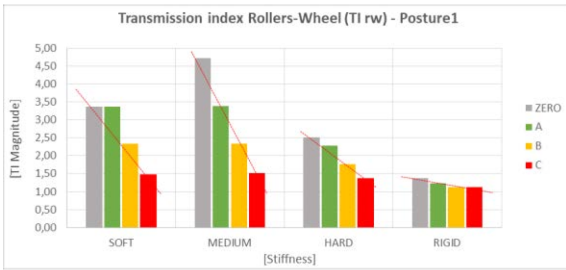

Figure 4.57: TI (rw) variation – Posture1

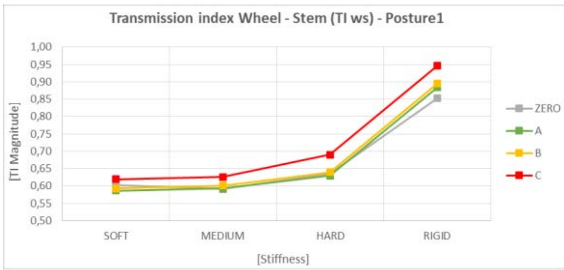

Figure 4.58: TI (ws) variation – Posture1

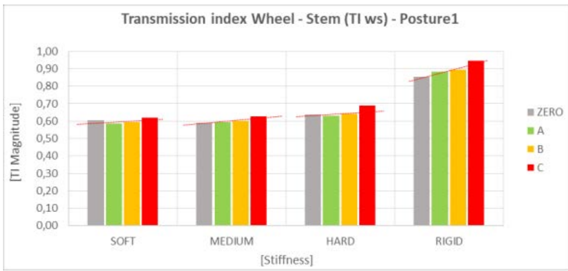

Figure 4.59: TI (ws) variation – Posture1

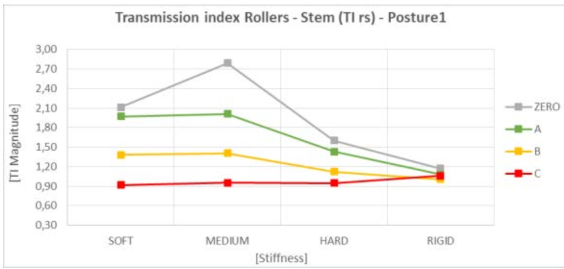

Figure 4.60: TIindex (rs) variation – Posture1

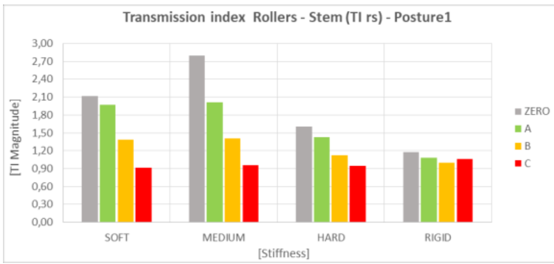

Figure 4.61: TI (rs) variation – Posture1

The definition of comfort index described above is remembered:

$TI(rw) = \frac{a_{fw}^{RMS}}{a_{fr}^{RMS}}$ = Transmission index between roller and front wheel

$TI(ws) = \frac{a_{st}^{RMS}}{a_{fw}^{RMS}}$ = Transmission index between front wheel and stem

$TI(rs) = \frac{a_{st}^{RMS}}{a_{fr}^{RMS}}$ = Transmission index between roller and stem

Fig. 5.56 clearly highlights how the effect of the suspension stem is to amplify the average acceleration value of the hub with respect to the roller more than what happens for the rigid stem. Figure 4.58 shows how the relationship between the accelerations between rod and hub is amplified as the stiffness of the rod increases. The global effect is visible in figure 4.60 which shows how for the roads

(ZERO, A, B) the global effect is to amplify the accelerations of the cushioned stem compared to the rigid one. This does not happen for the C road.

The nature of the study is comparative, therefore the percentage variations of the TI with respect to the rigid stem are reported. This will be done for both postures 1 and 2:

		Variation % of TI (rs) respect to RIGID STEM			
	Stem type	ZERO	A	B	C
M1_PA5060	SOFT	80,17	82,63	38,33	-12,95
M1_PA7060	MEDIUM	137,95	86,15	40,79	-9,96
M1_PA9050	HARD	36,60	132,47	12,47	-10,48

Table 4.62: Table of variation % of transmission index respect to rigid stem – Posture1

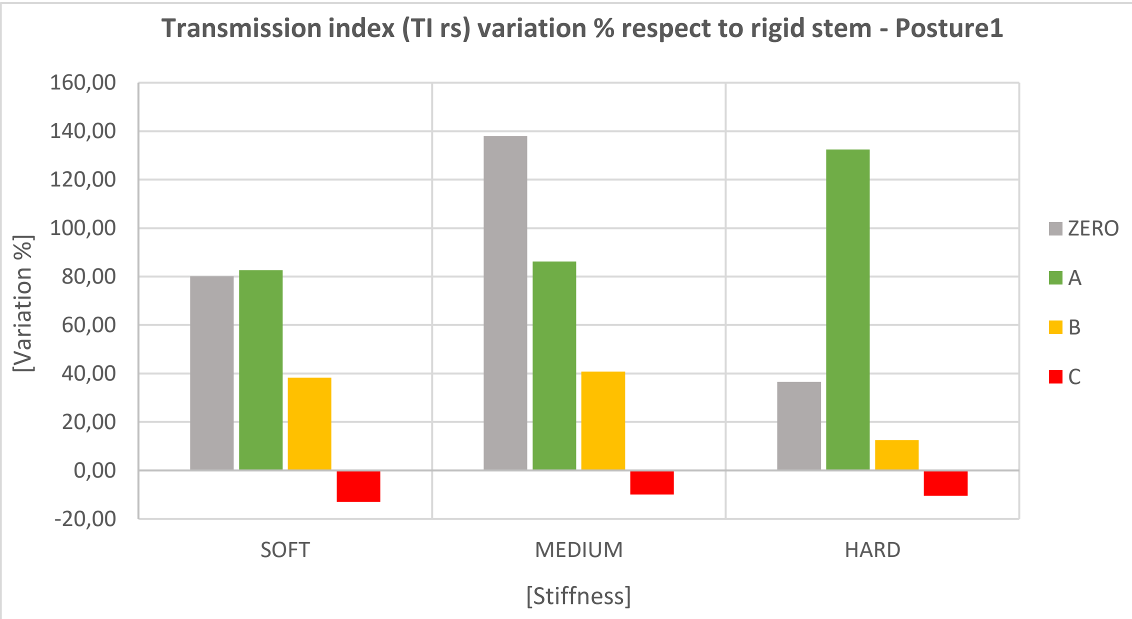

Figure 4.63: Variation % of transmission index respect to rigid stem – Posture1

The analysis of the transmission index between roller and rod and the evaluation of its reduction compared to the rigid case allows to provide a global evaluation.
It is observed that for Posture1 only for road C a significant reduction in the value of the average acceleration is obtained compared to the rigid case, while for all the other configurations of the stem (SOFT, MEDIUM, HARD) and for roads A and B the value average acceleration is greater than in the case of a rigid rod.

Transmission index for Posture2: TI_{rw}(roll-wheel) – TI_{ws}(wheel-stem) – TI_{rs}(rolls-stem)

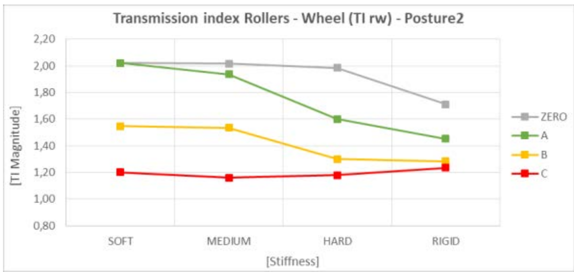

Figure 4.64: TI (rw) variation – Posture2

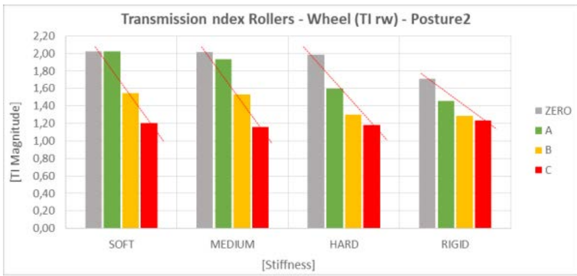

Figure 4.65: TI (rw) variation – Posture2

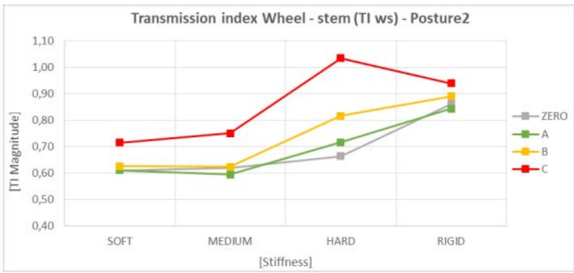

Figure 4.66: TI (ws) variation – Posture2

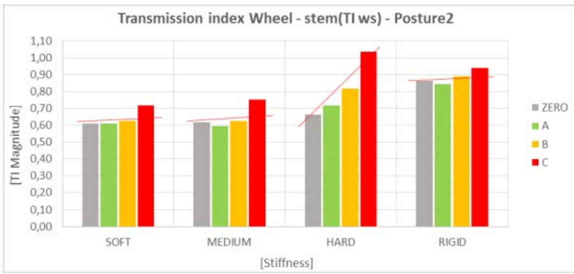

Figure 4.67: TI (ws) variation – Posture2

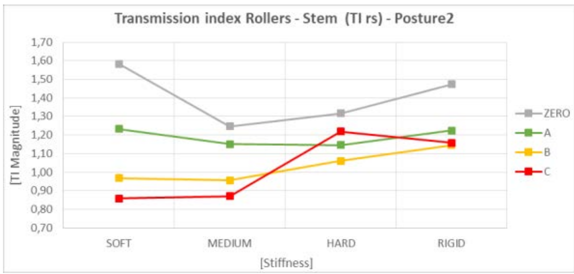

Figure 4.68: TI (rs) variation – Posture2

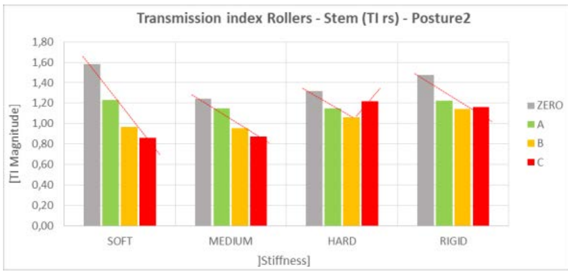

Figure 4.69: TI (rs) variation – Posture2

From figure 4.64 we observe the amplification of the accelerations at the hub with respect to those at the roller. This is valid for roads ZERO, A, B but not for C.

The TI is always greater for the RIGID stem, except for the case of HARD and Road C settings. The global effect is visible in figure 4.68. For road A there is a reduction in accelerations for the MEDIUM and HARD settings while there is an equivalence for the SOFT.

For road B the suspension stem is advantageous for all settings (SOFT, MDEIUM, HARD).

For road C only the SOFT and MEDIUM settings are advantageous but not for the HARD one.

		Variation % of TI (rs) respect to RIGID STEM			
	Stem type	ZERO	A	B	C
M1_PA5060	SOFT	7,34	0,71	-15,37	-25,84
M1_PA7060	MEDIUM	-15,46	-5,97	-16,47	-24,80
M1_PA9050	HARD	-10,63	93,57	-7,30	5,22

Table 4.70: Table of variation % of transmission index respect to rigid stem – Posture2

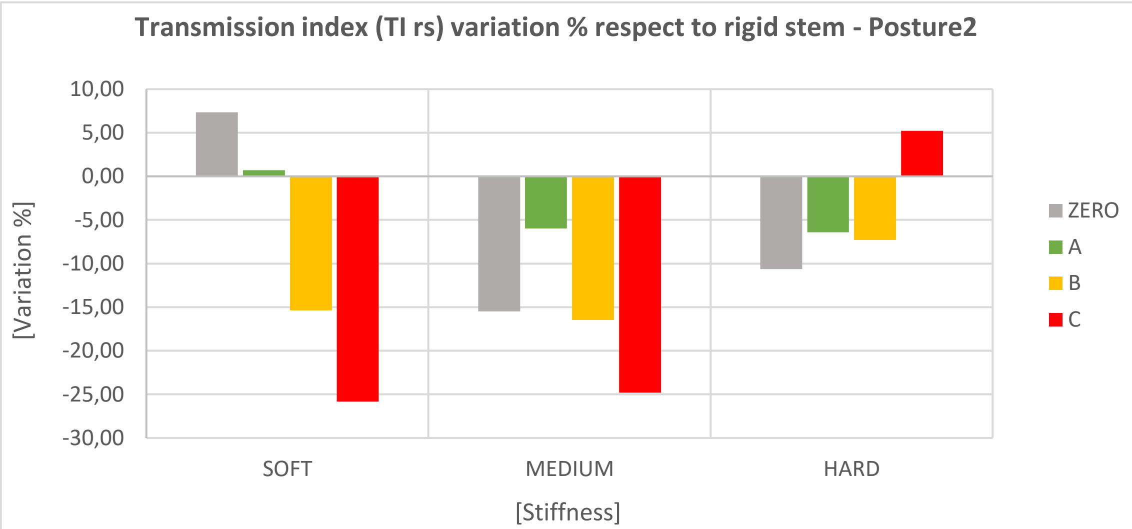

Figure 4.71: Variation % of transmission index respect to rigid stem – Posture1

The comparative figure 4.71 showing the percentage change with respect to the rigid case highlights the overall behavior. For Posture2 the suspension stem is always advantageous except for Road C and Hard setting and road A and ZERO and SOFT setting.

Transfer functions: Hrs (Rolls – Stem) for Posture1:

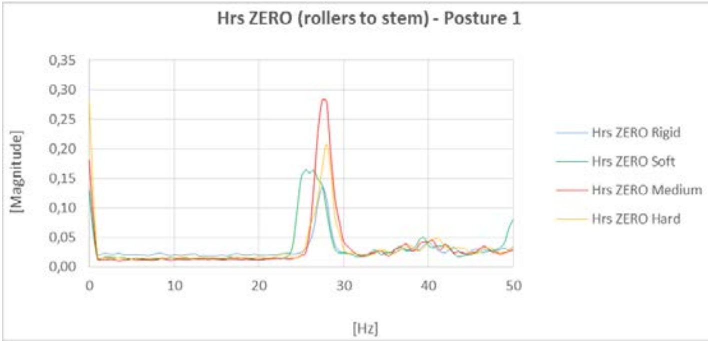

Figure 4.72: Transfer function graphic (roller-stem) for road ZERO - Posture1

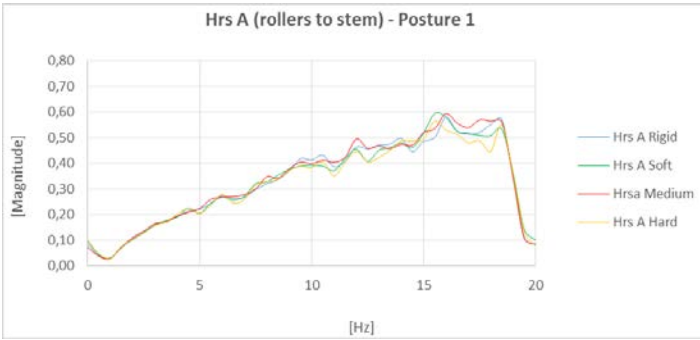

Figure 4.73: Transfer function grapich (roller-stem) for road A - Posture1

Figure 4.74: Transfer function graphic (roller-stem) for road B - Posture1

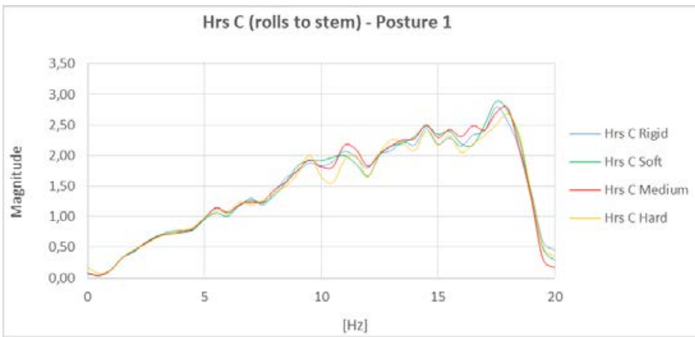

Figure 4.75: Transfer function graphic (roller-stem) for road C - Posture1

Transfer functions: Hrs (Rolls – Stem) for Posture2:

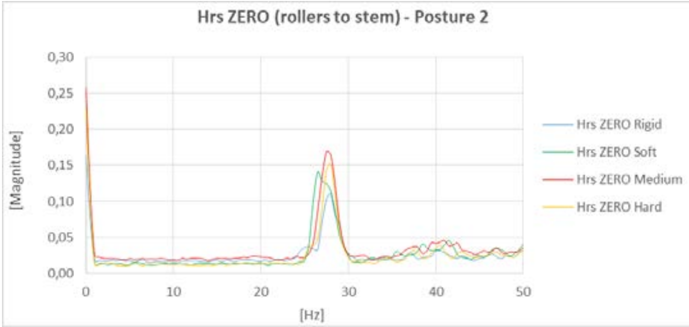

Figure 4.76: Transfer function graphic (roller-stem) for road ZERO – Posture2

Figure 4.77: Transfer function graphic (roller-stem) for road A – Posture2

Figure 4.78: Transfer function graphic (roller-stem) for road B – Posture2

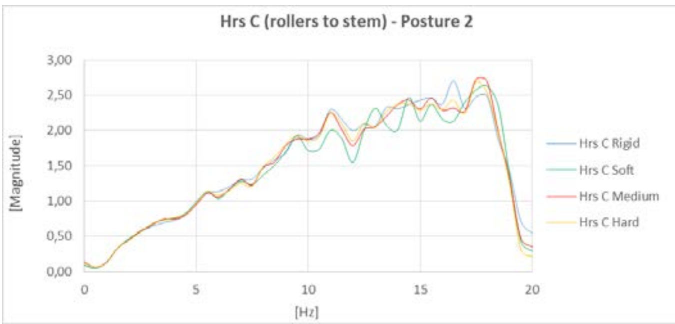

Figure 4.79: Transfer function graphic (roller-stem) for road C – Posture2

Power spectral density: The power spectral density relative to stem has been completely caluclated in appendix. Here are reported relative to Road B and C.

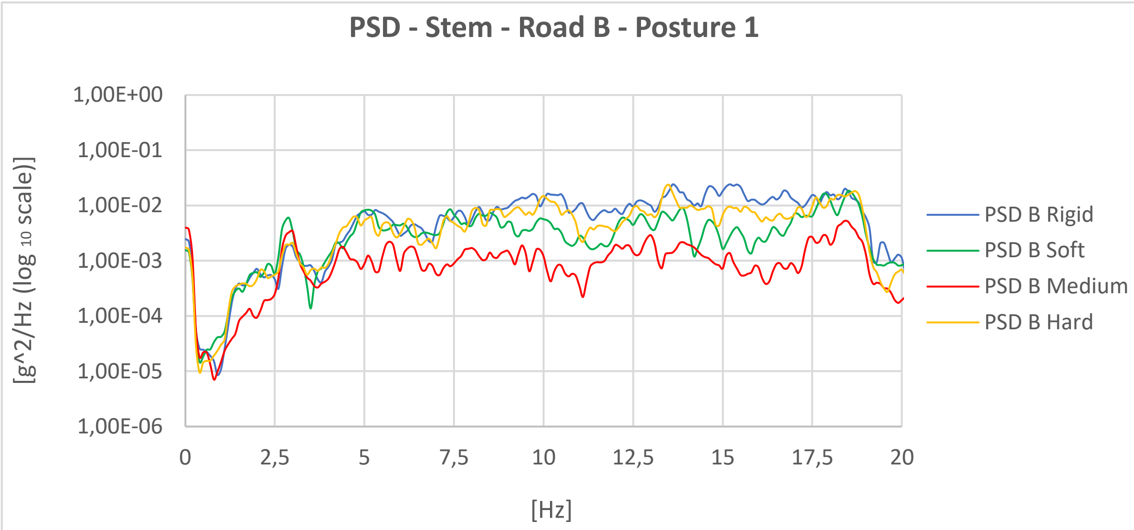

Figure 4.80: Power spectral density graphic for stem –Road B –Posture1

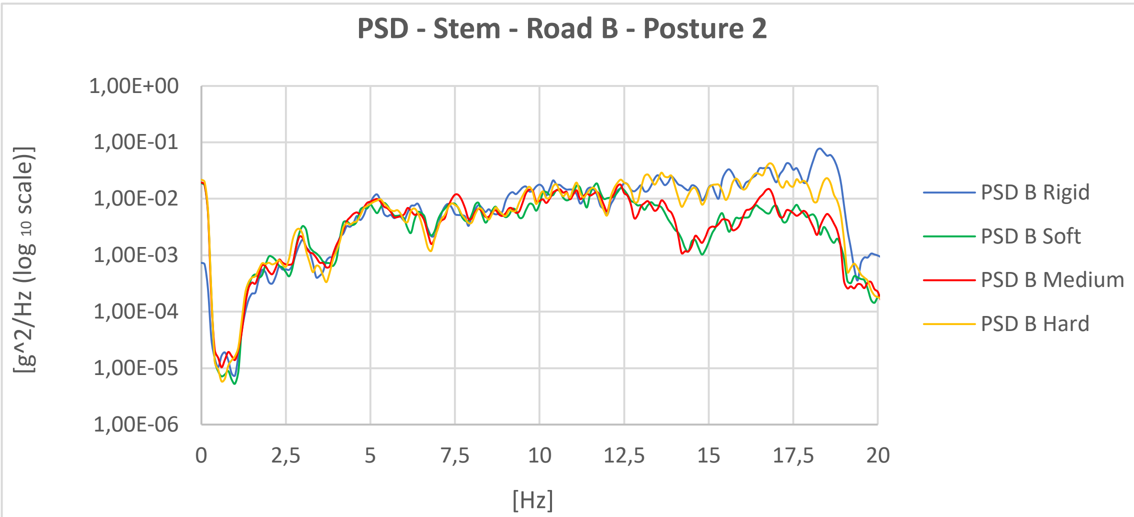

Figure 4.81: Power spectral density graphic for stem – Road B –Posture2

The comparison of the two graphs relating to the PSD highlights that for Posture1 (MEDIUM setting) the curve lies below all in an interval between 5 and 7 Hz. For the SOFT configuration, on the other hand, this occurs between 8 and 18 Hz. The associated spectral power level is in this case lower for both curves than in the RIGID case.
For Posture2 (For the SOFT and MEDIUM setting) there is a significant distance from the curves relating to the HARD and RIGID setting in an interval between 12 and 18 HZ.
This difference in behavior between Posture1 and Posture2 is certainly due to the coupling between the upper limbs and the handlebars. The two postures are characterized by a different load transfer to the handlebar and have different stiffness due to the posture and the bio-mechanical characteristic of the upper limbs.

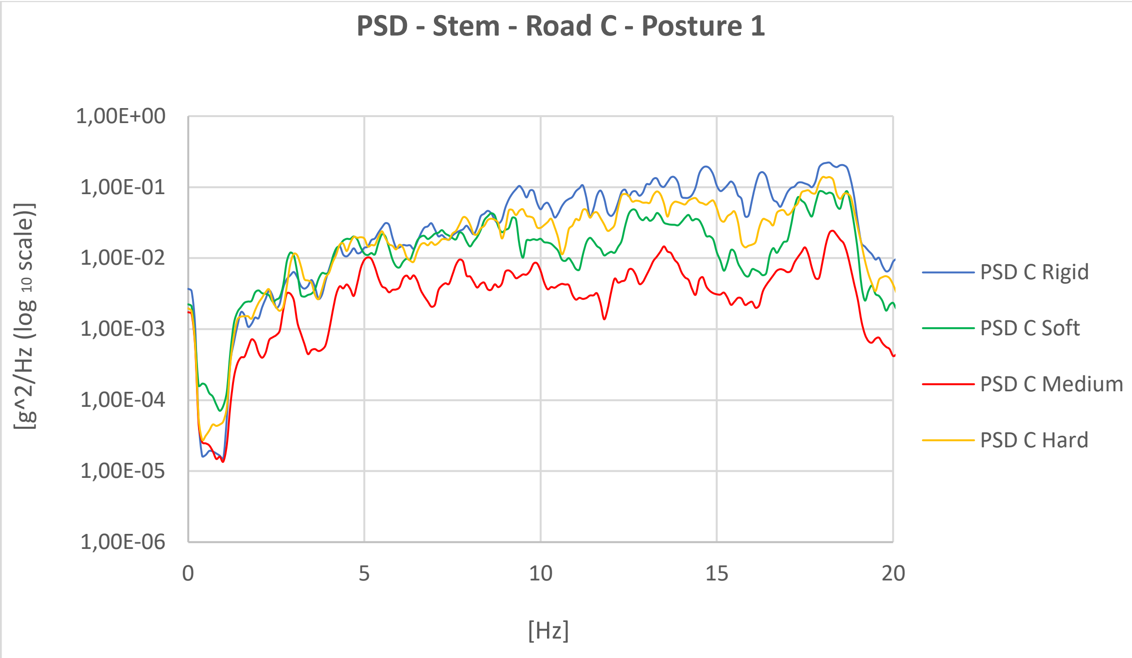

Figure 4.82: Power spectral density graphic for stem –Road C –Posture1

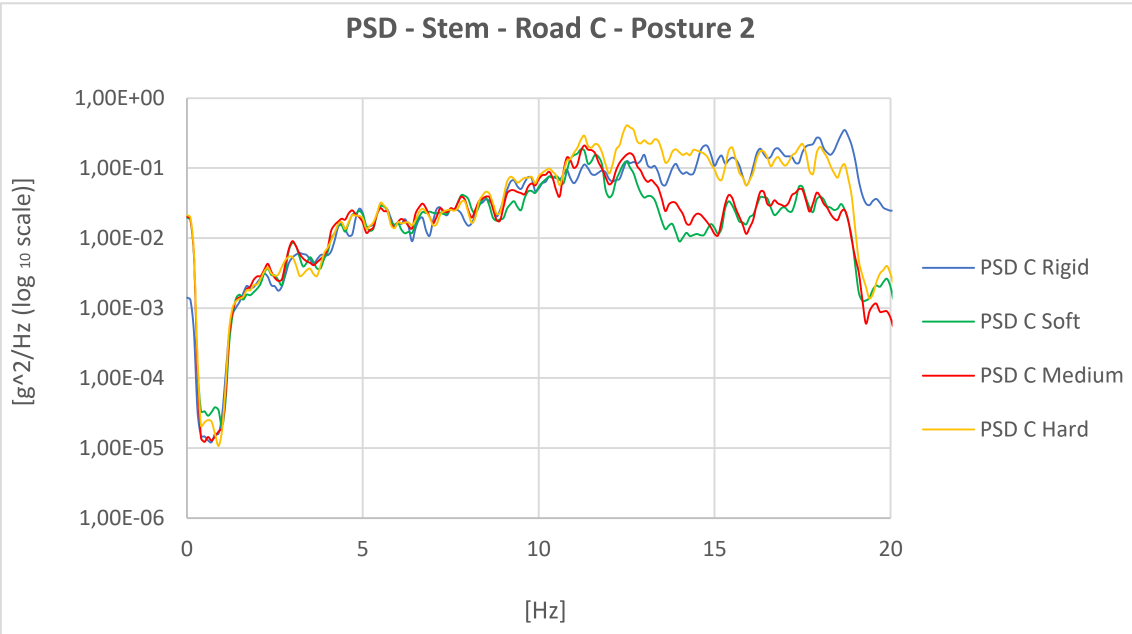

Figure 4.83: Power spectral density graphic for stem –Road C –Posture2

The differences described above between curves and postures are not found also in the transfer functions.

Comfort index: ISO 2631-1, [9] expresses the human body sensibility in seated position along the vertical Z axis. For this reason, a Comfort Index of different stems and posture, has been calculated. For the human body trunk is the reciprocal of the integral of the transfer function Hrs between 4 and 8 Hz but for the for the hand-arm system is between 12 and 16 Hz.

$$CI_{12\text{-}16} = \frac{1}{\int_{12}^{16} \text{Hrs df}} \tag{4.6}$$

Comfort index value for Posture 1

		Comfort index (12-16 Hz) - Posture 1			
		Harshness			
		ZERO	A	B	C
Stiffness	SOFT	42,411	1,201	0,566	0,251
	MEDIUM	45,481	1,139	0,573	0,258
	HARD	38,387	1,144	0,579	0,261
	RIGID	29,192	1,132	0,563	0,267

Table 4.84: Comfort index tabulation for different road Types – Posture1

		Variation % respect rigid stem - Posture 1			
		Harshness			
		ZERO	A	B	C
Stiffness	SOFT	45,282	6,040	0,497	-2,660
	MEDIUM	55,798	0,609	1,78	-3,521
	HARD	31,496	1,051	2,810	-2,10

Table 4.85: Comfort index variation % respect rigid stem tabulation – Posture1

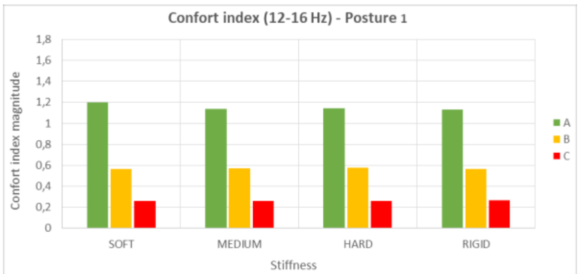

Figure 4.85: Comfort index – Posture1

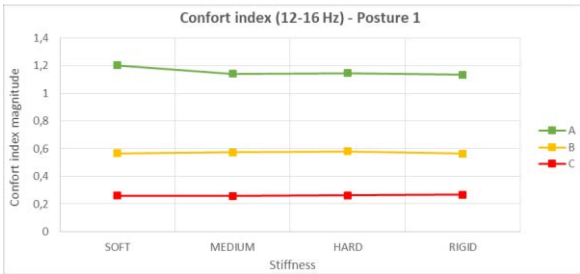

Figure 4.85: Comfort index grapic – Posture1

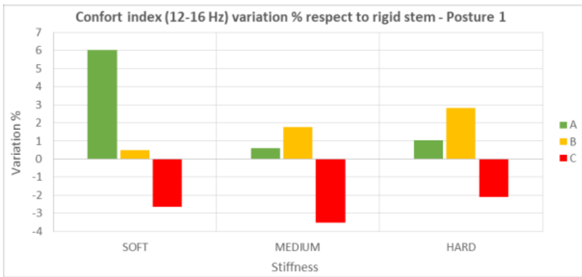

Figure 4.86: CI variation respect to rigid G.B. – Posture1

Figure 4.87: CI variation respect to rigid – Posture1

For road type C the CI compared to the rigid stem is lower while it is higher for road type A and B. There is a maximum value for the SOFT setting on road A and almost linearly increasing for road B passing from SOFT to HARD.

Comfort index value for Posture 2

		Comfort index (12-16 Hz) - Posture 2			
		Harshness			
		ZERO	A	B	C
Stiffness	SOFT	42,139	1,155	0,580	0,263
	MEDIUM	27,914	1,244	0,579	0,263
	HARD	32,468	1,188	0,586	0,261
	RIGID	20,083	1,154	0,567	0,261

Table 4.88: Comfort index tabulation for different road Types – Posture2

		Variation % respect rigid stem- Posture 1			
		Harshness			
		ZERO	A	B	C
Stiffness	SOFT	109,823	0,121	3,347	0,729
	MEDIUM	38,991	7,764	3,062	0,959
	HARD	61,667	2,973	4,362536	-0,076

Table 4.89: Comfort index variation % respect rigid stem tabulation – Posture2

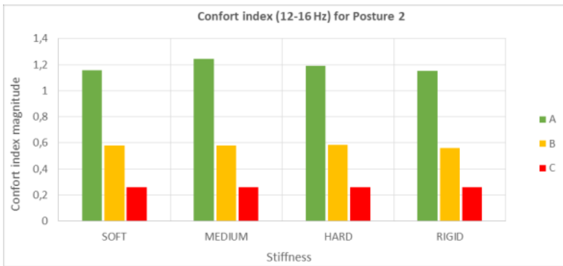

Figure 4.90: Comfort index – Posture2

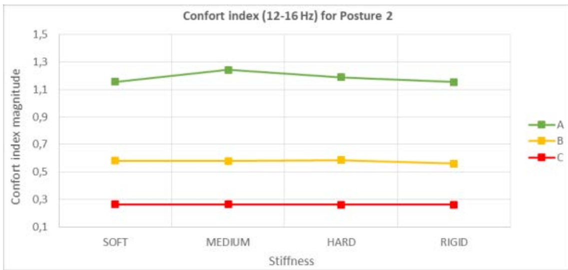

Figure 4.91: Comfort index grapic – Posture2

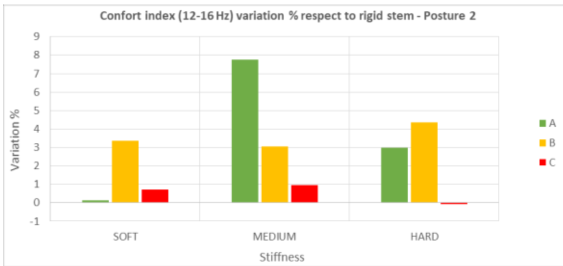

Figure 4.92: CI variation respect to rigid – Posture2

Figure 4.93: CI variation respect to rigid – Posture2

For the Posture2 it is evident that the comfort index is almost always positive when compared to the RIGID configuration. An exception is made for the roadC and HARD setting.

Transfer function detail: For each posture configuration (Posture1 and Posture2), for each stem configuration (soft, medium, hard, rigid) and for each type of road (A, B, C) it is noted that the comfort index decreases passing from road A to C. Below is a detail of the transfer function in the case of stem soft which highlights the fact that the integral of the transfer function decreases (between 12 and 16 Hz) passing from path A to C. This is valid for each configuration of test. This fact is due to the greater severity of road C compared to A in terms of displacement value.

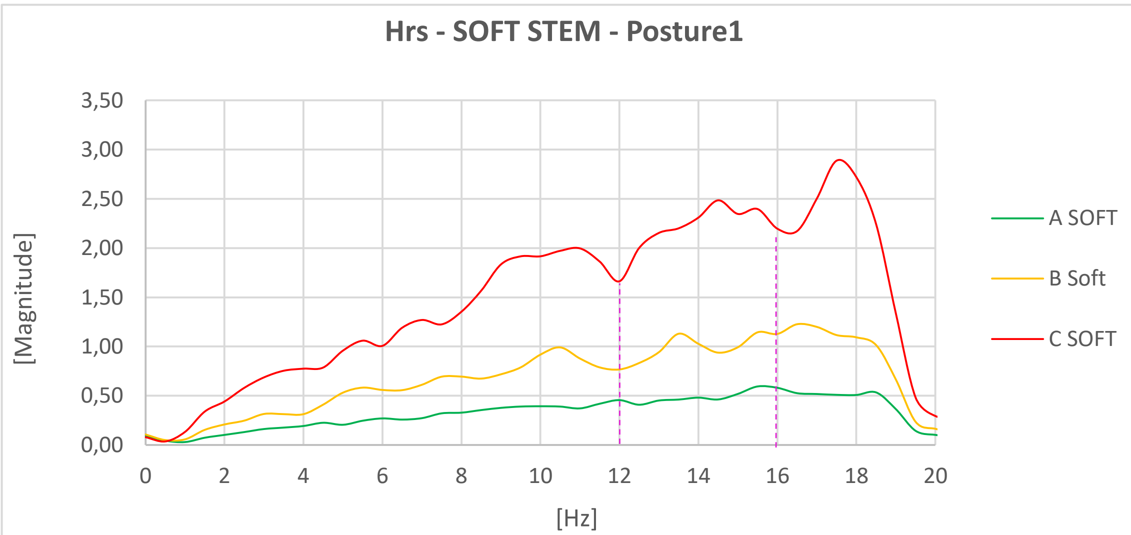

Figure 4.94: Transfer function comparison graphic for soft stem and posture 1 (A,B,C Road)

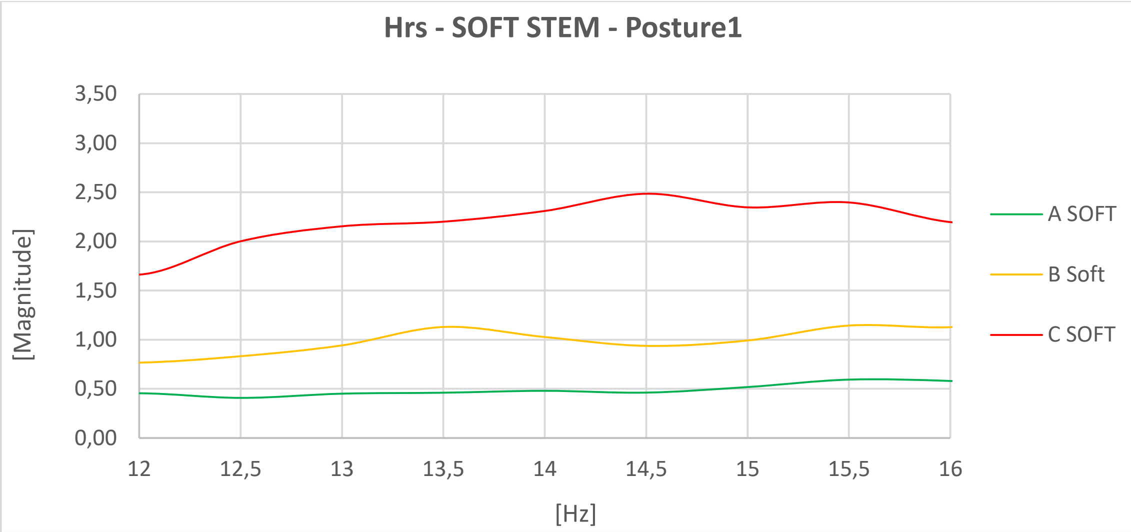

Figure 4.95: Detail of transfer function comparison graphic for soft stem and posture 1 (A,B,C Road)

Overall comparison of Comfort index and transmission index variation % respect rigid stem: Considering Comfort index e Transmission index variation % respect to RIGID stem a comparative table has been compiled to allow an overall view of the behaviour of the suspension stem. Red cells represent a condition of lower performance respect rigid stem and green cells represent better performance respect to rigid stem.

		Road A		Road B		Road C	
		CI	TI	CI	TI	CI	TI
Posture1	SOFT	6,04	82,63	0,50	38,33	-2,66	-12,95
	MEDIUM	0,61	86,15	1,78	40,79	-3,52	-9,96
	HARD	1,05	132,47	2,81	12,47	-2,10	-10,48
Posture2	SOFT	0,12	0,71	3,35	-15,37	0,73	-25,84
	MEDIUM	7,76	-5,97	3,06	-16,47	0,96	-24,80
	HARD	2,97	-6,43	4,36	-7,30	-0,08	5,22

Table 4.96: Comparison teble for CI e TI variation % respect rigid stem

Note:

Transmission index TI>0 (means higher acceleration value respect rigid stem) represent a worst condition

Comfort index CI>0 (means a better value of comfort index) represent a better condition

4.4.2 OUTDOOR TEST SESSION

Subjective qualitative assessments: To allow an evaluation of the system under examination, a mixed path was studied consisting of tramac, cobblestone and gravel and with curves to verify the change of direction. After carrying out the test run, the system settings were changed. At the end of the four tests the table was compiled according to the indications of the tester.

Test and evaluation circuit:

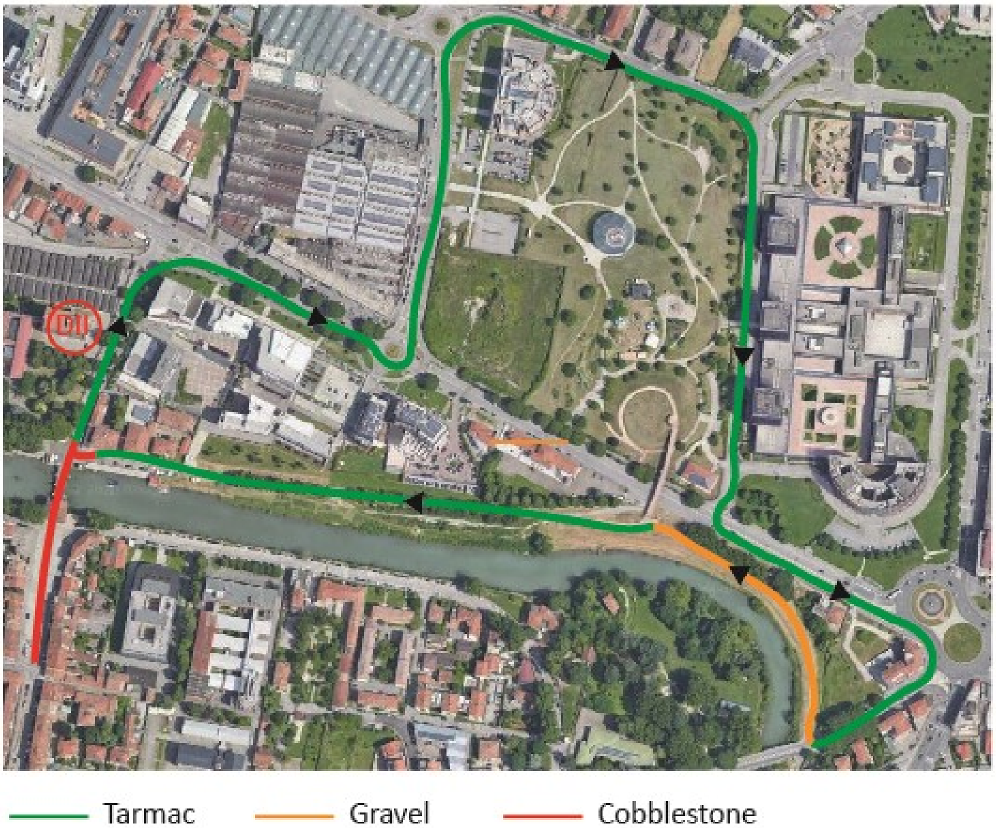

Figure 4.97: Test circuit map representation

Figure 4.98: Tarmac surface tested: Via Venezia,Via Pescarotto,Via Pietro Maroncelli, Piazzale Stanga

Figure 4.99: Gravel surface tested, Lungargine del Piovego, Padova

Figure 4.100: Gravel 2 surface tested, Lungargine del Piovego, Padova

Figure 4.101 :Cobblestone surface tested, Via del Portello, Padova

After each test the following scores have been attributed by the tester for each stem setting:

	Soft *Stem 50/60*	*Medium* *Stem70/60*	*Hard* *Stem90/50*	*Rigid* *Standard stem*
Driveability	7	9	9.5	10
Precision (cornering)	6	8	9	10
Accuracy	6	8	9	10
Readiness (scatto)	5	7	9	10
In vivo damping (VIBRATIONAL COMFORT)	8	4	1	0
Safety Feeling (braking)	5	8	10	10

Table 4.102: Score attribution fof outdoor test

- *Driveability: overall assessment of the ability to be used to perform its typical functions (pedaling, change of direction, braking ...)*
- *Precision: ability to respond to micro movements*
- *Accuracy: ability to follow the imposed route*
- *Readness: readiness during the shot going from sitting to raised position with load on the arms*
- *In vivo damping: effectiveness of damping while running*
- *Safety Felling: feeling of stability, firm grip and control during braking*

Below is the comparison with indoor tests relating to the only comparable parameter

	Soft *Stem 50/60*	*Medium* *Stem70/60*	*Hard* *Stem90/50*	*Rigid* *Standard stem*
In vivo damping (VIBRATIONAL COMFORT)	8	4	1	0

Table 4.102: Score attribution fof indoor test

4.5 DISCUSSION OF THE RESULTS

The analysis of the acquired and processed data relating to the load cell (forces and displacements) demonstrated system stability under varying test conditions.
With actuators in displacement control and with average values of force F2 aligned in the average value for the various configurations, they testify to the fact that the cylinders substantially compensate for the inertia of the rollers.

As regards the effective effectiveness of the suspension stem, there is no evident benefit in terms of its use when compared with the quantities studied, ie TI (transmission index) and CI (comfort index).

The introduction of a damping and a compliance in the system certainly modifies its behavior on a global level. (observe the variation of the hub accelerations).

From the observation of the comparative table (4.96) the best results in terms of reduction of the TI compared to the rigid stem and of the increase of the CI, would seem to be obtainable for Posture2.
For Posture1, on the other hand, a behavior is observed that contrasts the CI and the TI.

For road A and B, the CI is rewarding while (the system attenuates the effect of vibrations between 12 and 18 Hz) while the TI records an increase in the level of acceleration at the stem. The opposite happens for the road C for which the TI guarantees the presence of a lower average acceleration level at the stem.

The aspect of the forces that must be investigated through the sensorization of the handlebars and which could make the problem better understood but above all provide an additional element for the assessment of perceived comfort.

Most likely the sensation and perception of comfort cannot be reduced to a single index because each of them represents a part of the problem. The CI focuses on biomechanically harmful and annoying frequencies while the TI describes phenomena of amplification of accelerations. From this perspective, the analysis of forces could provide an important contribution.
This could be important to link the quantitative quantities acquired to the subjective evaluations which are substantially positive and favorable.
The problem is not trivial especially if we consider the variability of posture because the system varies its geometry but also the mechanical characteristics. (Posture1 greater load on the handlebar, greater stiffness and less damping - Posture2 less load on the handlebar, less stiffness and less damping). This significant change can be the key to the fact that the system allows better performance for Posture2 which, however, is not the one mainly used during displacement pedaling.

Since pedaling takes place mainly in Posture1 it would be interesting to find the correct combination of elastomers in order to obtain an improvement in CI and TI for this posture.

The rheological characterization of elastometers is certainly necessary to define the elastic and damping characteristics in order to understand their static and dynamic response. The test campaign could include tests without the mixed use of elastomers but coupled between peers (elastomers two by two equal: 50-50; 60-60; 70-70; 80-80; 90-90). Tests carried out in this way would allow the elimination of a variable to the problem.

As regards the tests that can be performed, it would be desirable to be able to compare the results obtained from indoor tests with outdoor tests and acquisitions and in such a number as to allow the calculation of a statistical error. This would allow to confirm or deny the results but above all to highlight any influences of the test bench.

5 FURTHER DEVELOPMENT

The improvements related to the cushioned stem study were discussed in the previous chapter. From a technical point of view, improvements are possible for the test bench and concern the possibility of reducing the inertia of the rollers. The possibility of using commercial rollers in composite material is proposed below.

Widther rollers (width= 800mm)

Total mass: 21,46 KG

Alluminium Rolls mass = 0,78m*2=1,56m*6.62Kg/m=10,32Kg

Structure mass : 21,46Kg-10,32Kg=11,14 Kg

Composite materials rollers (Carbon fiber): 1,17 kg/m*0,78*2m=1.82Kg

New mass:12,96kg

Reduction rate :39,60%

Smaller rollers: (width = 440 mm)

Total mass: 14,13 Kg

Alluminium Rolls mass: 0,44m*2=0.88m *6,62 Kg/m=5,82Kg

Structure mass: 14,13Kg-5,82Kg=8,31 Kg

Composite materials rollers (Carbon fiber): 1,17Kg/m*0,44*2=1,03Kg

New mass:9,34Kg

Reduction rate:34%

Currently the test bench can be improved in the smoothness of the rollers. However, this involves carrying out a free pedaling test. It is desirable to implement one of the proposed braking systems.

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Sitography:

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7 APPENDIX

PSD diagram relative to road ZERO – Posture 1

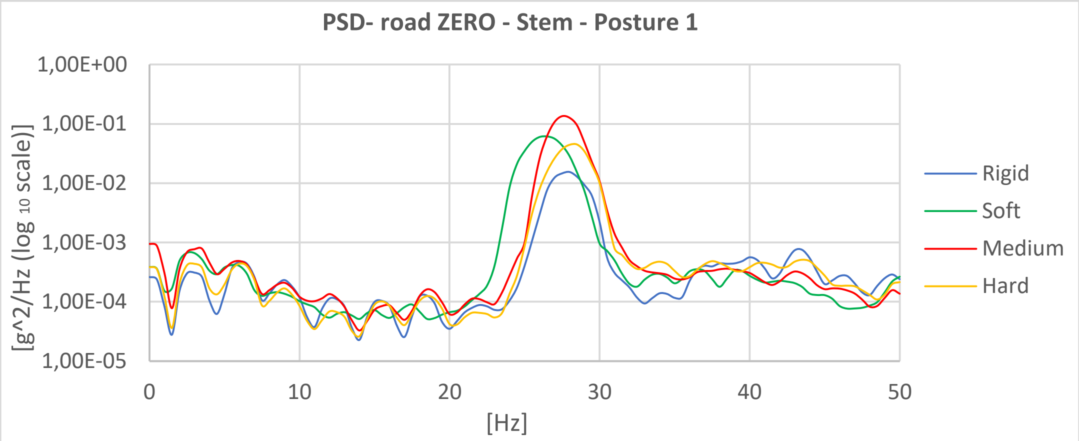
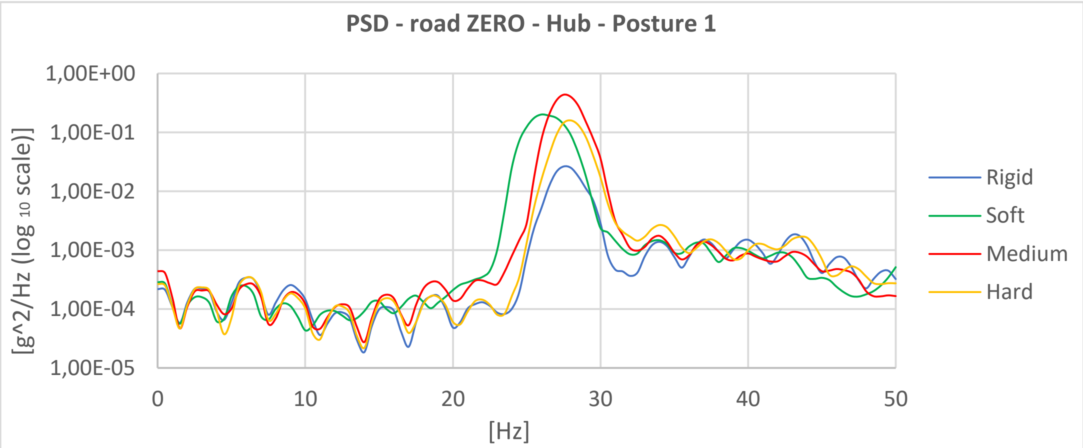


Figure A1



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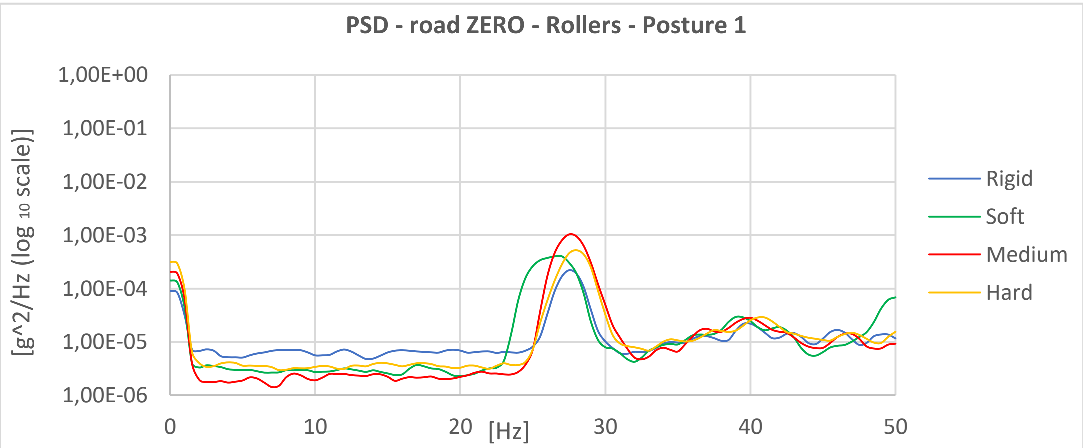
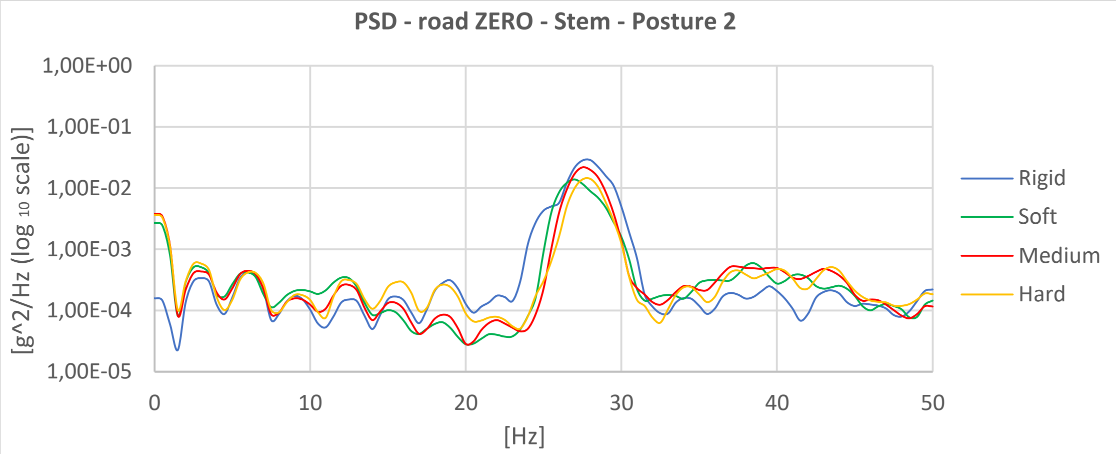


Figure A3

PSD diagram relative to road ZERO – Posture 2



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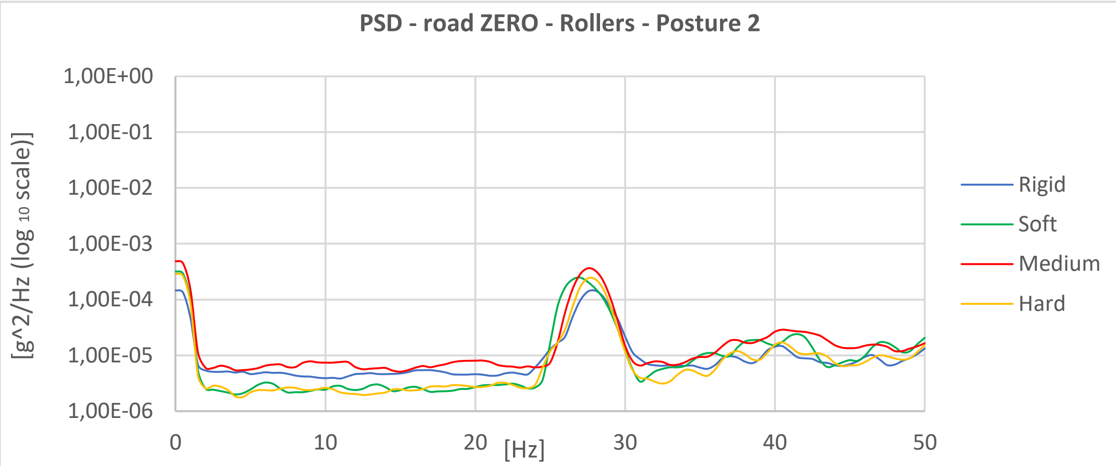
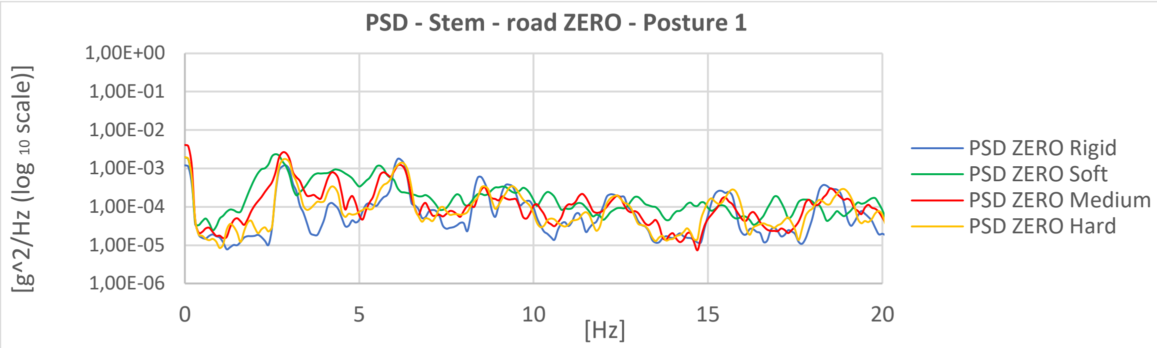


Figure A5

Figure A6

PSD for STEM – Posture1



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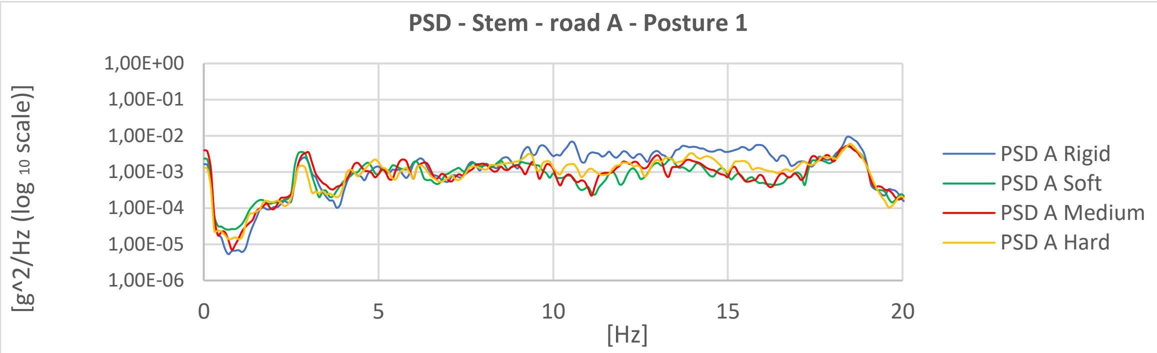

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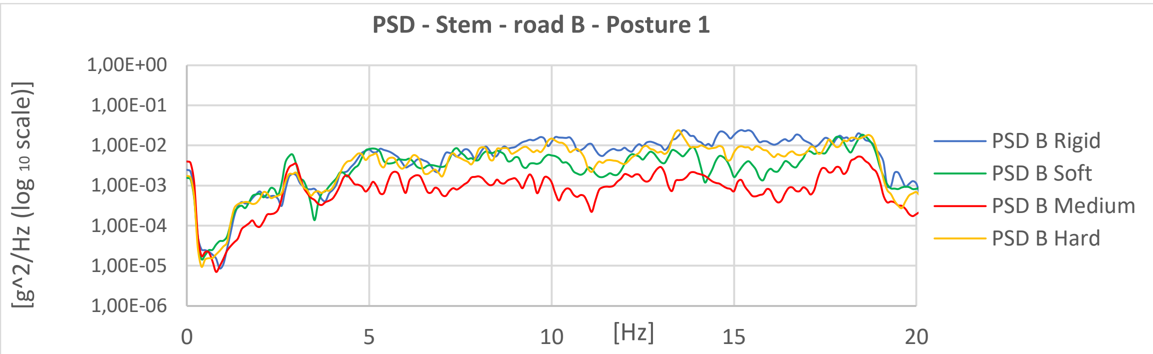

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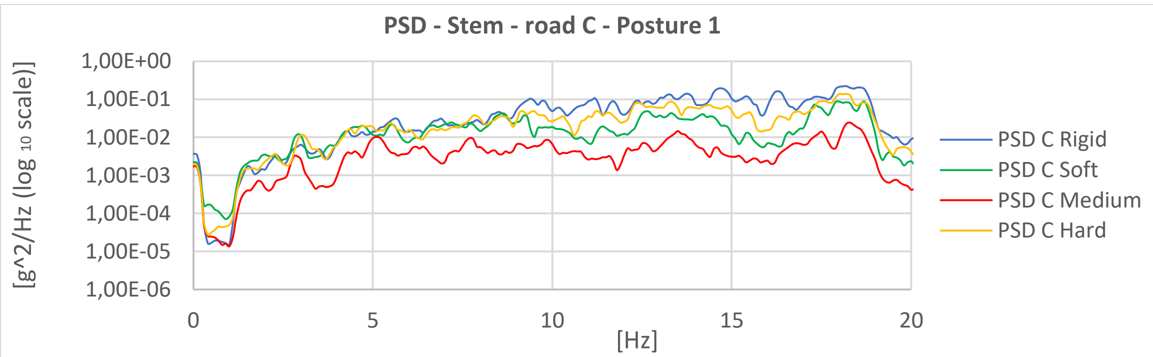
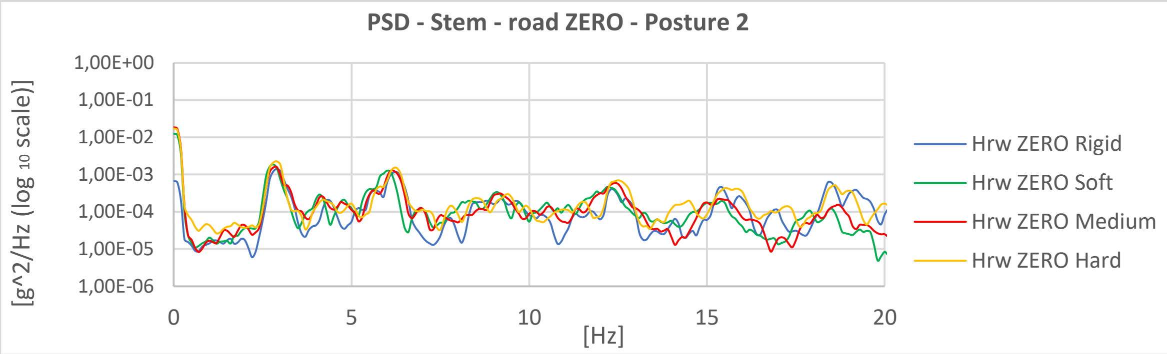


Figure A10

PSD for STEM - Posture2



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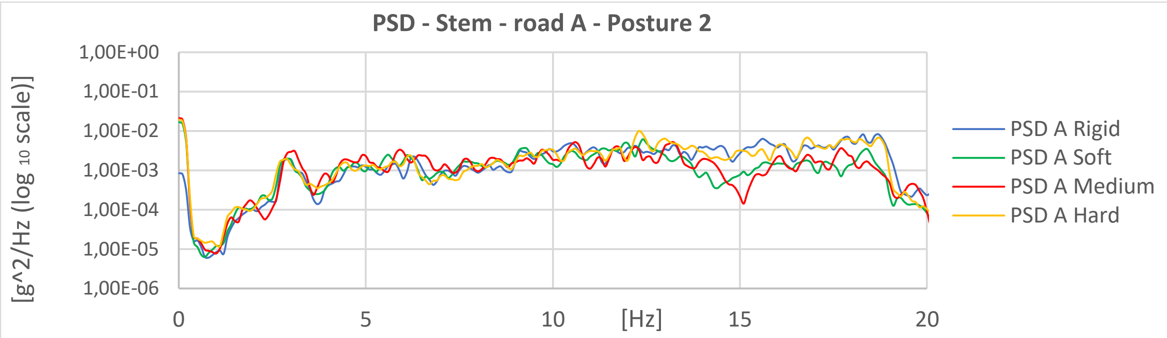

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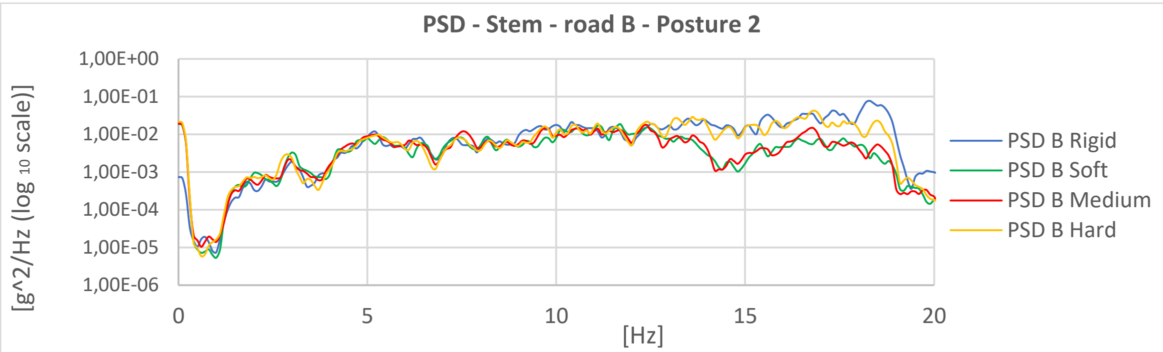

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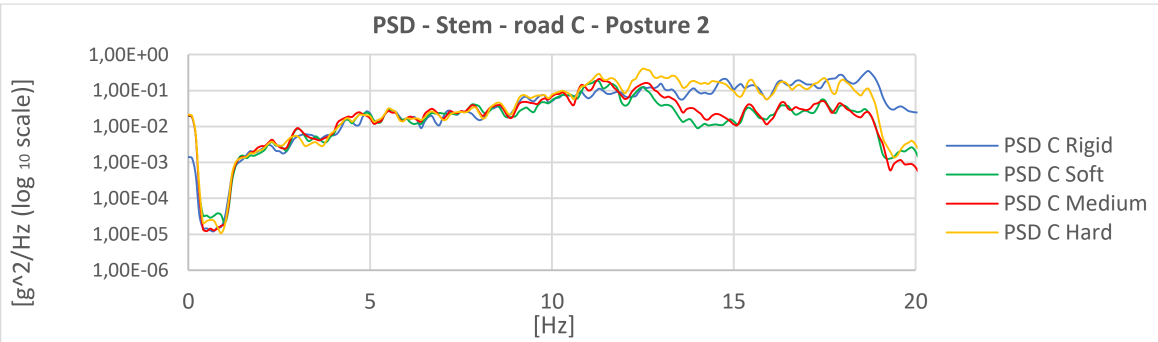
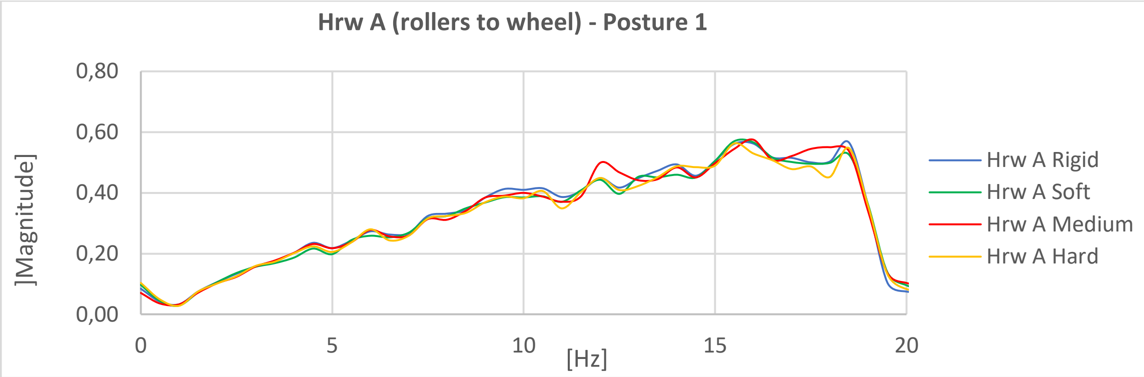


Figure A14

Transfer function - Hrw - Posture1



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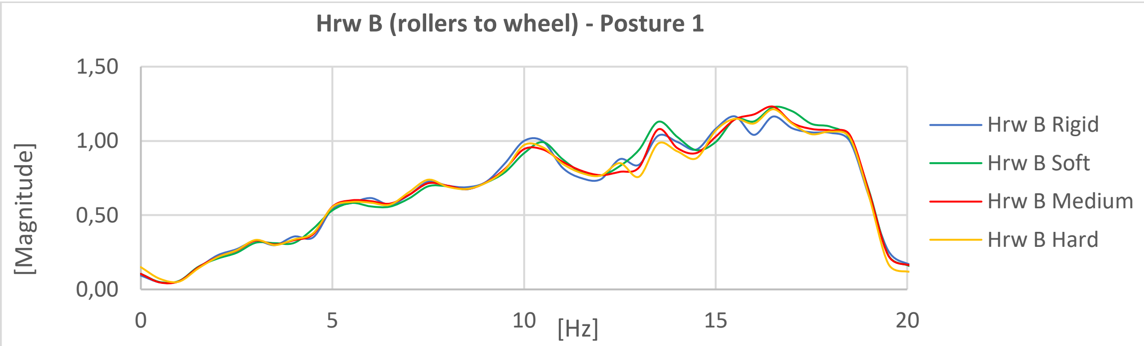

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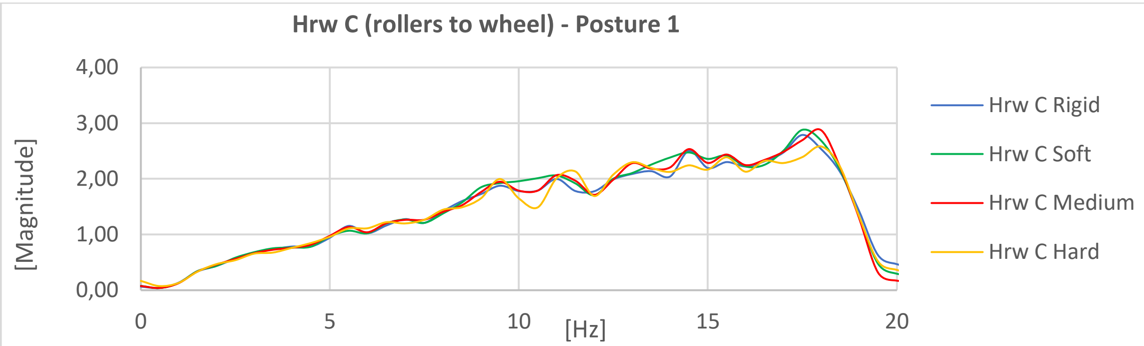

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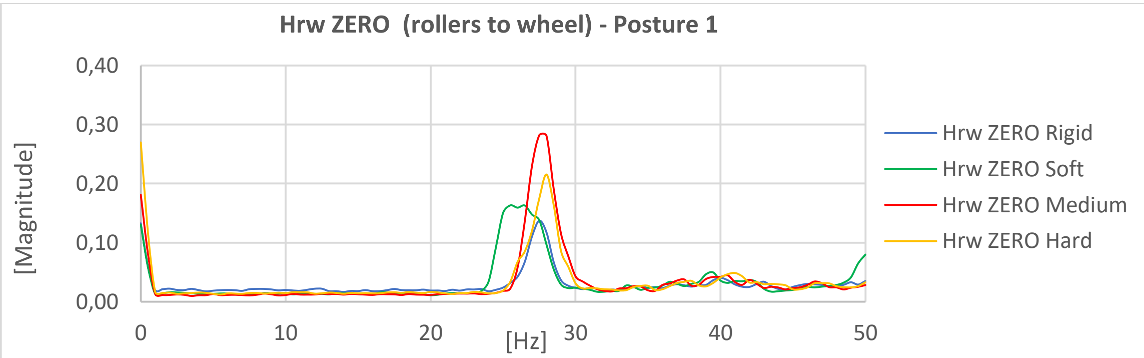
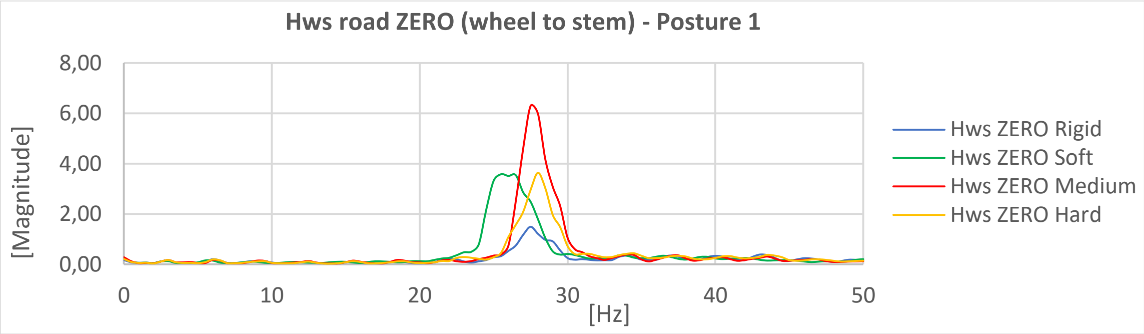


Figure A18

Transfer function - Hws - Posture1



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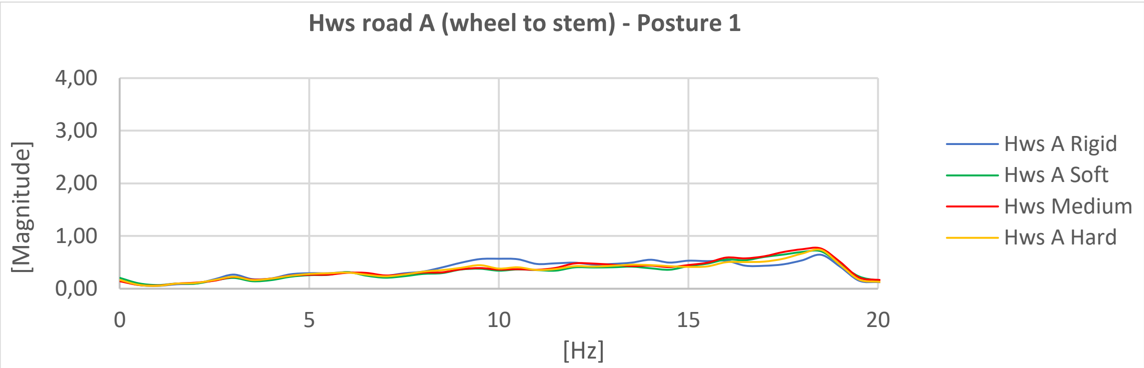

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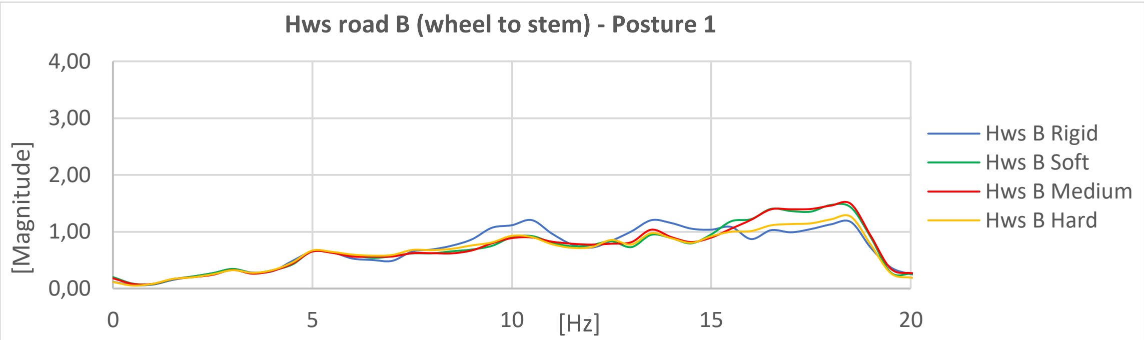

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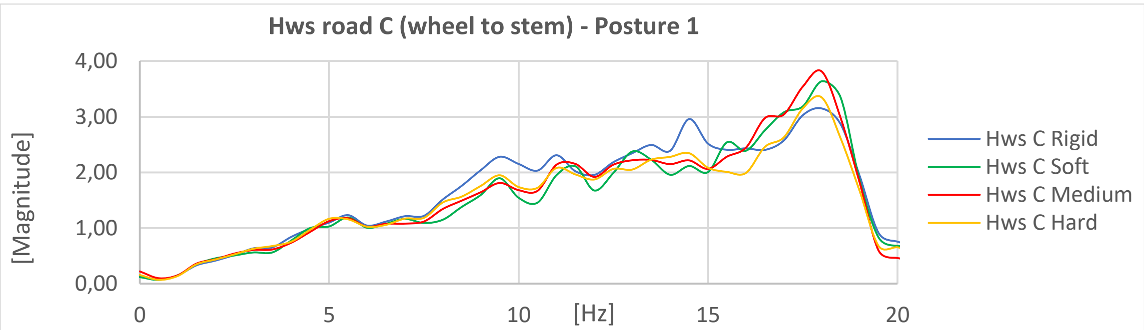
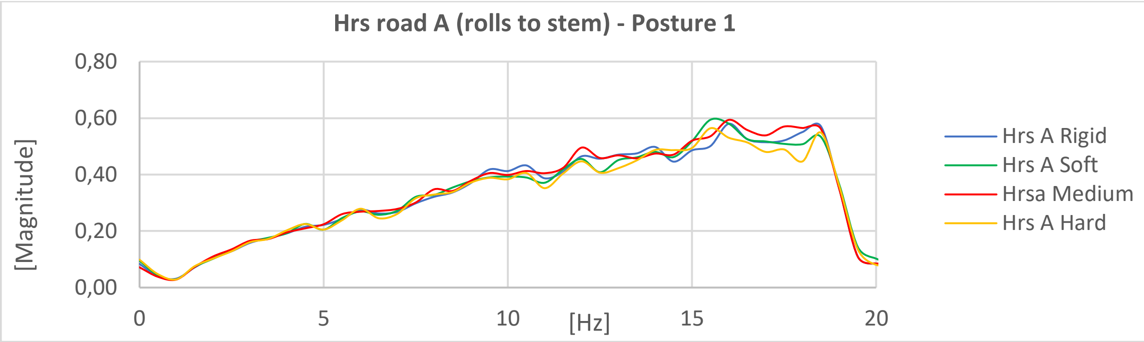


Figure A22

Transfer function - Hrs - Posture1



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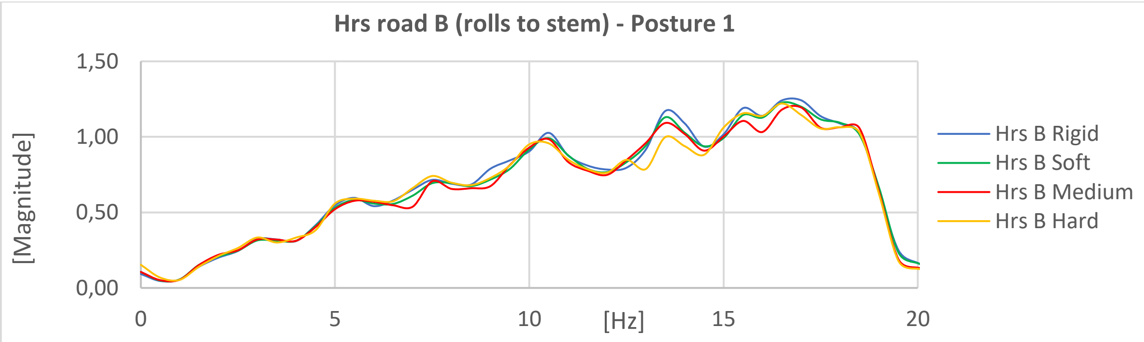

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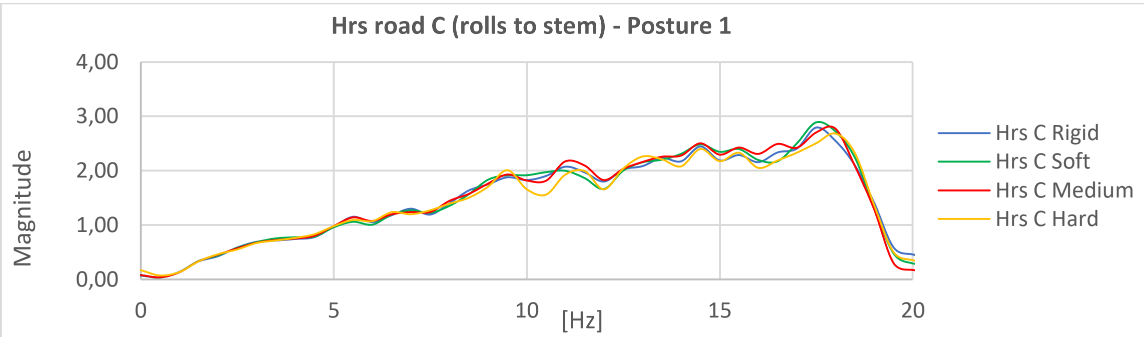

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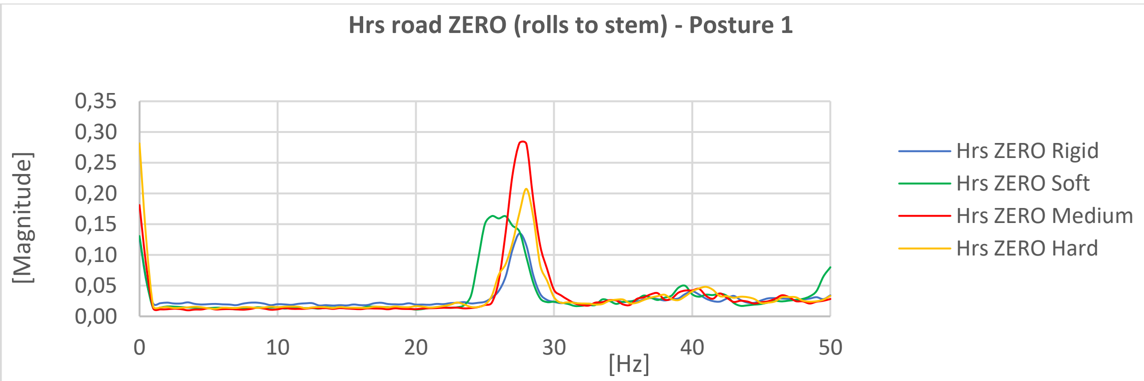
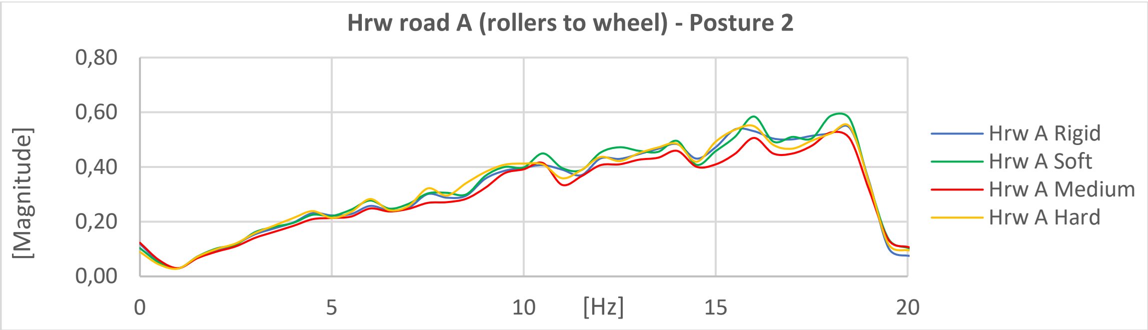


Figure A26

Transfer function - Hrw – Posture2



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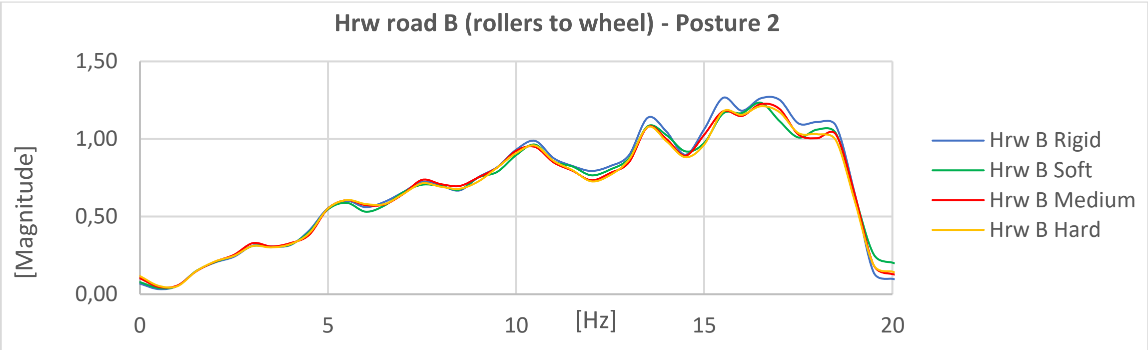

Figure A28

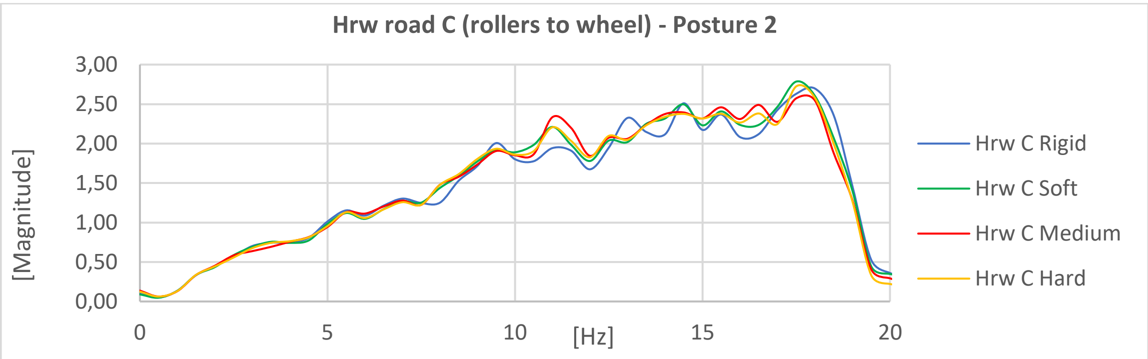

Figure A29

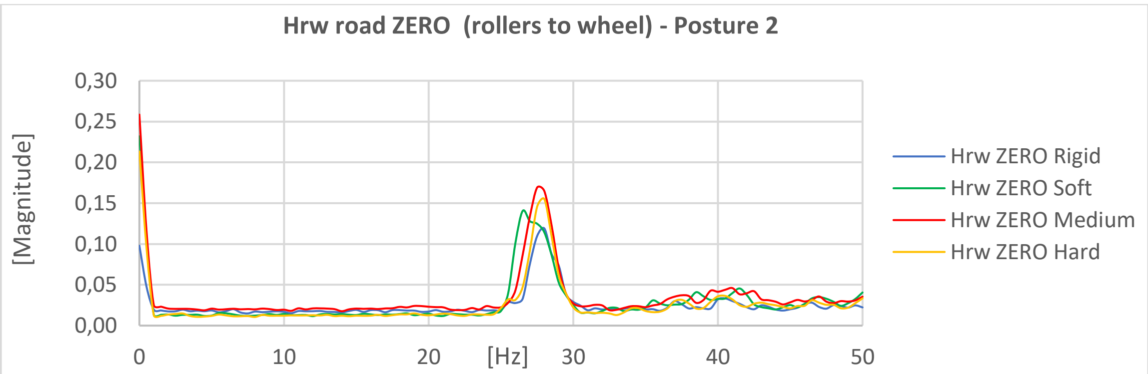

Figure A30

Transfer function – Hws – Posture2

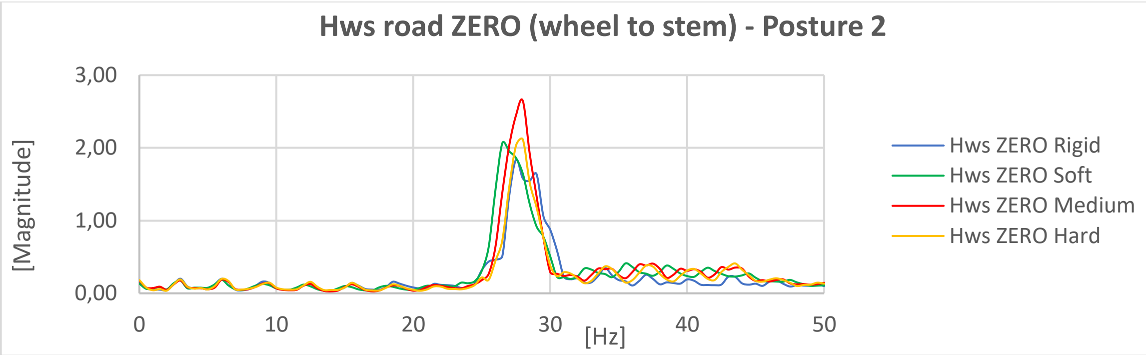

Figure A31

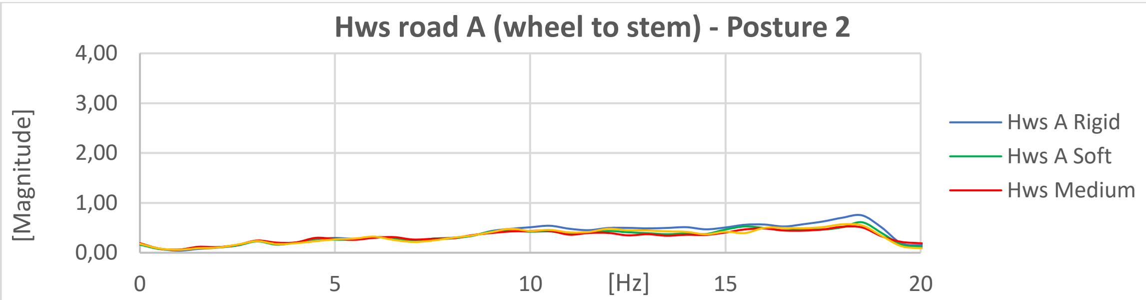

Figure A32

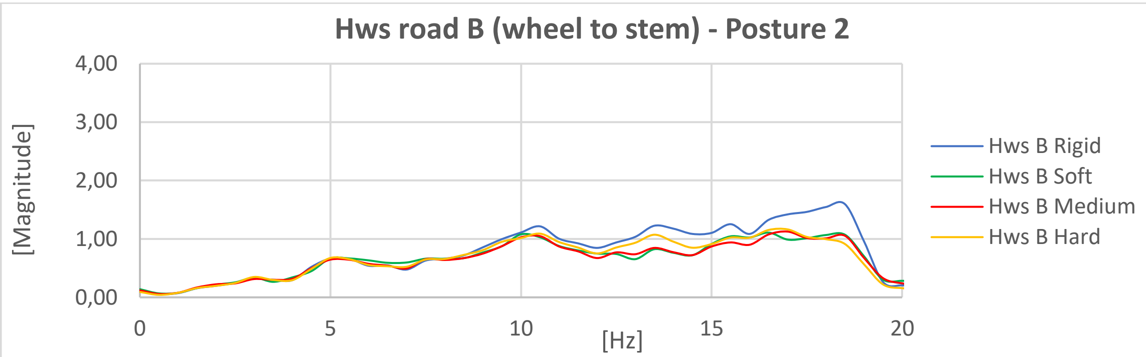

Figure A33

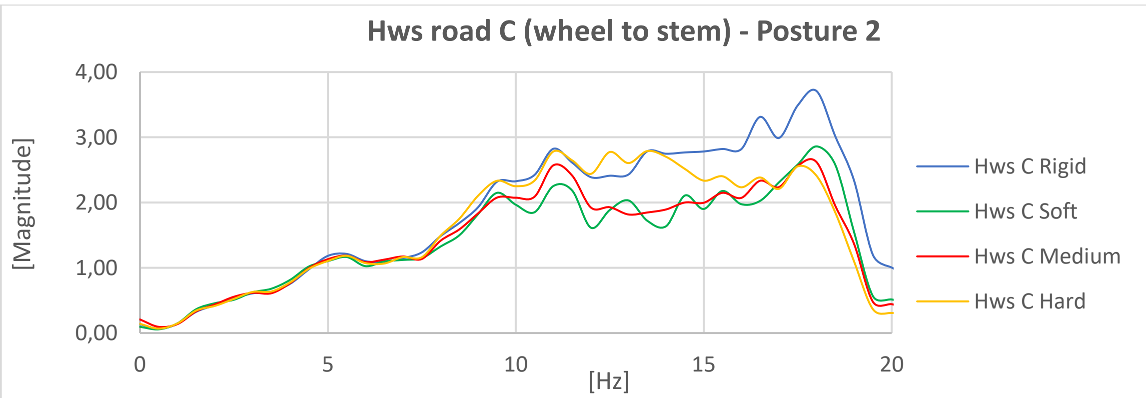

Figure A34

Transfer function - Hrs – Posture2

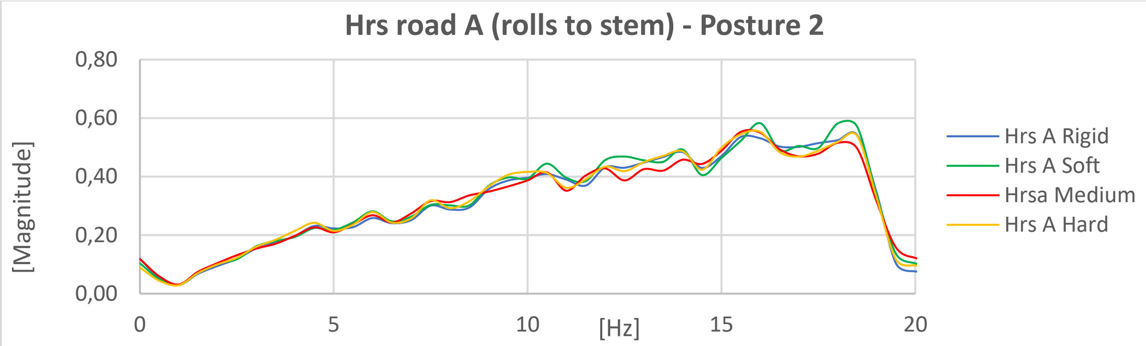

Figure A35

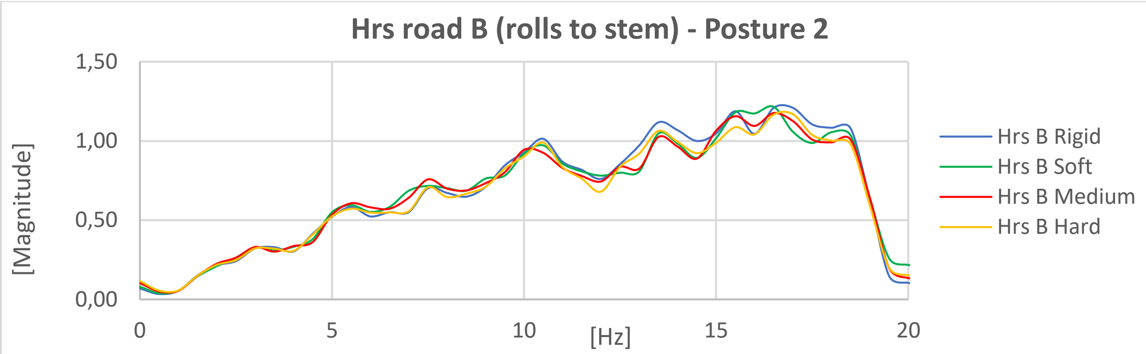

Figure A36

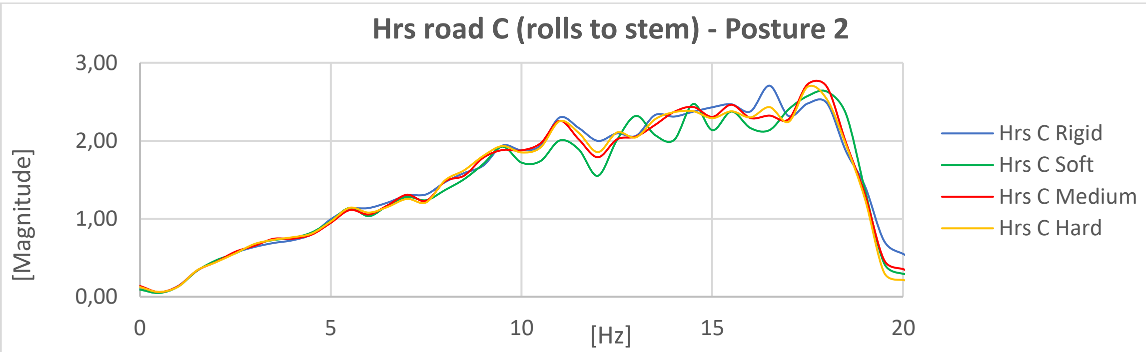

Figure A37

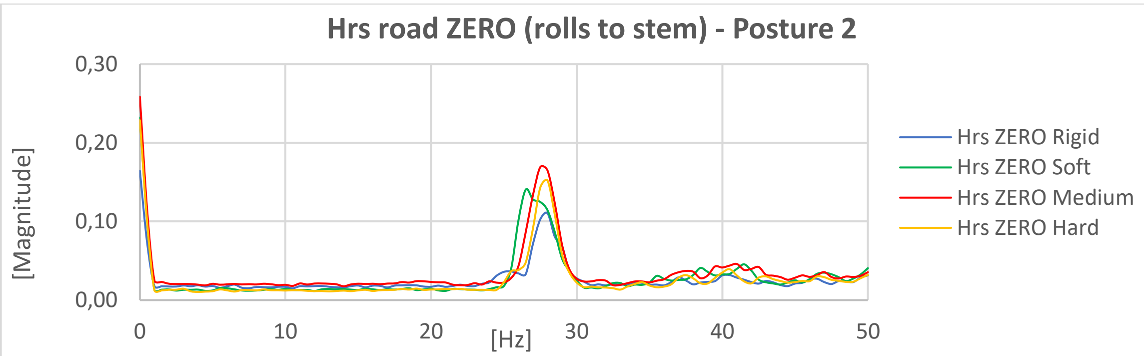

Figure A38

RINGRAZIAMENTI

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