Politecnico di Torino

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Tesi di Laurea Magistrale

Design of a Speed Multiplier for regenerative Shock Absorber

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Anno Accademico 2017 - 2018

Abstract

Preliminary design of a speed multiplier for regenerative shock absorber in vehicle suspension is carried out. Electromechanical Rotary Shock Absorber is considered, which is made up by leverage that converts vertical displacement given by wheel stroke in angular displacement as input to gearbox, that increase this angular speed for electric generator. Four leverage arrangements are investigated referring to a given double wishbone front suspension. Then two type of gearbox configurations will be analysed: fixed axes configuration and planetary configuration. KISSsoft has been used for gearbox solutions design. Gear pairs, shafts and bearings have been sized considering infinite life according to ISO 6336 Standards. Four parameters have been considered in this trade-off analysis so as to evaluate the best configuration: efficiency, overall dimension, Sound Level Pressure and equivalent mass. Standard ISO 14179 has been followed to compute power losses sources in gearbox configurations, which defines power losses due to: gears meshing, bearings and churning losses. Then a simplified approach proposed by Masuda has been followed for Sound Level Pressure prediction. Finally, working conditions at different Torque and Speed have been considered to evaluate Sound Level Pressure and efficiency map, hence an indication on energy harvesting potential from road roughness by means of Electromechanical Rotary damper.

Contents

Chapter 1

Introduction

The damping force in autovehicle suspension is necessary to accomplish one of its main functions, that is to isolate the vehicle from vibrations caused by the ground roughness. In standard viscous *Shock Absorbers*, the damping effect bring about irreversible energy loss in the form of heat, starting from the kinetic vibration energy of the suspension due to bump on the road profile. As claimed in [1] only the 12%-30% of the total available energy is used to move the vehicle. The rest of the energy is lost to engine, driveline inefficiencies or used to power accessories and to damper as well. Quantify energy losses in dampers is not simple, however according to [2], some findings point out that the amount of energy loss is about 4-10% of the total energy used to move the vehicle following an analytical approach. This amount of energy is function of the stiffness, the speed and characteristics of the vehicle and ground roughness profile. Segel in [3] analyzed the power dissipation in dampers of passengers car. The results point out that the power dissipation by four dampers is around 200 W when the vehicle is running on poor profile road at 13 m/s. Nowadays the economic and environment requirements push the research and technology development efforts to find new solutions in orders to improve the vehicle efficiency and then fuel consumption reduction. In order to obtain these objectives one possible solution could be energy harvesting from vehicle suspensions, that is *Regenerative Shock Absorber*.

1.1 Review on Regenerative Shock Absorbers

Regenerative Shock Absorbers (RSA) main purpose is to convert and harvest kinetic vibration energy of the suspension during the stroke, in different form of energy. Suspensions equipped with this type of damper are semi-active suspension. Beyond vibrations energy harvesting is possible contemporary to control the damping force, so that vehicle handling and rider's comfort are improved because the control system is able to adapt the suspension to different road profile. As described from Zhang Jin-qiu in [4] the RSA can be divided in two types according to the working principle:*Mechanical* and *Electromechanical*.

Mechanical Shock Absorbers capture the kinetic energy from the suspension and convert into potential hydraulic/pneumatic energy to be stored in accumulator. However, these hydraulic/pneumatic systems have some disadvantages: additional weight and installation space, hose leaks and ruptures may disable the whole system, the responding bandwidth of hydraulic/pneumatic systems is narrow, which confines the suspension performance.

Electromagnetic Shock Absorbers on the other hand are able to restore kinetic energy from the wheel as electric energy. This type of energy is more convenient to store and reuse, so that the efficiency of the device as a whole is increased. In recent times ESA attract interest for development of new solutions in regenerative suspension. These systems are equipped with Permanent Magnets Motor that can works as actuator or generator. The formal provides active force and the latter damping force. Is possible to change damping force by changing the shunt resistance. Is possible to distinguish different type of EMS considering structure configuration:

- *Direct-Drive Electromagnetic Suspension*
  In this structure configuration the traditional hydraulic *Shock Absorber* is replaced by linear permanent magnet motor. The mechanical energy from the wheel is converted in electric energy needing no transmission devices. This solution can't have high energy recovery efficiency because any speed increaser is present.

- *Ball Screw Electromagnetic Suspension*
  In this structure configuration a ball screw is used to convert the linear motion of the wheel into rotation.

- *Rack-Pinion Electromagnetic Suspension*
  Similar to previus case, Rack-Pinion is used to convert linear motion into rotation. Zhongjie Li in [5] developed and tested a prototype in which rack-pinion, bevel gears to change orientation of motion axes and gearbox were used. The average power values are 4.8 and 3.3 W, respectively, at 48 km/h and 32 km/h, or 19.2 and 13.2 W can be harvested on four shock absorbers at 48 and 32 km/h.

- *Planetary Gear Electromagnetic Suspension*
  Planetary gear set is helpful to increase motor efficiency and active force. In this solution gearbox is used to improve regenerative efficiency. A seen before generally a gear box to increase angular speed is used in addiction to ball screw or rack pinion.

- *Hydraulic Transmission Electromagnetic Suspension*
  In this configuration solution is combined hydraulic transmission and electric motor. Levant Power Corp. developed a damper called GenShock, combining hydraulic transmission and electric motor. A suit of rectifying pipe guarantees that the hydraulic motor driven by fluid rotates by a consistent direction

whatever the piston runs up or down. Because the rotation direction of electric motor doesn't alternate frequently, the regenerative efficiency is enhanced obviously.

As seen there are two conflicting aspects to be considered : *Speed* in input to Electric Motor and *Energy Recovery Efficiency*. The formal is possible to be increased using speed increaser but this leads with more inertia loss and then the latter to be decreased.

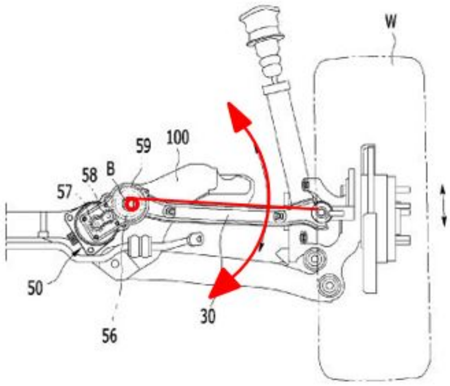
1.2 Review on Rotary Damper

In this section different *Rotary Damper* patents are reported. Four different patents are listed in order to show the state of art about *Regenerative Rotary Shock Absorber*. Many patents about this kind of damper are available today. However, to avoid repetitions, are reported 4 solutions; each of them contains particular features for what concerned the way in which the vertical displacement is converted in angular displacement as input for gearbox and architecture of speed multiplier. Furthermore, this research has been carried out in order to have informations about gearbox architecture. It is clear that gearbox architecture depends on speed ratio i that is necessary to perform as consequence of leverage transmission ratio τ, and this value depends on the electric motor characteristics as well. However, in claim of these patents are not included performances of the electric machine or transmission ratio given by gearbox. Anyway, they are useful to have a current general overview about leverage arrangements and speed multiplier solutions that is possible to use in *Rotary Shock Absorbers*.

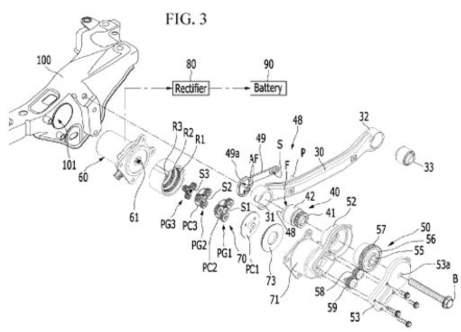
1.2.1 Hyundai [DE102013225356] 2014

This patent is illustrated in Figure 1.1a. As highlighted by the red arrows, this system uses the kinematics given by upper arm to convert vertical stroke in rotational displacement. SO the leverage is given by upper arm. Electric motor and gearbox are indicated by element, respectively 59 and 57. Afterwards, the electric motor is connected to a rectifier for AC/DC coversion and finally to the battery. In this system is also present a one-way clutch, indicated by element 56. This clutch can be configured to transmit power in bumping or rebounding from the wheel stroke.

From the second exploited view of Figure 1.1b is possible to figure out the architecture of this system, in particular for what concerned the gearbox. From the claim of this patent is possible to figure out that are present: 1 parallel axis stage (57,59) and then 3 planetary stages.(PC1-PG3). Interestingly, can be notice that also a hydraulic tube is present in this solution, for this reason is present the one way-clutch system . Traditional and rotary damper can share damping task at the same time. On the other hand it is not pointed out where is positioned the stiffness element.



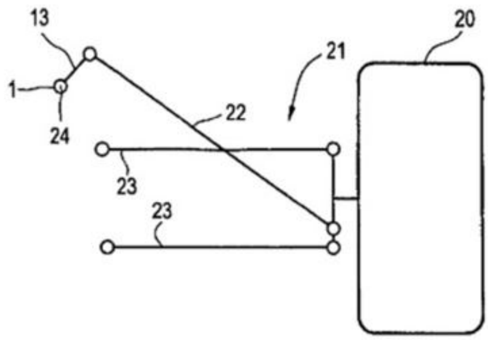
(a) *Leverage arrangement Hyundai*



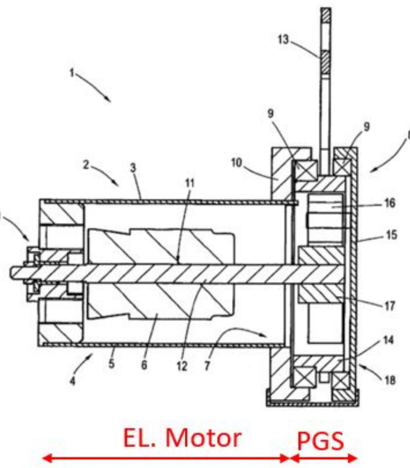
(b) *Exploited view*

1.2.2 Audi [DE102011102743] 2014

This patent patent from Audi is illustrated as follows. In 1.1a is sketched the double wishbone suspension as well as additional levers given by element 22, 13. Four bar linkage is obtained considering levers 23, 22, 13. The follower is element 13 that pivots about point 1 in which are placed gearbox and electric motor. In this patent is not claimed where is located spring element. In 1.1b is illustrated a section of the follower, gearbox and electric motor. This patent claims that is possible to realize this kind of gearbox by means of different architectures such us *Planetary Gears*, *Strain Wave Gearing* and *Cycloid gears*. However, in this patent are reported the configuration with planetary set as illustrated in 1.1b. In case 8 is included the gearbox which is a single stage planetary set, instead in case 2 is present the electric machine. A sees in previous patent, the electric motor is connected with rectifier and then battery.



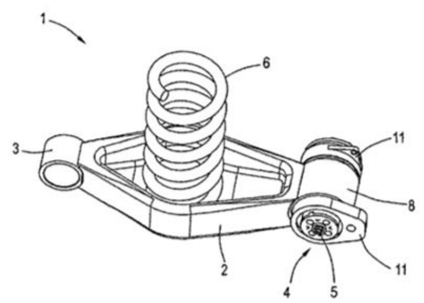
(a) *Leverage arrangement Audi*



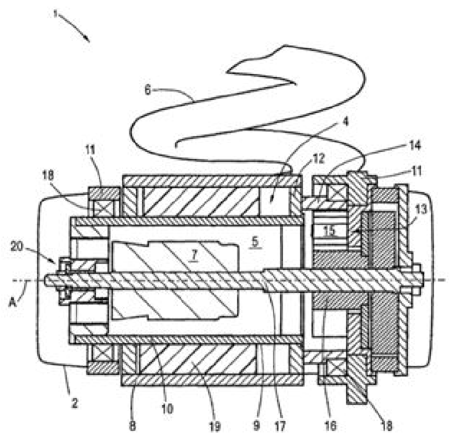
(b) *Secion view*

1.2.3 Audi [US9136743] 2013

This patent is illustrated in Figure 1.1a. The particular feature of this solution is that the *Rotary Damper* is integrated directly into the support of the suspension arm and noadditional leverage system is needed. The big advantage of this solution is the high compactness and lightweight. On the other hand the transmission ratio given by this kind of arrangement is surely very small. In Figure 1.1b is illustrated a section view of this patent. Similarly to the previous patent from Audi, is claimed that is possible to use different gearbox architecture. However, they illustrates the solution with 1 stage planetary set.



(a) *Leverage arrangement Audi*



(b) *Secion view*

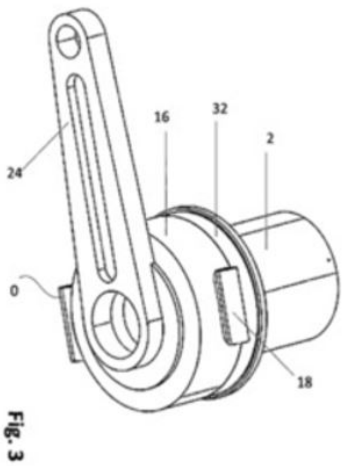
1.2.4 Audi [US9136743] 2013

This patent illustrated in Figure 1.1a uses a simple leverage system composed by a single lever, indicated by element 24.

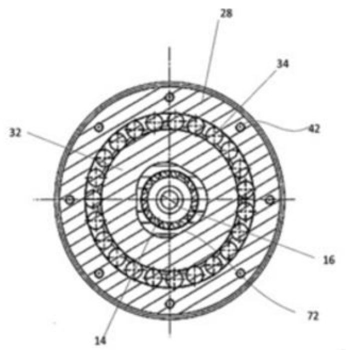
The particular feature of this patent is that it uses an Harmonic wave gearbox: The wavemaker is pointed out by number 32, the rotor is element 10, and the flexible unit is element 30. In this patent it is also indicated the multiplication ratio that this kind of gearbox performs which is equal to 1:50.

1.3 Objective of this thesis

The title of this thesis is *Design of speed multiplier for Regenerative Shock Absorber*. The focus is on the feasibility of electromechanical regenerative shock absorber similar to previous patents, giving particular attention on the mechanical part of this damper, made up of Leverage System and Gearbox. In the following paragraphs will be discussed both aspects, with reference to a given front suspension scheme that will be illustrated, and a given electric motor characteristic. The starting point is to



(a) *Harmonic Audi Patent*



(b) *Harmonic Gearbox*

analyse different type of leverage arrangements in order to obtain requirements for gearbox in term of speed ratio. For what concerned leverage system, different parameters have been considered: *Transmission Angle* and *Speed Ratio*. Transmission Angle is a parameter that can measure, in some way, the efficiency of the leverage system. Speed ratio can be defined as [mm/rad], in which millimeters measure the vertical stroke of the wheel and radiants measure the angular displacement of the input shaft of the gearbox. Damping force depends linearly on the angular speed of the electric machine, so the transmission ratio has to be as small as possible. In the paragraph referred to Leverage Arrangement, different solution have been analysed and, as consequence, the transmission ratio required from the gearbox is calculated. The first configuration that has been taken into account is a simple parallel axes gearbox, made up by 3 stage as illustrated below: the value of transmission ratio that is possible to reach with this configuration is around 125 considering transmission ratio equal to 5 per each stage. The drawback of this configuration is the high overall dimension. For this reason has been introduced power split in order to reduce module required and then overall diameter. This will be investigated as *Fixed Axes configuration*. Then planetary set has been investigated. According to catalogue from producer of speed increaser used in wind turbine gearbox, there is not a preferable configuration. It has to be noticed that in wind turbine gearbox large weight is not a big problem. However, in order to have an indication by speed increaser applications, *Planetary Configuration* has been considered. In particular, planetary set made up by 2 stages.

KISSsoft has been used for gearbox design, considering has input data Speed and Torque that are given by Solution 3 of the leverage analysis. For reasons that are discussed in the Leverage Design Paragraph, Solution 3 emerges as the more suitable solution. It is important to underline that this work, is preliminary work in order to figure out which leverage and gearbox arrangement can be used in order

to realize a first prototype, with minimum modifications on the suspension taken as reference. Two architectures of gearbox are been taken into account: Parallel axis and Planetary set. For both of this configurations, gears, shaft and bearings are been sized according to ISO standard.
The efficiency of the leverage system is not simple to calculate by means of analytical method. For this reason the efficiency has been considered as consequence of the transmission angle that has been limited in a range of 40° and 140°, according to literature regarding mechanism synthesis. On the other hand KISSsoft has been used in order to calculate efficiency in the gearbox. According to Standard ISO it is possible to calculate the losses both in gears meshing and bearings.

Finally the aim of this work is to make a comparison between an existing hydrostatic solution which has been already realized and tested at LIM. The electric motor characteristics considered here are the same that are performed by the electric motor used in hydrostatic solution. The objective is to figure out the possible advantages given by an electromechanical solution compared with hydrostatic solution for a given suspension scheme.

Chapter 2

Leverage Design

In general the Rotary RSA under investigation is composed by three main different elements as shown below, in which the arrows correspond to mechanical connections. The input signal is the speed v given from the wheel stroke.

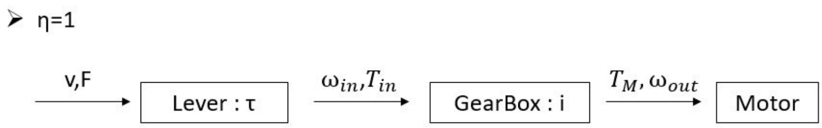


Figure 2.1: Logical Scheme

Considering the following data as input:

- v = 2000 mm/s *Wheel vertical Speed*
- F = 1000 N *Wheel vertical force*
- *Overall transmission ratio* $\frac{v}{\omega_{out}}$ k = 1.5 mm/rad

The overall transmission ratio is a consequence of the electric motor characteristic that it is considered as reference.

Considering for sake of simplicity the efficiency equal to 1, the relation $\tau \cdot i = k$ is verified. Because the overall transmission ratio is known, the first step is the transmission ratio calculation provided by the leverage τ and then the Transmission ratio i required from speed multiplier.

2.1 Basic considerations on 4-bar mechanism design

By *Mechanism Synthesis* [6] can be pointed out the design or creation of the linkage mechanism proportion to yield a desired set of motion characteristics. Contrary to *Kinematic* or *Dynamic Analysis*, in *Mechanism Synthesis* bar lengths are not given. The considerations in this section are referred only to planar mechanism, in particular to 4-bar linkage. Different type of 4-bar mechanism can be recognized with *Grashof' Law*.

Grashof's Law. *In a 4-bar linkage, if the sum of the shortest and the longest link-lengths must be less then or equal to the sum of the remaining two link-lengths, then the shortest can rotate fully rotate relatively a neighbouring link.*

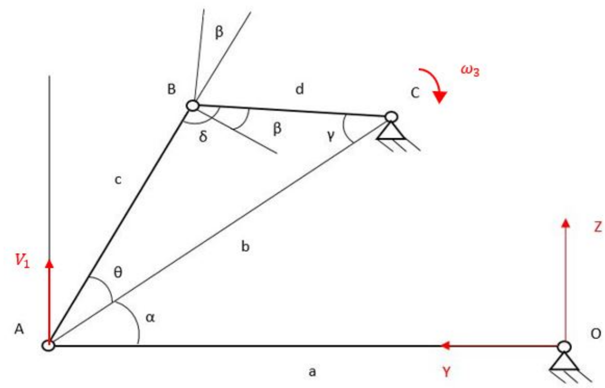


Figure 2.2: 4-bar linkage

In formula the *Grashof's Law* can be expressed as follow:

$$BC + AO \leq AB + CO \tag{2.1}$$

BC and AO links are respectively the shortest and the longest links, whereas AB and CO are the remaining links. If the first member of inequality 2.1 is equal to the second, the mechanism is a *Parallelogram Mechanism*. If inequality 2.1 is satisfied the mechanism, depending on the shortest link position is possible to understand if the linkage is: *Crank-Crank*, *Rocker-Rocker*, *Crank-Rocker*.
Very large number of analytical techniques are available for *Mechanism Synthesis*. In case of *Crank-Rocker* mechanism, if the angular displacement corresponding to complete *Crank* rotation is known, the synthesis can be carried out following an analytical approach [7] starting from the position equations of the coupler, point A and B, in its two limit positions. That is when the crank BC is collinear with coupler AB or when they are folded. Freudenstein equation [7] allows to design 4-bar proportion, and not absolute link dimensions because the solutions are infinite, considering 3 input and output angle as known. By Freudenstein equation equations system in 3 unknown can be solved in order to guarantee 3 precision output positions starting from 3 input positions. One approach is different from the other considering as known angular *Crank* and *Follower* displacements, or one link length and angular displacement of the *Follower*.

In any case two important parameters has to be considered:

- *Speed Ratio*
- *Transmission Angle*

The *Speed Ratio* can be defined as $\tau = \frac{v_{in}}{\omega_{out}}$, that can be considered as Mechanical Advantage. Referring to Figure 2.2 the input signal is the point A linear speed V_3, instead angular speed ω_1 about point C of BC link is output signal, so that the dimension of speed ratio is [mm/rad]. The numerator is the speed of the input signal due to suspension vibrations, the denominator is the angular speed of the output link about the output point. In linkage mechanism its value is not constant and it is a function of the input displacement.
The *Transmission Angle* is defined as the angle between the coupler and the follower. The quality of the transmission in term of velocity and forces [8] is usually represented by the *Transmission Angle*. It varies throughout the range of operation and is most favorable when it is equal to 90°. Transmission of motion is impossible when the transmission angle is equal to 0° or 180°. Transmission angle too small or too large causes errors of motion, high sensibility to friction in joints that is manufacturing errors and noise. Furthermore, transmission angle too much different from 90° causes non linearities as well as less smooth motion. As seen before if, for example, has to be designed a mechanism that for a complete crank rotation realizes a certain sweep angle of the follower, the solutions are infinite. The infinite solutions are proportional. Freudenstein in [9] perform the best links proportion in order to have an optimum *Transmission Angle* solving a cubic equation. The interesting thing is that in this way, only one of the roots of the cubic equation has physical meaning and then the solution is unique. Throughout the motion of the link, the difference of the transmission angle from 90° is minimize and the quality of the mechanism is maximized.
In practice the objective is to control and minimize the difference of *Transmission Angle* from 90° during the motion. Range between [40°-140°] for transmission angle is generally recommended.

The first objective of the leverage is to realize the higher transmission ratio, starting from input signal which is wheel vertical speed. As consequence the transmission ratio required from the gearbox is smaller, hence better compactness and in some case better efficiency.
The second objective of the leverage is to realize higher transmission ratio as possible with minimum modifications to the suspension taken as reference.
Last objective of the leverage arrangement is to respect suspension scheme geometry in stroke motion, the interference with suspension elements and wheel arch are are not allowed. The aim of this chapter is to describe four different leverage arrangements that are been investigated and compared taking into account the suspension scheme shown in figure 2.3 below:

As it is shown the suspension scheme that has been considered as reference is a high double wishbone suspension. It is important to highlights the fact that the kind of electromechanical shock absorber considered in this work can be realized only in this kind of suspension. In practice, because of rotary damper can't accomplish structural task, it is mandatory to have a scheme such as double wishbone suspension. Hence, this kind of shock absorber can't be used in scheme like *McPherson* strut because in this suspension the Hydraulic Damper is used also as structural

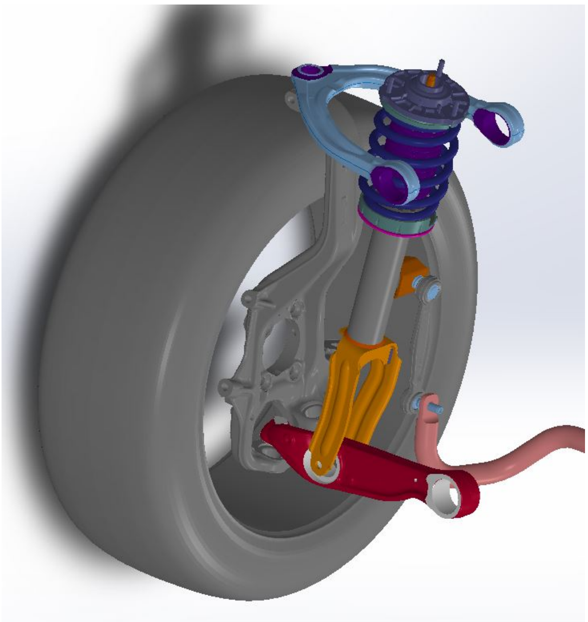


Figure 2.3: Reference suspension

element. In the case of leverage for *Rotary Damper* and considering the hypothesis listed before the main objective of leverage are:

- *Small Speed Ratio*
- *Acceptable Transmission Angle*
- *No interference with other suspension elements*

Low *Speed Ratio* means high value of angular displacement for a given input linear displacement of signal v from wheel.

The *Transmission Angle* has to be as near as possible to 90° degrees in order to minimize the force components download to bearing or to stretch or compress the links. The leverage throughout the motion from DS to US, has not to interfere with the other elements of the suspension, that is steering rod and transmission shaft. In the following sections are described different solutions that are been investigated for leverage arrangement, starting from simplier solutions that don't require any additional levers to more complex solutions in which additional levers are considered in suspension scheme.

2.2 Leverage - Solution 1

The first leverage arrangement that has been considered is the simpliest one. Using the kinematic of the lower arm of the suspension in possible to obtain a simple kinematics as indicated in figure 2.4 below.

For simplicity the suspension can be represented as follow in figure 2.5.

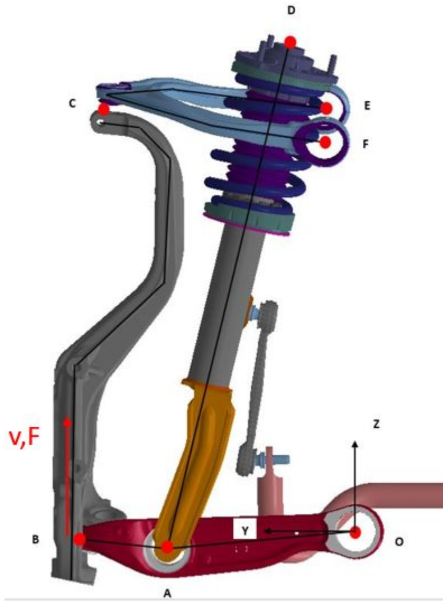


Figure 2.4: Leverage solution 1

Hydraulic tube and spring are not represented for simplicity but in this solution the suspension scheme is kept as it is; hydraulic tube accomplish only structural function for a correct working of the spring. Gearbox and electric motor are placed in point O. The stroke motion is converted by means of the rotation of the lower arm about point O.

Transmission ratio given by solution 1 can be calculated as following:

$$\tau = \frac{stroke}{lower\ arm} = 346.4mm/rad$$

As consequence the gearbox transmission ratio required is:

$$i = \frac{k}{\tau} = 0.0043 \qquad \frac{1}{i} = 231$$

Considering v=2000 mm/s and F=1000 N values it is possible to calculate torque and speed as input for the gearbox:

$$T_{GBin} = 346.4Nm \qquad n_{GBin} = 12700rpm$$

It is important to highlight the fact that it this leverage arrangement the transmission ratio is constant. However, the transmission ratio required from the gearbox is high.

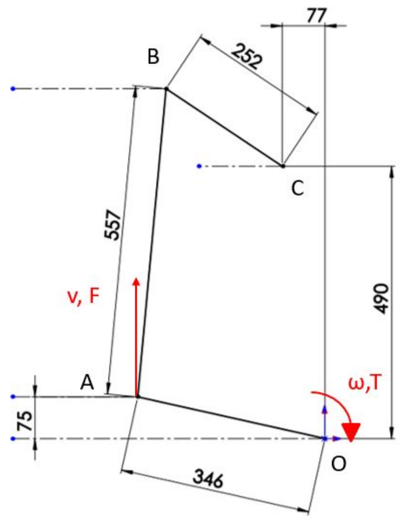


Figure 2.5: Leverage solution 1 scheme

2.3 Leverage - Solution 2

The solution 2 that has been investigated is represented in figure 2.6. It is similar to the previous one but the difference is in the gearbox and electric motor position. In this case the gearbox is placed in the hinge of the upper arm, point C. The kinematic leverage that can be obtained is a 4 bar linkage.

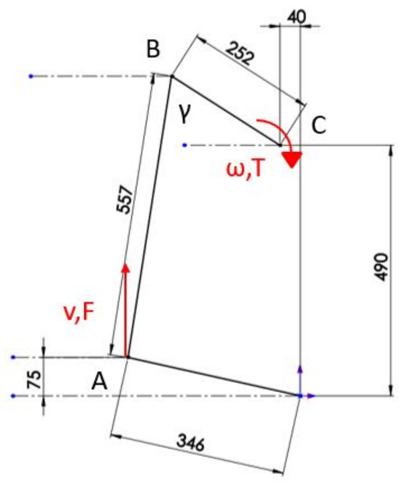


Figure 2.6: Leverage solution 2

Similarly to solution 1, the suspension is kept as it is with both shock absorber to accomplish only structural task, and the spring. The transmission ratio can be calculated with kinematic analysis. The transmission ratio is not constant in this case, hence the transmission ratio in three stroke positions is calculated. In the table 2.1 are indicated transmission ratio for three remarkable positions: Upstroke(US), Nominal(NOM), Downstroke(DS) and then a mean value of this function.

τ_{US} $\left[\frac{mm}{rad}\right]$	τ_{NOM} $\left[\frac{mm}{rad}\right]$	τ_{DS} $\left[\frac{mm}{rad}\right]$	τ_{mean} $\left[\frac{mm}{rad}\right]$
255.7	251.1	225.7	244.2

Table 2.1: Transmission ratio τ from US to DS for solution 2

The τ_{US} corresponds to the worst situation: for a given stroke displacement, less amount of degrees are obtained. In fact, the smaller is the transmission ratio τ the better. Hence, the gearbox transmission ratio i can be calculated considering this value.

$i = \frac{1.5}{\tau_{US}} = 0.0059 \quad \frac{1}{i} = 171$
$T_{GBin} = 171 \quad n_{GBin} = 74.7rpm$

2.4 Leverage - Solution 3

The Solution 3 that has been considered is sketched in figure 2.7. This leverage arrangement can be considered as an improvement of the solution 2. The kinematism is still a 4 bar linkage in which two levers are linked to the lower arm as indicated in 2.7.

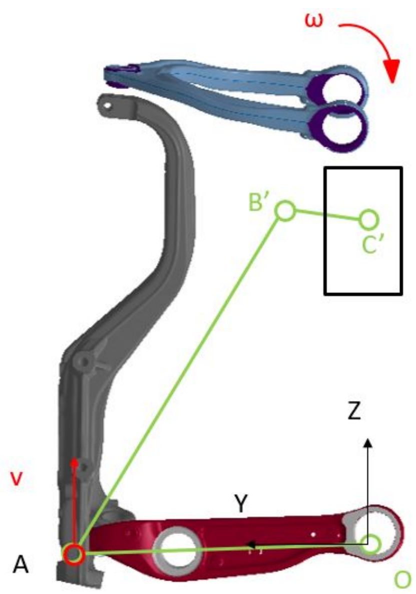


Figure 2.7: Leverage - Solution 3

A 4-bar linkage has been considered as indicated in green because this type of linkage is a good compromise between simplicity and efficiency. In *Rotary Damper* design, the 4-bar linkage has not to accomplish a particular position in time or a particular speed ratio function, as typically happen in mechanism synthesis. For this reason, contrary to what discussed in the previous section, has not been followed any analytical approach.
It has been implement a MatLab function. The aim of this function is to consider different 4-bar general configurations and cut all the configurations that don't verify certain constraints that is possible to tune depending on design requirements. Referring to Figure 2.2 the 4-bar linkage design process used, can be summarize as follow:

1. *Hypothesis*

   The function has been set up considering the following hypothesis:

   - *Vertical Stroke of point A is the known input signal*
   - *AO length is known*
   - *Down Stroke* (DS), *Nominal Condition* (NC) and *Up Stroke* (US) are three reference condition in which *Speed Ratio* has been performed.

   The NC condition is characterized by horizontal AO link. The DS and US condition correspond to minimum stroke and maximum stroke. The angle between AO link and Y axes both in DS and US is defined as ϵ angle.

2. *Variable definition*

   Considering the figure Figure 2.2 the first step has been the variable definition. The d length and γ angle are the two independent variable. For a given position of point C, different combinations of d, γ can give a fully defined configuration.

3. *Different link proportion investigation*

   Referring to Figure 2.8 below different configurations have been investigated in NC initially, each with a different combination of (d,δ,Y_C, Z_C,AO) varying the independent variables d, δ for each position of point C.

4. *Angle computation*

   Each angle can vary throughout the motion, then the following angle are been calculated in each situation DS, NC and US. In the following, the expression used to define the configurations are reported:

$$a = OA \quad b = AC \quad c = AB \quad d = BC$$

$$c = \sqrt{b^2 + d^2 - 2bd\cos\gamma}$$

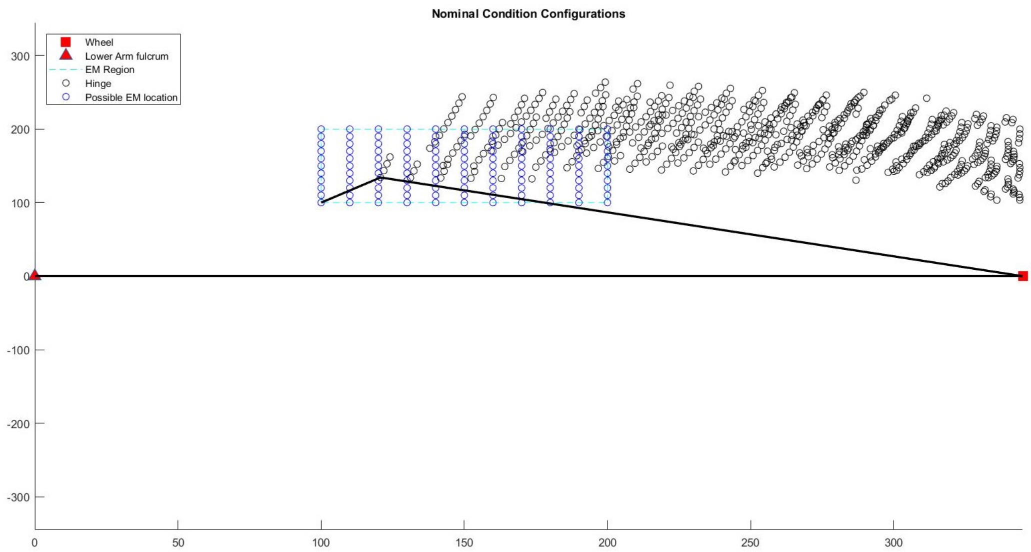


Figure 2.8: Configuration investigated

$$\theta = \arcsin\left(\frac{d}{c}\sin\gamma\right)$$

$$\delta = 180° - \theta - \gamma$$

$$\beta = \delta - 90°$$

$$\omega_3 = \frac{V_1(\sin(180° - \epsilon - \theta - \alpha))}{\epsilon(\sin(\theta + \gamma))}$$

$$\tau = \frac{V_1}{\omega_3}$$

Can be pointed out that, in practice, τ is independent on V_1, in fact it depend only on the linkage geometry that is link lengths and angles.

5. *Contraints imposition*

   In order to cut the configurations that are not acceptable the following criteria have been imposed:

   - *Maximum* Y_B, Z_B
   - *Maximum* Y_C, Z_C
   - *Acceptable Transmission Angle* δ *range*
   - *Acceptable* τ *variation from nominal condition*

The previous constraints are applied at NC, DS and US. The configurations that do not satisfy the constraints are kept out.

6. *Remaining Configuration*

   Afterwards constraints imposition, is possible to obtain the remaining configurations characteristics reported in table. Table 2.2 reports main geometric characteristics of the best solution investigated of this kind. In the first three columns there are τ values in US, NOM and DS respectively, [mm/rad] is the dimension.From the fourth to eighth column there are: τ_{mean}, d, γ, Y_C and Z_C respectively. It is important to underline that it has been imposed also a tolerance about speed ratio variation equal to 40%, in order to obtain smoother motion conversion from wheel stroke into rotational displacement.

τ_{US} $[\frac{mm}{rad}]$	τ_{NOM} $[\frac{mm}{rad}]$	τ_{DS} $[\frac{mm}{rad}]$	τ_{mean} $[\frac{mm}{rad}]$	d $[mm]$	γ $[deg]$	Y_C $[deg]$	Z_C $[mm]$
68.18	100.26	73.91	80.78	96	80	-100	490

Table 2.2: Example of 3 configuration characteristics remained

As seen before the function allows to investigate vary large number of a general 4-bar linkage, perform geometric characteristics hence *Speed Ratio* and then exclude the configurations that don't verify the constraint imposed. Below are indicated the values of gearbox requirements, Angular speed and Torque that correspond to the geometric characteristic indicated in table 2.2, considering the worst case that happen in nominal condition.

$$i = \frac{1.5}{\tau_{NOM}} = 0.015 \qquad \frac{1}{i} = 66$$

$$T_{GBin} = 100.26 \qquad n_{GBin} = 190.5$$

Interestingly, it has been possible to figure out the behaviour of τ varying the position of point C. The results point out that the larger of $Y_{C'}$ the better for the transmission ratio of the leverage system.

2.5 Leverage - Solution 4

The leverage solution 4 that has been investigated is sketched in figure 2.9. Gearbox and electric motor are placed in the lower arm hinge, in point O.

Two additional levers indicated in green: PH and HO are linked to hydraulic tube. This kind of leverage has been investigated in order to keep as it is the reference suspension scheme. In this way hydraulic shock absorber accomplish only structural task and the spring can works properly.

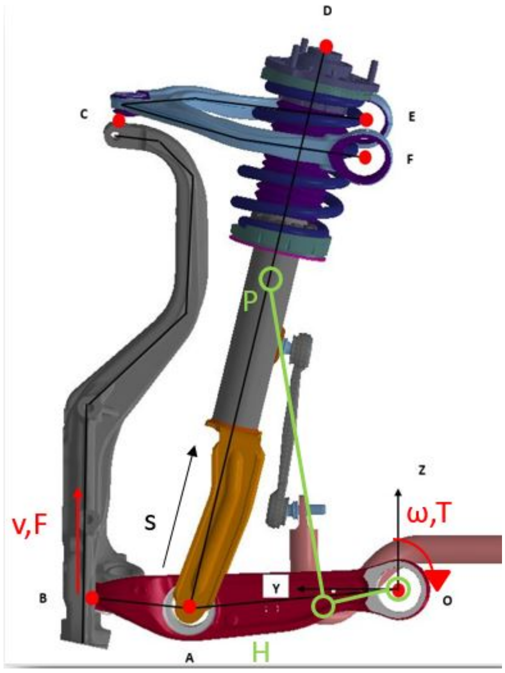


Figure 2.9: Configuration investigated

The procedure that has been followed in order to find the best leverage proportion is summarized as follows:

Hypothesis:

- Hydraulic damper accomplish only structural function for spring element
- Levers PH and HO are added to the suspension scheme
- Gearbox and electric motor are placed at point O

Procedure:

- Find the best leverage proportion that realize minimum transmission ratio[mm/rad]
- Different combinations levers PH and HO are investigated for different s_P coordinates of point P on hydraulic shock absorber tube
- Configurations that don't satisfy the constraints are eliminated

Each combination of levers PH and HO has been evaluated for different coordinate s_P on hydraulic tube. The constraints that are been imposed deal with the *Transmission Angle* range allowed, which is PHO angle. Similarly to previous solutions this range has been imposed from 40° to 140°. Following the procedure explained before, different leverage proportions are been investigated. The table 2.3 reports the transmission ratios for the best leverage proportion.

τ_{US} $\left[\frac{mm}{rad}\right]$	τ_{NOM} $\left[\frac{mm}{rad}\right]$	τ_{DS} $\left[\frac{mm}{rad}\right]$	τ_{mean} $\left[\frac{mm}{rad}\right]$
87.6	114.6	105.4	102.5

Table 2.3: Transmission ratio τ from US to DS for solution 4

The results refers to a HO=72 mm, PH=383 mm and s_P coordinate equal to 360 mm.

Is possible to calculate the requirements for gearbox considering the worst τ value, which in this configuration occurs in nominal condition.

$i = \frac{1.5}{\tau_{NOM}} = 0.0131 \qquad \frac{1}{i} = 76.4$

$T_{GBin} = 114.6Nm \qquad n_{GBin} = 190.5$

2.6 Comparison between Leverage solutions

As highlighted in the beginning of this chapter the objective of this comparative analysis regarding the leverage solutions is to figure out the best value of the *Transmission ratio* which is possible to perform. Each leverage arrangement have some advantages and disadvantages that can be summarized in the following table:

SOLUTION 1	SOLUTION 2	SOLUTION 3	SOLUTION 4
PROS	PROS	PROS	PROS
➢ No modifications to the suspension are needed ➢ Lower position of Gearbox and Electric Motor respect to center of mass ➢ Constant value of transmission ratio τ	➢ No modifications to the suspension are needed ➢ Better transmission ratio τ than solution 1	➢ Better transmission ratio τ of leverage mechanism ➢ Smaller GearBox transmission ratio ***i*** is required (could be advantageous to increase gearbox efficiency and reduce the size)	➢ The suspension scheme modifications is minimum ➢ Lower position of GearBox and electric Motor
CONS	CONS	CONS	CONS
➢ Worst transmission ratio τ ➢ Larger transmission ratio of the GearBox is required (hence, reduced efficiency and increased size)	➢ Gearbox transmission ratio i required is still high (hence, reduced efficiency and increased size)	➢ Hydraulic damper and spring element have been removed by leverage: requires a solution for fixing the spring element	➢ Worse transmission ratio τ ➢ Damping force direction could influence negatively a correct working of the spring

Figure 2.10: Pros and Cons for Leverage solutions

Solution 1 and Solution 2 are very simple because they take advantage from the kinematics given by the suspension scheme itself. However, both these two solutions perform very small *Transmission Ratio*, as consequence the gearbox specification require very high speed ratio. From the literature is possible to figure out that speed ratio larger than 100 carry out smaller efficiency. Solution 3 and Solution 4 include the possibility of install additional levers in order to minimize the transmission ratio τ. The former realizes the best speed ratio but in this leverage arrangement big modifications are needed, in fact the 4 bar linkage would substitute hydraulic tube and then the spring. For this reason remains the problem of fixing the stiffness element. The latter has a worse speed ratio compared to Solution 3 but it is simpler and no big modification are needed apart from the hinge on hydraulic tube. It is important that the direction of the damping force on the hydraulic tube passes through the dome on upper plate. In this way bending moment on hydraulic tube are reduced.

It is important to underline that this analysis has been carried out in order to find a solution with minimum modifications required on the suspension taken as reference and clearly a gearbox speed ratio required reasonable, in order to keep efficiency as large as possible. For these reasons Solution 1 and Solution 2 have been discarded, mainly for the very high speed ratio required from gear train. On other hand, solution 4 has been discarded for complexity and large modifications required on suspension scheme. Finally, Solution 4 is the solution that requires a reasonable transmission ratio from the gearbox and at the same time very small modifications on suspension scheme. This Solution 4, and then the values of Speed and Torque calculated as input in the gearbox will be considered in the following paragraph for what concerned trade of analysis about different configurations of gearbox.

Chapter 3

Gearbox

3.1 Introduction

In this chapter are described different architecture that are been evaluated for a preliminary Gearbox design. Different parameters are been considered for the comparison between the solutions:

- *Overall diameter of Gearbox*
- *Total Weight*
- *Compatibility with the existing Electric motor*

The last point refers to the following figure 3.1, in which is illustrated the case of the existing electric motor that has been considered as reference for the *Rotary Damper Design.*

Main dimensions are given by:

- *Overall diameter of electric motor case = 88 mm*
- *Overall length of electric motor case = 90 mm*

The aim of this part of the project is to design a gearbox as compact as possible according to case dimensions. It is clear that could be unlikely to have the same diameter of the case illustrated, anyway the diameter of 88 mm has been considered as objective. In practice, it would be possible to have also a flange with larger diameter than 88 mm to fit the gearbox at the end of the electric machine. As consequence of the diameter has been evaluated also the overall weight of the gearbox, including shafts, bearings and gears.

Important aspect of this part regarding gearbox design, are the assumption assumed for gear pair and shaft sizing. In particular the data available for what concerned the input speed and torque are respectively equal to 2000 m/s and 2000 N. This value of force is the limit value that can be reachable rarely in the life of

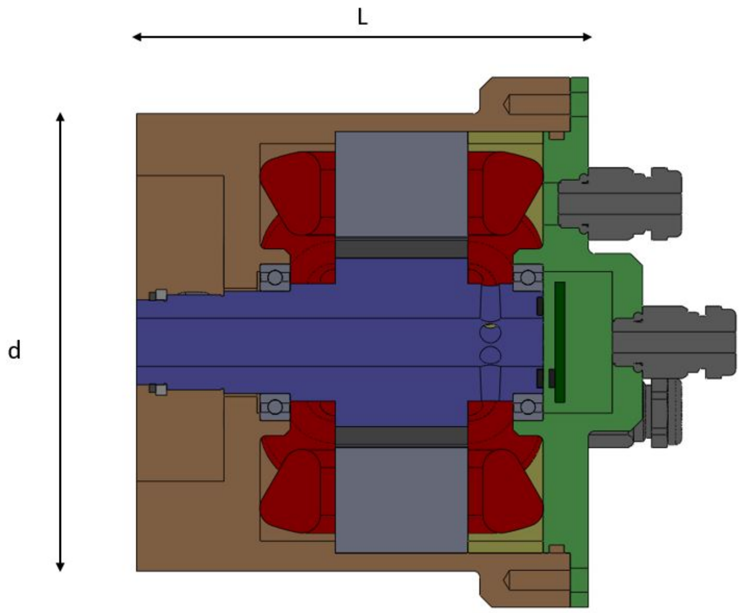


Figure 3.1: Electric Motor section

auto vehicle suspension. Most of the shock absorber working condition is limited in the upward linear zone in which damping force is below the limit value equal to 2000 N. Hence, it has been chosen to use value of force equal to 1000 N at the wheel in order to set at this load the gearbox sizing. Definitely the input data that are been assumed for gearbox sizing are:

- *F=115 Nm*
- *n=113 rpm*

This data are the results of the transmission ratio τ performed by Solution 3 leverage arrangement illustrated in previous paragraph. It is clear that, load application in this gearbox will not be stable at 1000 N at the wheel hub, but overload and alternating conditions will be surely verified. In order to take into account this type of load application mode, correction factors according to ISO 6336 method B are introduced as will be discussed in following section. An evaluation of the force at wheel hub has been calculated considering an existing Simulink suspension model. From this model is possible to calculate damping force in time in the case of a vehicle running on an ISO standard profile at 100 Km/h. The results point out that for most of the time life the force reach 1000N. In the following graph is illustrated the load spectrum for a simulation of 120 s. At this first step of the design this load spectrum gives a first indication on the damping force at which the damper works on this type of road profile. It is clear that the damper has to resist even worse load conditions, that will be taken into account with proper safety factors. Furthermore, the following load spectrum gives an indication in term of torque and angular speed at which the gearbox will works. More accurate load condition can be described considering also the contributes of cycles at lower torque and speed in order to evaluate fatigue strength and damage condition. This aspect will not be considered here.

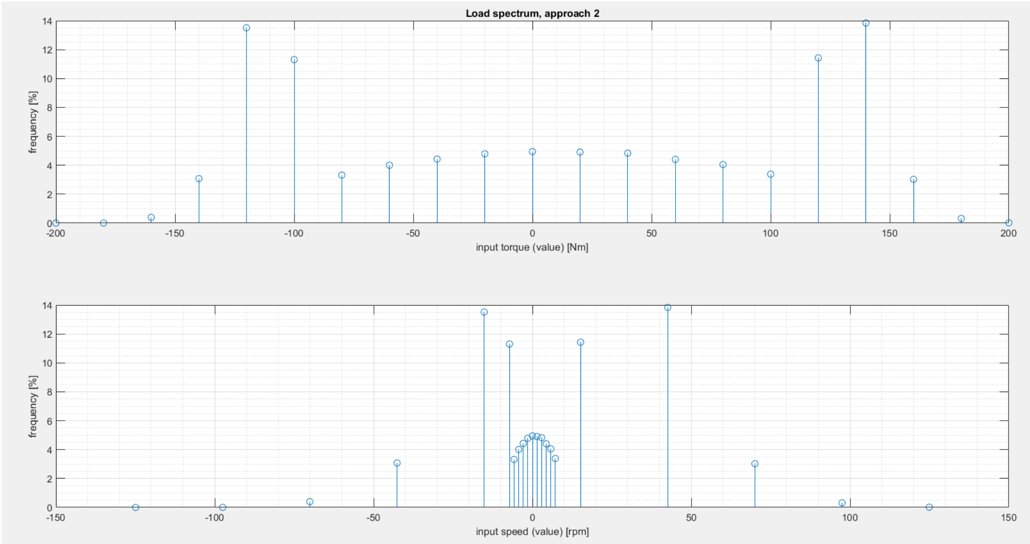


Figure 3.2: Load spectrum

On the Y axis is possible to read the frequency at which is reached a certain input Torque at the gearbox. Torque equal to 113Nm corresponds to a Force at wheel hub of 1000 N.
Preliminary evaluations regarded the efficiency calculations from analytical point of view. According to literature the efficiency for a gear pair meshing is close to 0.98-0.99. In first approximation the overall efficiency of a 3 stage parallel axes gearbox has 0.94 as total efficiency. However, the overall dimensions will be considerable. For this reason, power split has been introduced. Then planetary set has been considered considering a typical wind turbine gearbox, which is represented in following fig. 3.3. This configuration has 3 stages: the first is planetary, the input speed and torque enter in the carrier shaft. Then second and third stages are simple parallel axes gear pair.

The efficiency of the planetary stage can be calculated as *forward mode*, considering Carrier Shaft as input and Sun Shaft as output.

$$\eta = \frac{\eta_0(\tau_0 - 1)}{\tau_0 - \eta_0}$$

On the other hand the efficiency in *backward mode*, where the sun Shaft is the input and Carrier Shaft is the output can be calculated as follows:

$$\eta = \frac{1 - \eta_0\tau_0}{1 - \tau_0}$$

In which:

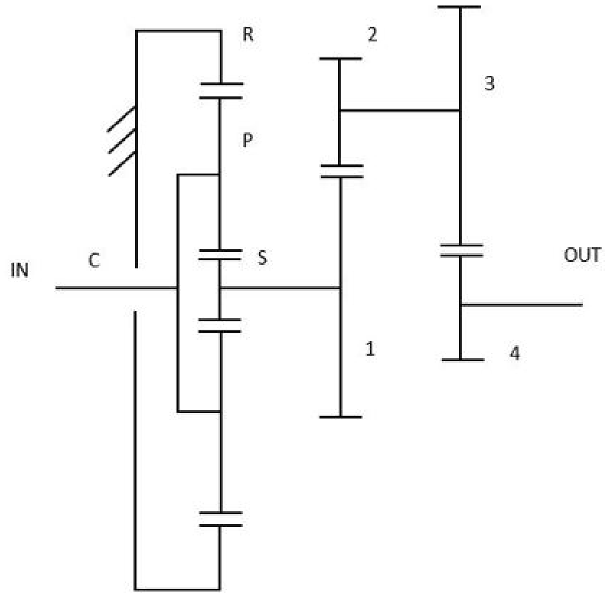


Figure 3.3: Wind turbine scheme

- τ_0=equivalent transmission ratio
- η_0=efficiency for a gear pair

In this case the efficiency is close to 0.9292. These relations can give preliminary indications on the efficiency values of the overall gearbox. In order to have reduction of overall dimension and higher efficiency have been considered two configurations that will by illustrated in following sections: *Fixed axes configuration* made up by 3 stages and a power split. *Planetary configuration* made up by 2 equal stages, to reduce gear pairs and then efficiency.

3.2 Parallel Axes Architecture

As stated previously the transmission ratio required by gearbox is the results of the transmission ratio τ given by the Solution 3 of the leverage system and the characteristic of electric motor which can provide $T_{GB} = 1.1Nm$ @ n=10000 rpm. Hence, the input data for gearbox sizing can be listed as follows:

- *F=115 Nm*
- *n=113 rpm*
- i=88.

The transmission ratio i=88 has been considered as conservative value. The transmission ratio required considering leverage Solution is 76. Here is considered

i=88 so as to consider errors in leverage mounting that can cause a reduction of τ. In this section *Parallel Axes architecture* is investigated. It has been chosen to use 3 stages. According to catalogue from main gearbox producer, in order to keep high values of efficiency is better to use not more than 3 stages, when possible. This solution is made up of 3 stages:

- *Stage 1: Parallel Axes with power split*
- *Stage 2: Parallel Axes with power branches merge*
- *Stage 3: Ordinary parallel axis stage*

The scheme below illustrates the power flow throughout the stages. Equal amount of power goes in the paths of the first two stages.

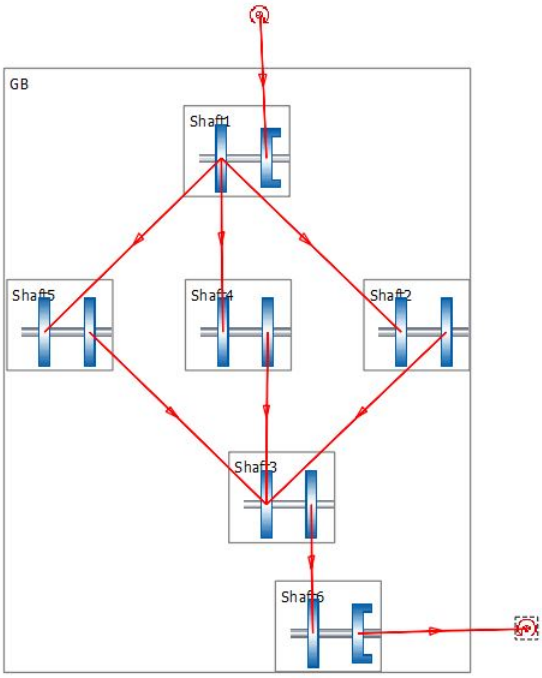


Figure 3.4: Parallel axes power flow scheme

Power split configuration has been chosen in order to reduce the overall dimension, in fact simple parallel axes gearbox would required too high diameter, incompatible with this application.

3.2.1 Gear Sizing

The gears has been sized according to standard ISO 6336 method B. In Kisssoft is possible to use two routines called *Rough Sizing* and *Fine Sizing*. The first allows to set the transmission ratio for a gear pair and a range for pinion teeth number. This function gives a first approximate information about the module and then dimensions of the gear pair needed to transmit total amount of power that in this case is, in reference condition:

$P_{in} = 1.36KW$

The second routine *Fine Sizing* allows to calculate different gear pair characteristics setting a range for module and interaxis. This function is useful to compare these different configurations at the same time. Finally, tuning face width b it has been possible to find a compromise in order to reach minimum Safety factor for bending and Safety factor for flank strength in fatigue condition.

$SF_{min} = 1.4 \quad SH_{min} = 1.0$

It is important to underline that the fine sizing has been carried out controlling efficiency of the gear pair and the overall dimension. The closer the overall diameter is to electric motor diameter, the better.

One parameter that affect overall dimension of the gearbox is the transmission ratio split. For the sake of simplicity, from a practice point of view, the τ_1 and τ_2 are equal. The transmission ratio of the τ_3 is calculated as consequence starting from overall transmission ratio τ_{tot}.

The material that has been chosen for gears is a 18CrNiMo, which is a common material used in power transmission, by means of gears:

$\sigma_s = 850Mpa \quad \sigma_{max} = 1200Mpa$

In the following are described in detail the gear pairs geometric characteristic of the three stages. In order to obtain $\tau_{tot} = 88$, the best compromise has been reached splitting the transmission ratio throughout the stages in this way: $\tau_1 = 4$, $\tau_2 = 4$ and $\tau_3 = 5.5$. Spur gears has been considered for gearbox design.

- *1 Stage*

  Transmission ratio has been chosen to be smaller than $\tau_{stage} = 6$; according to literature, meshing efficiency will be around 97-99 % as long as the transmission ratio is smaller than $\tau_{stage} = 6$. First stage performs transmission ratio equal to $\tau_1 = 4$. In this stage the power is equally splitted in three different paths through 3 fixed planets every 120°, in order to reduce module and then diameters for each gear pairs. In the following table3.6 are reported Gear pairs geometric characteristics and load in the first stage.

  The gear sizing has been carried out according to ISO 6336 method B. Life fatigue factor Y_M has been imposed equal to 0.7, because the load on teeth is alternated. Load application factor has been imposed equal to 1.5 in order

	Power [W]	Torque Torque[Nm]	z [-]	m [mm]	b [mm]	alfa [deg]	K_a [-]	Y_M [-]
Gear 1	0.4537	38.33	69	1	20	20	1.5	0.7
Gear 2	0.4537	9.44	17	1	20	20	1.5	0.7

Table 3.1: Gear pair data stage 1

to take into account the overload that is possible to have considering random load given by road roughness.
The stress at root teeth and flank are listed in the following table 3.2, where the Safety factors are been calculated considering 18CrNiMo as material for both pinion and gear:

	σ_{root} [MPa]	σ_{flank} [MPa]	SF [-]	SH [-]
Gear 1	342.6	1235.17	2.21	1.05
Gear 2	313.96	1238.16	1.65	1.01

Table 3.2: Gear pair safety factors stage 1

It is important to underline that the correction factors K_v, K_{Ha} and K_{Hb} are given by standard. The first take into account sliding on tooth profile given by angular speed. The second and third factor misalignment in transversal and longitudinal direction.

- *2 Stage*

Transmission ratio at τ_2 is constrained by the inter axis between fixed planets axis and the axis of the driven pinion on shaft 3. This stage performs $\tau_2 = 4$. Because at this stage the torque is reduce through the 3 power paths, τ_2 could be higher than τ_1. However, inter axis constraint would increase excessively the overall diameter of stage 2. In the following table are reported Gear pairs characteristics in the second stage.

	Power [W]	Torque Torque[Nm]	z [-]	m [mm]	b [mm]	alfa [deg]
Gear 3	0.4537	9.44	90	0.75	12	20
Gear 4	0.4537	2.31	22	0.75	12	20

Table 3.3: Gear pair data stage 2

Second stage is the most critical, in fact the overall diameter is given by this stage. On one hand, the driven gear in this stage is the pinion, for this reason planets interference has to be take into account. On the other hand, transmission ratio limitation is given by inter axis constraint that has to be equal to the inter axis in the first stage.

	σ_{root} [MPa]	σ_{flank} [MPa]	SF [-]	SH [-]
Gear 3	280.17	938.92	2.66	1.35
Gear 4	259.72	982.88	1.94	1.24

Table 3.4: Gear pair safety factors stage 2

These results concerned all the gear pairs that include fixed planets of the stage 2. To avoid repetitions the result are reported once.

- *3 Stage*

The third stage is a simple parallel axis stage. The torque is reduced, hence the module required and the overall diameter can be reduced. On the other hand, the angular speed at stage 3 is very high. This aspect has been taken under control with safety factor at scuffing and sizing the gears in order to have reasonable K_v. In the following table are reported Gear pairs characteristics in the second stage.

	Power [W]	Torque Torque[Nm]	z [-]	m [mm]	b [mm]	alfa [deg]
Gear 5	1.36	6.9	85	0.75	16	20
Gear 6	1.36	1.3	16	0.75	16	20

Table 3.5: Gear pair data stage 2

	σ_{root} [MPa]	σ_{flank} [MPa]	SF_{min} [-]	SH_{min} [-]
Gear 5	235	3.05	1016	1.22
Gear 6	216	2.26	1023	1.16

Table 3.6: Gear pair safety factors stage 2

Stage 3 is the most critical for what concern scuffing and correction factor K_v. This is due to the very high speed of the shaft which is close to 10000 rpm. Speed factor $K_v = 1.24$. Misalignment factors are $K_{Ha} = 1.21$ and $K_{Hb} = 1.9$.

3.2.2 Shaft and Bearing Sizing

In this subsection Shaft and Bearing sizing are described. Power split it is also an important advantage for bearings. In this way, because of the circular symmetry of the fixed planets position it is possible to balance reaction forces on the bearings, with a improvement on bearings life.
Starting from input data, power split ratio and gears geometric characteristics it is possible to calculate reaction forces on the bearing and then bending moment on shafts.

- Shaft 1

  The following figure represent Shaft 1. The material chosen for shaft is C45, Yield stress of this materials:

  $\sigma_s = 490MPa \quad \sigma_{max} = 700MPa$

  Bearings are positioned both to the left of the gear 1 in order to reduce axial length dimension of the gearbox, with no particular disadvantage on shaft deflection.

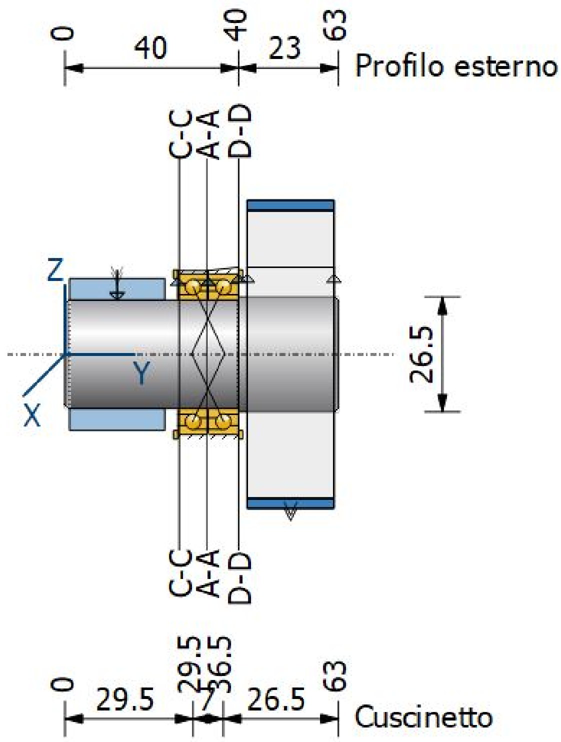


Figure 3.5: Shaft1

On the left are presents three loads:

- $T_{in} = 115Nm$
- $F_{rad} = 30N$
- $F_{axial} = 30N$

T_{in} is the input torque given by leverage system Solution 3. F_{rad} and $F_{axial} = 30N$ are the radial and axial components, as result of the force direction on the follower of the leverage system. These force are not constant during complete stroke of the wheel. However, the values considered are a first approximation in order to consider the consequence on shaft and bearing loads.

A consequence, bending moment and torque on shaft 1 are illustrated in following figure.

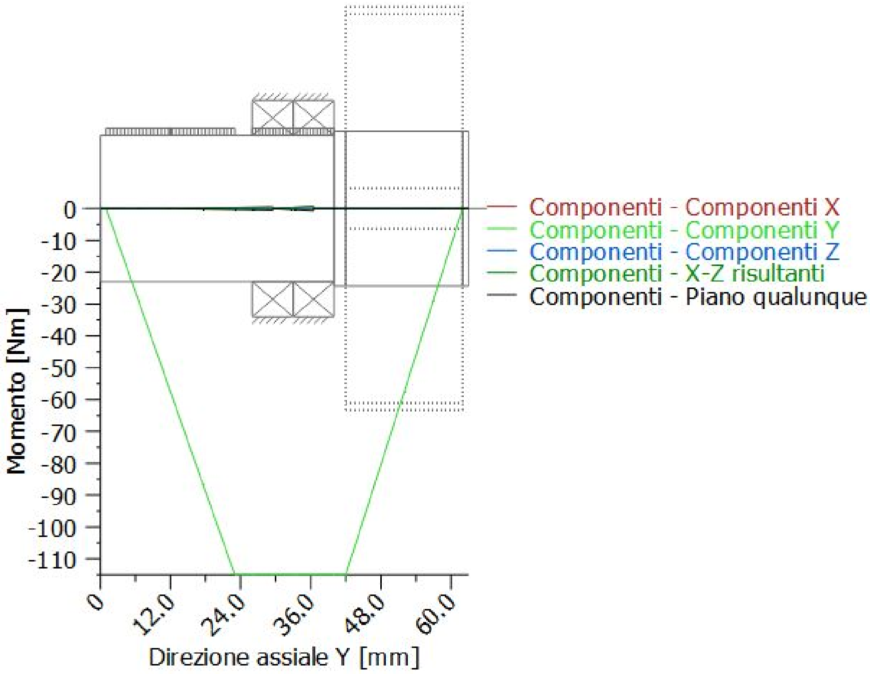


Figure 3.6: Bending moment shaft 1

Linear trend range is the result of the distributed load on the input coupling and face width of the gears at first stage.
It is clear that the symmetrical disposition of the gear pair of the first stage affects positively the reaction forces on bearings, that are balanced. For this reason bending, is given by radial and axial forces applied on the left of the two bearings.

Strength calculation are based on DIN standards, then notch effect factors are calculated as consequence of the geometry of the shaft.

Notch effect in bearings zone is given by interference of inner ring on the shaft. The bearings sizing has been carried out according to ISO 281. It has been

	β	SF
A-A	1.65	5.0
C-C	1.0	6.86
D-D	2.12	5.7

Table 3.7: Shaft1 safety factors

possible to calculate the life of the bearings; at this stage the minimum life has been imposed to be equal to 20000 hours. The bearings on shaft 1 have to support radial and axial loads. For this reason they are arranged in X manner. In the following table are reported the bearing data and service life.

	Code	R_x	R_y	R_z	Service Life
Bearing 1	SKF 71805 ACD/P4	-0.364 N	28.092 N	39.462 N	>1000000 h
Bearing2	SKF 71805 ACD/P4	0.243 N	-28.282 N	-49.129 N	>1000000 h

Table 3.8: Bearings on Shaft 1

Considering the symmetry of the gear pairs arrangement, the load on this shaft is in practice equal to zero. For this reason the bending deformation can be neglected referring to Shaft1.

- Shaft 2

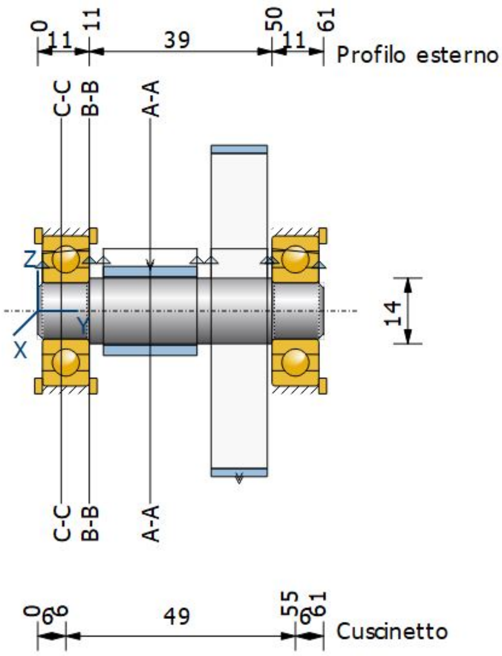
Shaft 2 is the shaft on which are keyed the fixed planets of the stage 1 and stage 2. In this case the forces are not balanced, for this reason this shaft is the most critical shaft. Shaft 2 is illustrated as follows.

As stated before loads on this shaft are not balanced for the presence of three fixed planets. Starting from the input data and gear pair sizing it is possible to calculate the reaction forces on the bearings and then bending moment which is reported below:

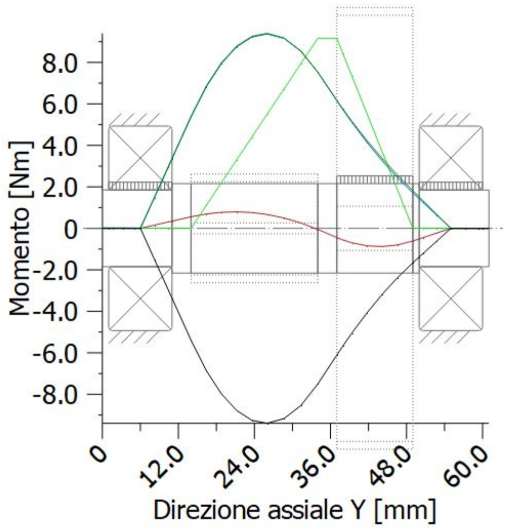
	Code	R_x	R_y	R_z	Service Life
Bearing 1	SKF 6201	-672 N	0 N	64 N	45000 h
Bearing 2	SKF 6201	-283 N	0 N	-101 N	511800 h

Table 3.9: Bearings on Shaft 1

Light green line is Torque, that grows linearly through the face width of gears, instead the other curves represent bending moment on different planes. These curves have a parabolic trend where the gears are placed because the load is



(a) *Shaft2*



(b) *Bending Moment Shaft2*

distributed along face width.

The pinion on the left is the driven gear of stage 1. The gear on the right is the driver gear of stage 2. It ha been evaluated also a configuration with one bearing between gears but he configuration with the bearings on the end of the shaft is an advantage for shaft deflection. Both gears are keyed on the shaft with interference. In the previous figure three critical section are highlighted. Strength calculation for what concerned these 3 sections are listed below in 3.9

Notch effect is given in the remarkable section by interference between bearing and shaft mate and by shoulder on the shaft for bearing positioning.

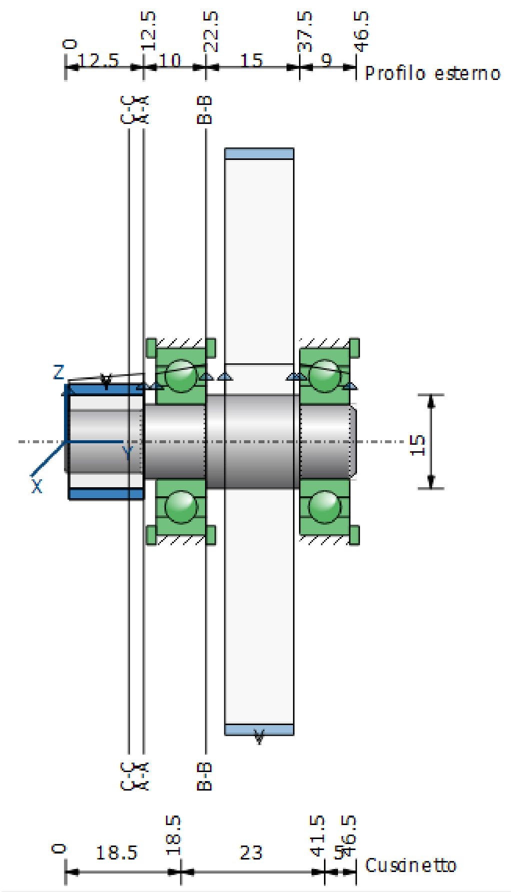
	β	SF
A-A	1.65	7.81
B-B	1.68	9.46

Table 3.10: Shaft2 safety factors

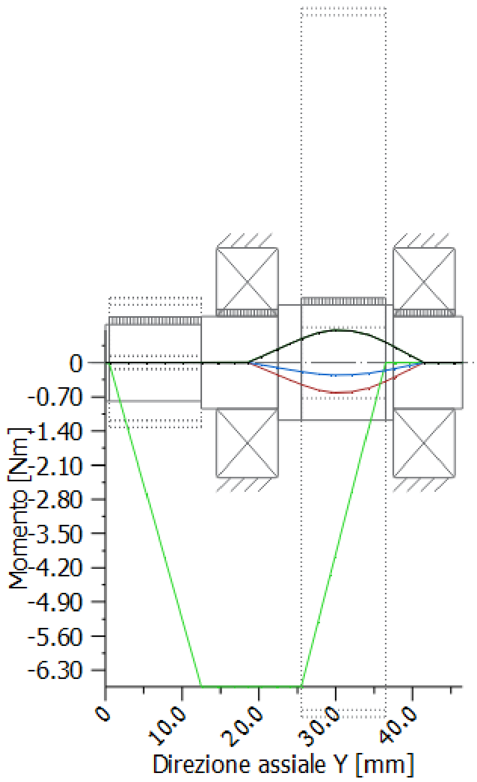
The same calculation concerns the others shafts that support the other 2 fixed planets, for this reason they will not shown here.

- Shaft 3

The shaft 3 is illustrated in following figure. This shaft supports partially the driven pinion of the stage 2 and the driver gear of the stage 3. Similarly to what has been stated for shaft 1, on the pinion on the the left are mated three planets every 120°. This is an advantage for the bearing reaction forces that can be balanced. On the right, gears pair meshing of the stage 3 represent the load that stress Shaft 2. The power is merged in this shaft; starting from gears pair characteristics is possible to calculate load and moment on shaft 3.



(a) *Shaft3*



(b) *Bending Moment Shaft3*

Different curves refers to different plane at which the bending is calculated. The black curve is referred to a general plane. The green line is referred to torque transmitted to through gears meshing, the value increase linearly in face width engagement of gear pair. The value is constant between pinion and gears, which is the output, at 6.3 Nm. Notch effect is given by interference mating between bearings and shaft and shoulder for bearing positioning. In the following table are reported Safety Factors and notch effect factors for the section highlighted in 3.11.

Similarly to shaft 1, the calculation for bearings life calculation has been carried out according to ISO 281. The arrangement of the bearings at the end of

	β	SF
A-A	1.42	7.46
B-B	2.13	11.0

Table 3.11: Safety factors for Shaft3

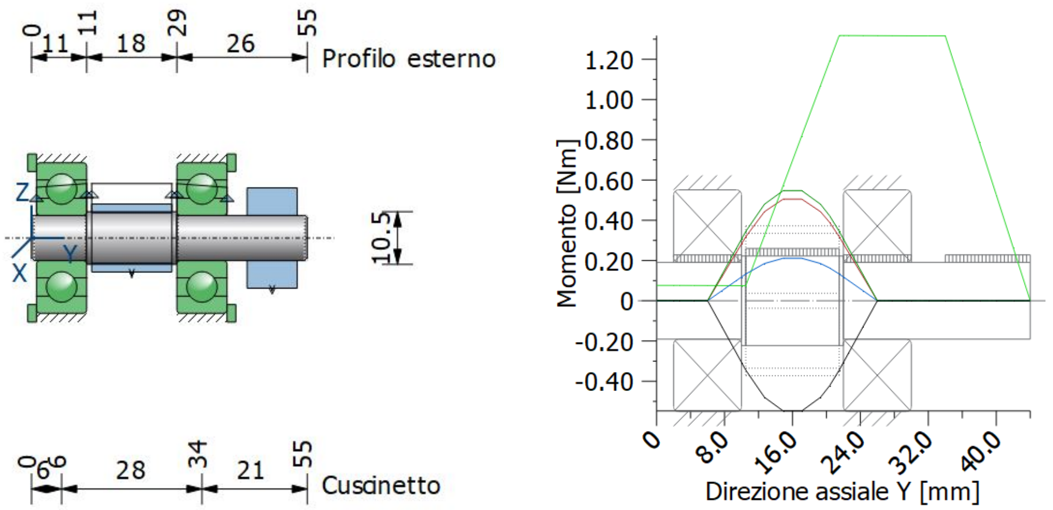
the shaft 2 has been considered for the smaller maximum inflexion.

	Code	R_{xz}[N]	Service Life[h]
Bearing 1	SKF 16101	84	>1000000
Bearing 2	SKF 16101	71	>1000000

Table 3.12: Bearings on shaft 3

- Shaft6

  Shaft 6 is the output Shaft on which is keyed only one pinion, output gear of the stage 3. The shaft 6 geometry is illustrated below. The light blue part on the right represents the output coupling. Starting from forces calculated at stage 2 on gear pairs, it is possible to calculate the reaction forces and bending moment on Shaft 6.



(a) *Shaft3* (b) *Bending Moment Shaft3*

The black line is referred to a general plane, considering bending. The green line refers to torque, which is constant between input and output pinion. Considering the very low value of torque at this stage the sizing of the shaft is mainly governed by bearing requirement. Safety factors through each section are higher than 20. Considering the high speed at this stage, it has been necessary to consider larger dimension of bearings, in particular of the outer ring. In fact the inner diameter is constrained by pinion diameter. Thermal problems can cause scuffing or bad working condition at these speed values. In the following table is are illustrated code bearings, load and service life guaranteed. It has been evaluated also roller bearing instead of ball bearing but it has been decided ball bearing in order to reduced dimension in Y direction of the global reference frame. Furthermore, roller bearings in this case would be a over sizing beacuse the load at this stage is very low.

	Code	R_{xz}[N]	Service Life[h]
Bearing 1,2	SKF W 629	76	230000

Table 3.13: Bearings on shaft 6

The gearbox with fixed axes configuration is illustrated in following figure 3.7 after gear pairs, shafts and bearings sizing. Considering the working condition as multiplier, power input enters at left side, then it is splitted and enters in second stage, with larger gears as fixed satellites and then the power is merged in a pinion. Finally the third stage with single parallel axis meshing with power output towards gearbox side on the right.

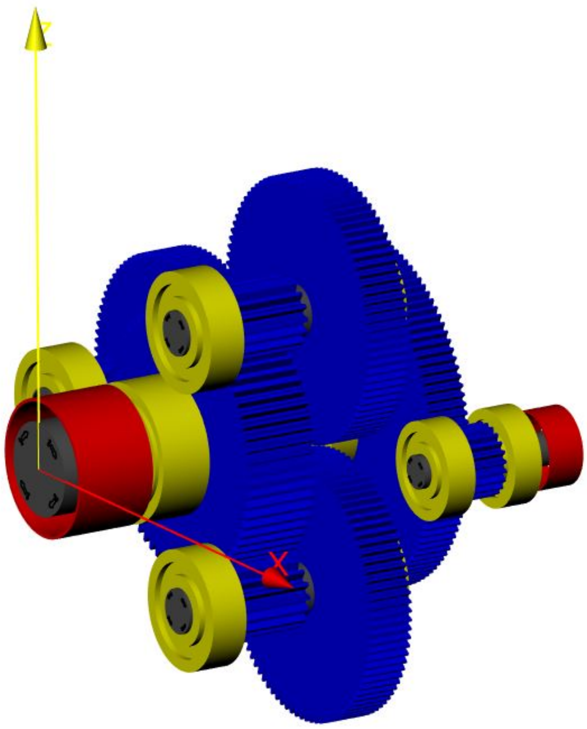


Figure 3.7: Fixed axes gearbox configuration

3.2.3 Efficiency

Efficiency calculation is available in KISSsoft according to ISO standard 14179-1. Power losses are due to following elements:

- *Bearing*(PVL)
- *Gear meshing*(PVZ)
- *Lubrification*(PVZ0)
- *Seals*(PVD)

Hence, total power losses can be calculated as the sum of these contributes. In this work will be consider only the first three sources, seals losses will not be considered. Power losses in bearing depends on the radial load, geometry and type of bearing. Power losses in meshing depends on teeth profile, gear material and lubrification type. Generally, main power losses are attribute to gear meshing, instead for very high speed churning losses become considerable.
As seen in previous sections the gearbox sizing has been carried out considering a fixed value of input Torque and Angular Speed. Clearly, the input data depend on the roughness of the street, for this reason the evaluation of the input data have to be evaluated from a probabilistic point of view, aspect that deals with random vibration. However, in this work efficiency map is calculated for different combinations of Torque and Angular speed. These combinations have been interpolated in order to obtain an efficiency map, for a range of input Torque and Speed. In the following figure is illustrated the efficiency map of the fixed axes configuration.

The colorbar illustrates the range of efficiency which is reached in the Torque and Speed range considered. The maximum efficiency reached is equal to 0.943. The efficiency decreases at low values of torque. Furthermore, it can be noticed that the are at higher efficiency is reached for higher value than 40Nm and higher value than 40rpm. For lower values the efficiency starts to decrease. However, even in this case the efficiency is close to 0.92.

3.2.4 Sound Level Pressure

Noise in gearbox is generally an important parameter to take under control. This aspect depends on many parameters: teeth profile, lubrification and materials. An accurate noise prediction of a gearbox requires FEM methods simulations that could be time consuming. In this work will be considered Masuda [10] formula so as to obtain SLP generated by gears meshing. The approach proposed by Masuda is a simplified formulation of noise prediction, that is reported as follows:

$$L = \frac{20(1-\tan(\beta/2))}{\sqrt[8]{\epsilon}}\sqrt{\frac{5.56+\sqrt{v}}{v}} + 20\log(W) + 20\log(X) + 20$$

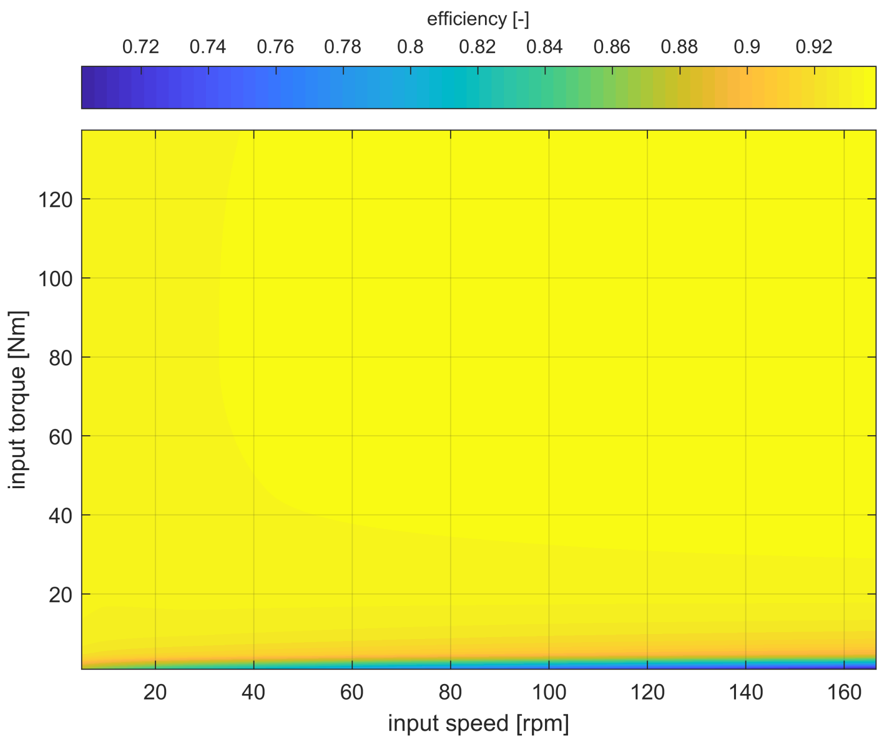


Figure 3.8: Efficiency map of fixed axes configuration

In which:

- β= helical angle
- ϵ=contact ratio
- v=speed on pitch line in m/s
- W=power transmitted in Watts
- X=vibration displacement amplitude normalized by static deflection

In practice the first two contributions consider geometric characteristic of the gear meshing. The second contribution considers the power that has to be transmitted in gear meshing. The fourth contribution takes into account the vibration caused by gear meshing. The result is given in dB of the acoustic wave amplitude. Noise sources are all the gear meshing. These meshing ave been considered positioned in the same point. According to masuda formula the Sound level pressure is measured at 1 m of distance by the overall Sound Source.

$$L_{tot} = 10log(10^{\frac{S1}{10}} + 10^{\frac{S2}{10}} + 10^{\frac{S3}{10}})$$

Where:

- S_1=Meshing 1 Level pressure sources
- S_2=Meshing 2 Level pressure sources
- S_2=Meshing 2 Level pressure sources

A this step of the trade off analysis this approach in useful in order to make a comparison between the two configurations. The focus is not in the precise value of sound level pressure, which will be calculated in a more precise way in a fining design stage, but in a indications about level pressure that can point out which of the gearbox configuration considered could be quieter. In the following figure3.9 is reported the SLP map of the fixed axes configuration.

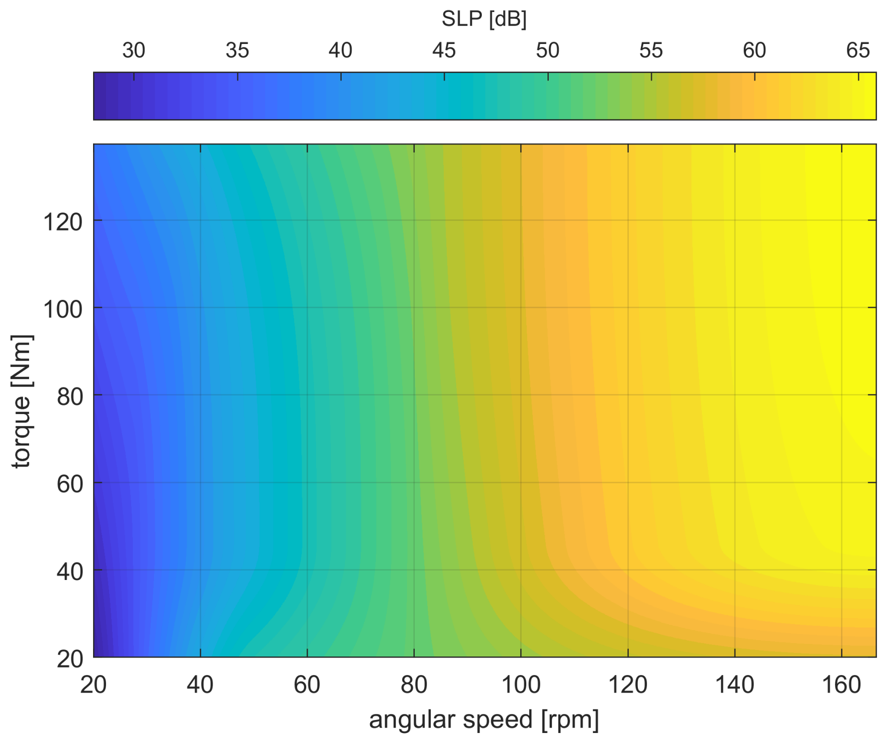


Figure 3.9: SLP map for fixed axes configuration

The plot confirms that sound level pressure is related to power transmitted but also on the term that consider the displacement vibration due to gear meshing. In particular this term creates a non linear trend at larger values of power transmitted. It can be also noticed that a larger area, below 80 rpm guarantees a sound level pressure smaller than 55 dB. As stated before this value has to be verified with more precise method, but it is even at this preliminary design stage, a good indication for what concerned noise prediction.

3.3 Planetary Set Architecture

The second gearbox configuration that has been investigated is a 2 stages planetary gear train. For the sake of simplicity the second and the first stage are equal. In this chapter are described gears, shaft and bearings sizing of the Kisssoft model that is illustrated in the following figure 3.10.

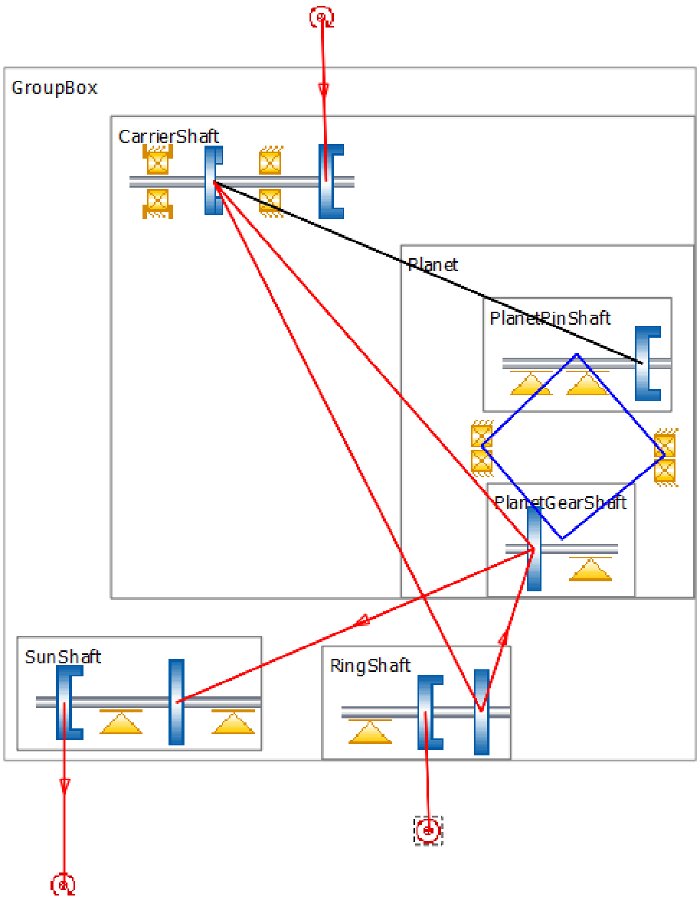


Figure 3.10: Planetary Stage flow

This figure represents the flow power throughout carrier shaft, planet shaft, ring shaft and sun shaft. The model consist of a main box which is *GroupBox*. This contains *CarrierShaft*, that contains *Planet* box. *PlanetPinShaft* and *PlanetGearShaft* represent coaxial shafts that support the planets. The scheme is then repeated for the second stage. The input Torque and Speed to the system are given to carrier shaft of first stage. The output Torque and Speed are at sun shaft. The ring gear is fixed. The output from sun shaft of the first stage is the input Torque and Speed for *Carrier Shaft* of second stage.

3.3.1 Gear Sizing

The input data are the same considered for the configuration with parallel axes:

$$T = 115Nm \quad n = 113rpm$$

Planetary stage is capable of perform higher transmission ratio, compared to parallel axes. Considering spur gears it is possible to reach multiplication ratio equal to 9-10 per stage, with a considerable improvement in term of compactness. The overall transmission ratio is the same considered for fixed axes configuration i=88. For simplicity the stages considered are equals.
Gears Sizing process has been carried out in Kisssoft, by means of *Rough Sizing* and then *Fine Sizing* routines, taking into account a compromise between efficiency and overall diameter. In the following will be described the gears geometric characteristics and strength calculated according to ISO 6336. The geometric characteristics of the gears are listed in the table 3.14. The material used for the following computations is the same considered for the previous fixed axes configuration which is 18CrNiMo7-6.

	Sun Gear	Planet Gears	Ring gear
z	15	52	-123
b[mm]	15.5	15.5	15.5
m[mm]	1	1	1

Table 3.14: Planetary set gears

In both stage there are three planets. The transmission ratio performed by this stage is 9.3, so as the total transmission ratio is equal to 88. Strength calculations are carried out taking as reference the ISO 6336 standard. In the following table are reported stresses results and the safety factors.

The minimum safeties that have been considered are: $SF_{min} = 1.4$ and $SH_{min} = 1.0$ values that are generally recommended.
Second stage is equal to the first stage. In the following tables 3.16are reported the results about stresses and safeties.

A stated at the beginning of this chapter, gear sizing has been carried out considering a compromise between meshing efficiency and dimension. The smaller is the ring gear the smaller is the efficiency. For this reason has been considered meshing efficiency per stage higher than 0.97 with diameter as close as possible to 88mm which

	Sun Gear	Planet Gears	Ring gear
SF	2.18	1.80	2.96
SH	1.04	1.21	2.31
σ_{root}	229.02	234.79/ 292.69	252.75
σ_{flank}	1184.97	1273.22/ 1322.61	1313.00

Table 3.15: Safeties and stresses first stage

	Sun Gear	Planet Gears	Ring gear
SF	4.62	4.13	6.84
SH	1.74	1.54	3.44
σ_{root}	105.60	108.25/121.34	104.78
σ_{flank}	1191.94	1237.41/ 1285.41	1276.07

Table 3.16: Safeties and stresses second stage

is the objective given from Electric Motor housing. The following figure illustrates the dimension of the first stage:

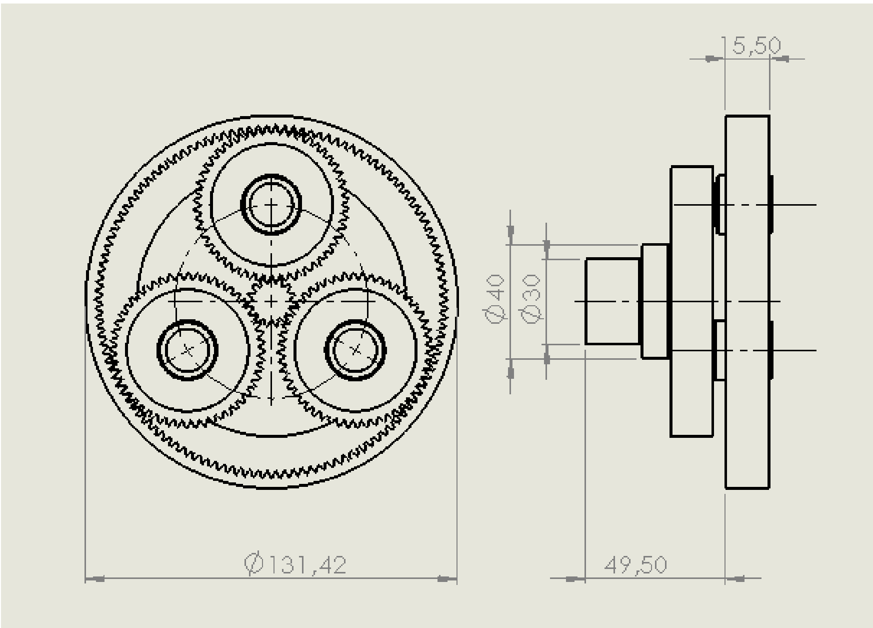
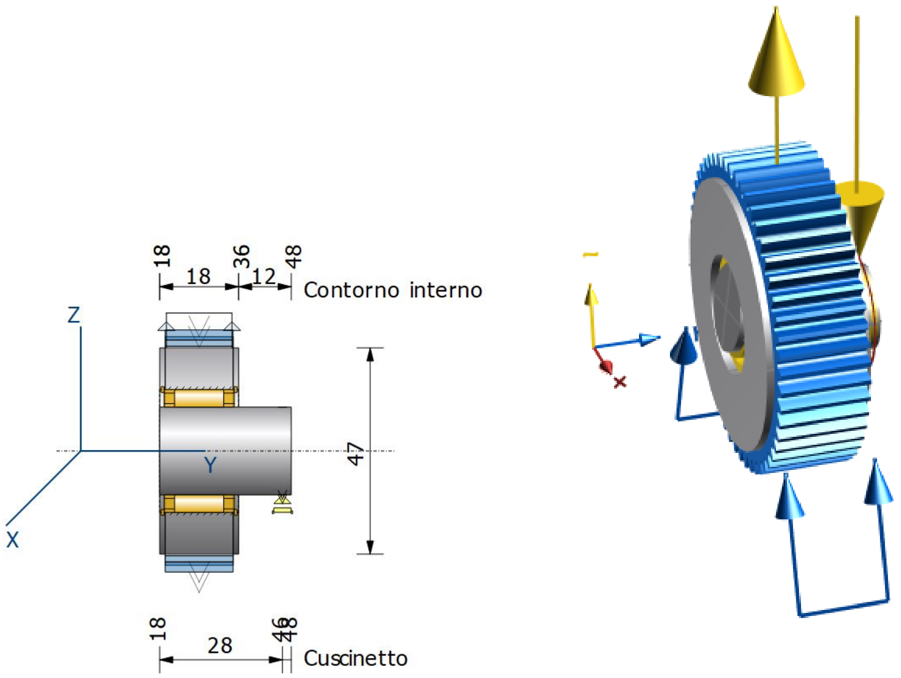


Figure 3.11: Planetary Stage dimensions

3.3.2 Shaft and Bearing Sizing

In this chapter Shaft and Bearings sizing in planetary gear train configuration are discussed. In the following figure is illustrated the Pin Shaft which is the shaft that supports planet gears, and the deformation due to load. Referring to the figure at the right, blue arrows corresponds to meshing load, instead yellow arrows are referred to the torque given by the *Carrier Shaft*.



(a) *Planet Pin shaft* (b) *Planet Pin shaft deformation*

Pin Shaft is rigidly connected to the carrier shaft. In this configuration has been considered a carrier that support planets from only one side. Different carrier geometries are possible, but the evaluation on this aspect is out of interest of this work. The important thing that has to be verified is the strength of the Pin Shaft.

Pin Shaft is supported from the support on the right side, that represents the cylinder of the carrier shaft. Hence, the Pin Shaft is loaded by the forces between Sun-Planet and Planet-Ring. The maximum deflection of the *PinShaft* is equal to $5.18 * 10^{-3}$. The pin deformation is similar for the second stage, but the values of displacement are smaller compared to stage 1.

Different bearings arrangements are possible. This depends on the carrier and gears geometry. In general, needles bearings are used for planets support. As illustrated in the previous figures it is considered a SKF K 20X28X16, fixed from both sides.

In the following figure is illustrated the deformation of the entire gearbox, considering two stages:

The input is given to right side to *Carrier Shaft* of stage 1. The output of stage 1, *Sun Shaft* is rigidly connected to *Carrier Shaft* of stage 2. There are no bearing between stage 1 and stage 2. The output of stage 2 is *Sun Shaft* illustrated to left side of figure 3.12. Th maximum deflection is reached by the sun shaft of stage 2 on the left equal to $32{*}10^{-3}$mm. As stated before, the bearing present in this gearbox

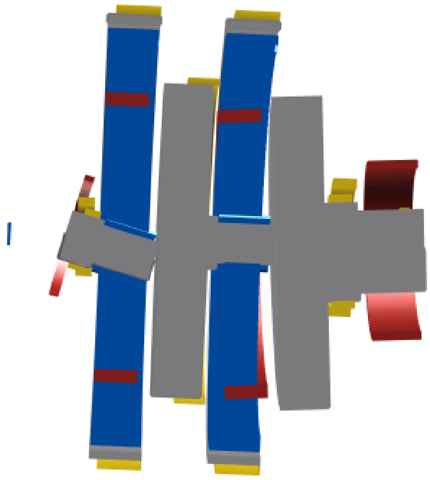


Figure 3.12: Main line shaft section displacement

configuration are 2, positioned at the end of the *Carrier Shaft* and *Sun Shaft* on the left end. In the following table are reported the code of this two ball bearing:

	Code	Service Life[h]
Bearing 1	SKF 61802	1000000
Bearing 1	SKF 6180	1000000

Table 3.17: Bearings *Main Line Shaft*

It is important to underline that at this design step has been considered simpler *Carrier Shaft* geometry. It possible to investigate other mainly, in order reduce total weight of the gearbox. This analysis will not be considered in this work.

3.3.3 Efficiency

The efficiency is calculated according the same standard used in the fixed axes configuration. Compared to previous configuration, the planetary configuration has lower number of bearings. However, gears meshing is larger in planetary gears. In the following figure is reported the efficiency map of planetary configuration.

It can be noticed that the maximum value of efficiency can be comparable to the configuration with fixed axes, equal 0.94. This is because the fixed axes solution has 3 stages, instead the planetary configuration has only 2 stages. Hence, the single fixed axes stage has higher efficiency compared to planetary gears. However, the total efficiency is similar in the two cases. The plot points out a trend which is different in this planetary configuration. In particular it can be noticed that efficiency begin to decrease starting from 100 Nm for every value of angular speed

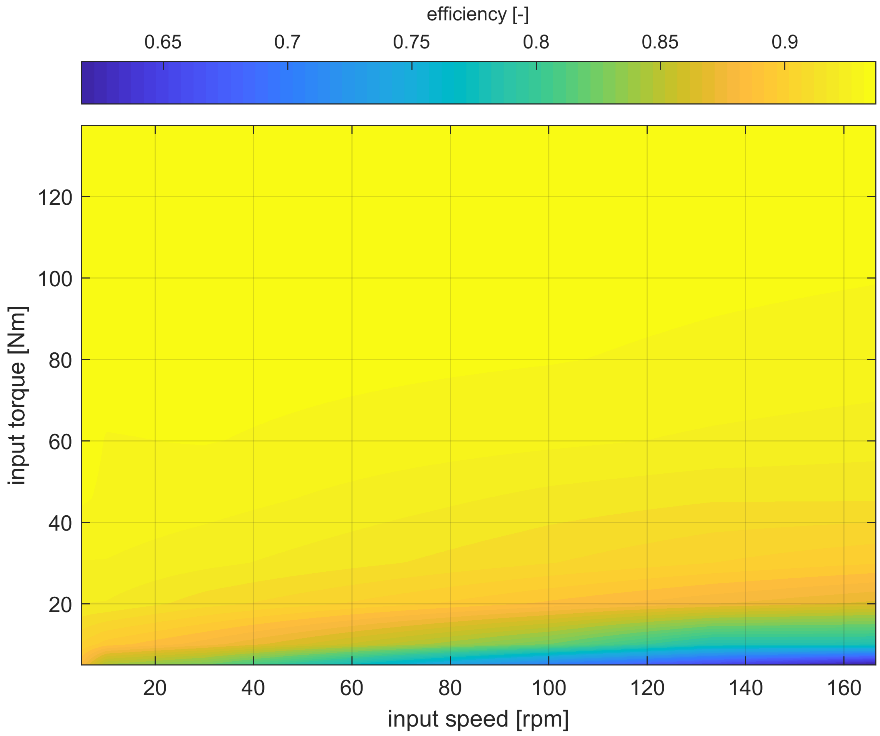


Figure 3.13: Efficiency Planetary configuration

given as input. The lowest values of efficiency are reach for very low value of torque, smaller than 20 Nm. The situation in term of efficiency get worse for higher value of angular speed. As considered in previous case the power losses sources than have been considered are given by:

- PVZ = Meshing Losses
- PVL = Bearing Losses

Meshing losses are due to engagement between Sun-Planet, Planet-Ring gears. On other hand the power losses due to bearings, correspond to losses in the two *Support Bearing* at input and output side and the losses due to *Connection Roller Bearing*. This contribute gives the larger amount of losses. This is due to the fact that both inner and outer bearing ring rotate. The higher power losses are cause by connection roller bearing of second stage. Instead the two bearing support are not loaded, so the power losses there are very small.

3.3.4 Sound Level Pressure

Sound level pressure is calculated according to Masuda formula[10]. The following map illustrates the SLP map for different range of Torque and Angular Speed. The trend in this case is slightly different from the SLP for fixed axes configuration. In particular, the power is mainly related to the power transmitted in gear meshing. Considering the colorbar, in this case the maximum value of sound level pressure is equal to the 75 dB.

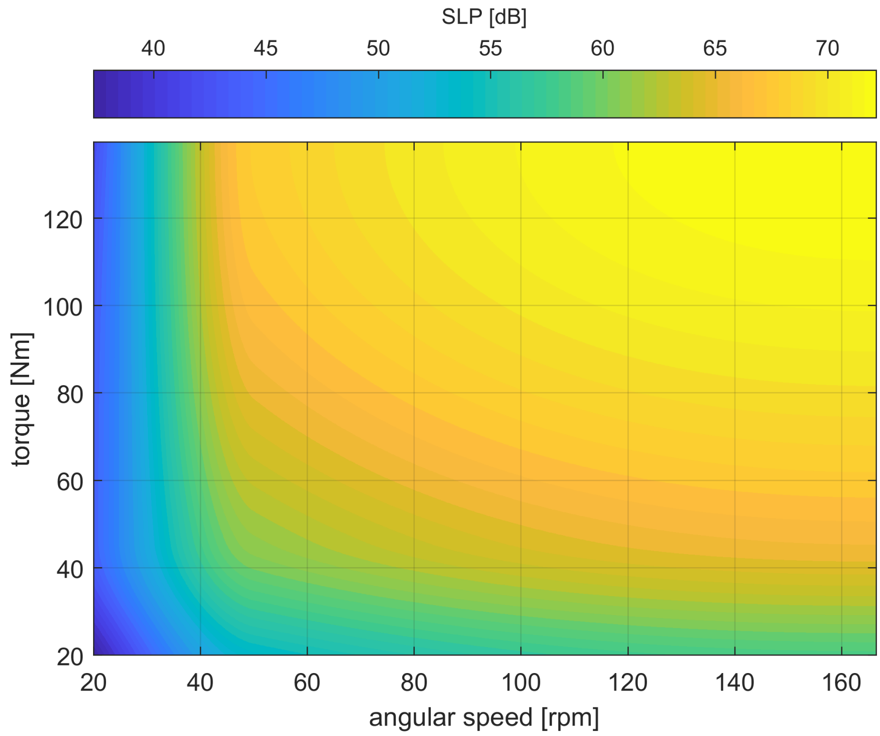


Figure 3.14: SLP Planetary configuration

It can be noticed that in this case of planetary configuration, SLP reaches higher value, above 65 dB, starting from 40 rpm as angular speed. Differently to what has been seen in previous case with fixed axes gearbox, the SLP increases linearly with power transmitted. The computation of sound levels have been computed considering as noise sources the gears meshing between: *Sun-Planet*, *Planet-Ring*. As seen for previous configuration these noise sources have been considered at the same point. Then is possible to compute the total Sound Level Pressure with the following relation measured at 1 m of distance:

$$L_{tot} = 10log(10^{\frac{S1}{10}} + 10^{\frac{S2}{10}} + 10^{\frac{S3}{10}})$$

Where *S1*, *S2* and *S3* are referred to noise sources caused by gears meshing.

In the following figure is illustrated the planetary drawing of the first stage. The second stage is equal to the first stage.

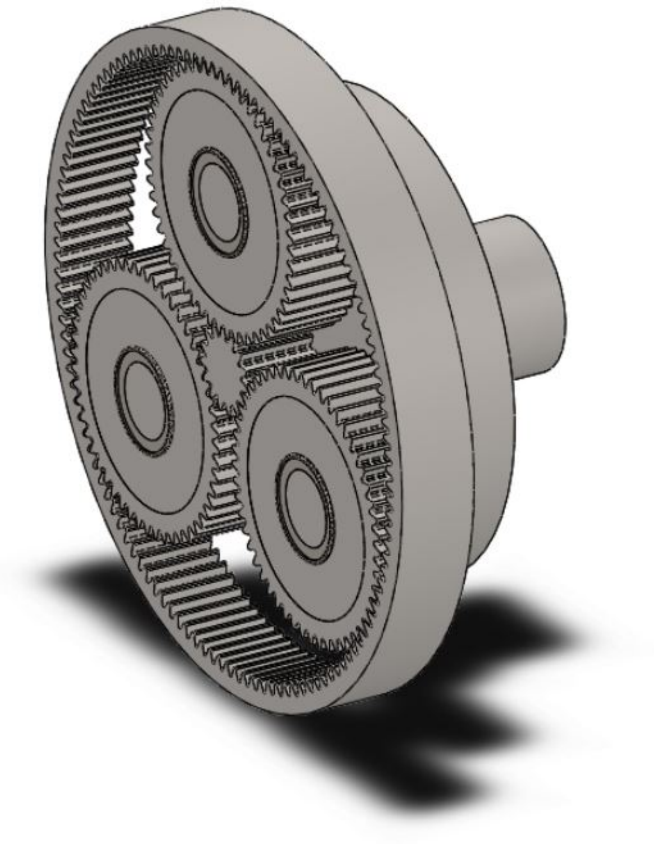


Figure 3.15: Planetary Stage dimensions

3.4 Equivalent Mass

Equivalent mass of two gearbox configurations has been evaluated. For both solutions the equivalent mass of the leverage and of the electric motor is the same:

- $m_{EMeq} = 14kg$
- $m_{LEVeq} = 0.255kg$

The former is a data, given from the characteristic of the Electric Machine. The latter has been calculated, referring to the figure 2.9. For the sake of simplicity lever PH has been considered vertical and lever HO has been considered orizontal in nominal condition. The length of the leverage are: PH=383mm and HO=72mm. Lever HO can rotate about one end which is point O. The mass moment of inertia can be calculated in following way, subscript 2 refers to lever HO. Considering lever HO as a cylinder of diameter 10mm and density equal to $8000kg/m^3$.

$J_2 = \frac{1}{3}m_2l_2^2$

For what concern lever PH, inertia property is simply the mass, m_1, because according to the previous assumptions this lever can only translate in vertical direction. Finally the equivalent mass the leverage can be calculated:

$m_{LEVeq} = m_1 + \frac{J_2}{l_2^2}$

This equivalent mass referred to leverage system is the same for both gearbox solution. It a simplification of the actual leverage equivalent mass, but it can be considered at this design step acceptable because during normal application the the stroke is very small.

- Equivalent mass moment of inertia Fixed axes configuration

  The equivalent mass moment of inertia can be calculated starting from the moment of inertia of the single gears:

	kgm^2
J_1	3.28e-4
J_2	1.45e-6
J_3	20.58e-5
J_4	7.52e-7
J_5	60.21e-5
J_6	8.43e-7
τ_1	4
τ_2	4
τ_3	5.5

Table 3.18: Equivalent mass moment of inertia

  The previous mass moment of inertia are referred to gears considering one of the power brache. The mass moment of inertia referred to the input side which is the side of the leverage can be calculated:

$$J_{eq} = J_1 + (J_2 + J_3)\tau_1^2 + (J_4 + J_5)\tau_1^2\tau_2^2 + J_6\tau_1^2\tau_2^2\tau_3^2$$

  This value is the equivalent mass moment of inertia of the fixed axes configuration gearbox. It is possible to compute the equivalent mass of the gearbox, dividing the mass moment of inertia by the leverage ratio. Then adding at this value the equivalent mass of the leverage, the total equivalent mass can be calculated. The total equivalent mass is equal to $m_{eqTOT} = 27kg$

- Equivalent mass moment of inertia Planetary configuration

The same approach can be considered for what concerned the planetary configuration. In this case the equivalent mass moment of inertia can be computed referring to the carrier shaft. In the following table are summarized the value of mass moment of inertia of the single gears of planetary stage. The inertia of the shafts and bearings are neglected.

	kgm^2
J_c	9.5e-4
J_p	81.55-6
J_r	92.61e-5
J_s	6.88e-7

Table 3.19: Equivalent mass moment of inertia

The masses and radius related to planets are: $m_p = 0.196kg$ and $r_p = 70e{-}3m$. The angular speeds of the single parts of the planetary gearbox are:$\omega_r = 0rpm$, $\omega_p = 13.27rpm$, $\omega_s = 46rpm$ and $\omega_c = 5rpm$. The subscripts used are referred to *Carrier*, *Sun* and *Planets*. The total mass moment of inertia can be calculated:

$$J_{eq} = J_c + 3m_p r_p^2 + J_s(\frac{\omega_s}{\omega_c})^2 + 3J_p(\frac{\omega_p}{\omega_c})^2 = 0.528kgm^2$$

Ring gear contribute to mass moment of inertia is equal to zero because the the ring gear is fixed. From the calculation of the mass moment of inertia is possible to calculate the equivalent mass of the gearbox seen at the input side:

$m_{eqGB} = \frac{J_e}{\tau_{lev}^2} = 40kg$

As seen in previous case it is possible to calculate the total equivalent mass adding to leverage and gearbox the equivalent mass of the electric motor.

$m_{eqTOT} = m_{eqLEV} + m_{eqGB} + m_{eqEM} = 55kg$

The main contribution in this case of planetary set is given by the revolving planets. For this reason the value of equivalent mass at the input side is higher respect to fixed axes configuration.

In the following table are summarized the results referred to inertia properties in both Gearbox configurations:

	Fixed Configuration	Planetary Configuration
$J_{eq}[kgm^2]$	0.164	0.528
$m_{eq}[kg]$	12.43	40
m_{tot}	3.2	3.9
$m_{eqtot}[kg]$	27	55

Table 3.20: Equivalent mass moment of inertia comparison

From the previous table can be noticed than there is a considerable difference between the *Fixed axes* and *Planetary*. The term m_{tot} refers to total mass considering *Bearings*,*Shafts* and *Gears*. On other hand in term of equivalent mass, the difference is equal to 28 Kg comparing fixed axes and planetary configuration. The higher value of equivalent mass of planetary configuration is due to planets masses. In particular, because the planetary multiplier has large diameter for planets gears, the inertia contribute in order to revolve these parts affect considerably the total equivalent mass.

3.5 Comparison

In this section are summarized main characteristics of the two gearbox that have been investigated before. In particular comparing the parameters: *efficiency*, SLP, *overall dimension* and then *equivalent mass*. The total efficiency for two gearbox is calculated considering also the efficiency of the electric motor. In the following figures is illustrated the efficiency map of the Electric Motor considered as reference.

The value of total efficiency is comparable in both fixed axes and planetary set configuration, even though the efficiency of single stage is lower in planetary set than fixed axes configuration. The overall efficiency is 0.8125 for fixed axes gearbox and 0.8099 for planetary set. The efficiency comparing two configurations is mainly affected by the efficiency of electric motor. For this reason the difference between two configurations is not evident. However, it is clear that fixed axes configuration has an higher efficiency considering single stage.
Sound Levels Pressure have been calculated with Masuda formula, which is a simplified approach. For this reason the absolute value of dB Sound Level Pressure could be not so precise, but in this case it is a good tool in order to make faster comparison, and figure out which is the quieter gearbox. The results point out a difference between two configurations. In particular fixed axes gearbox has a maximum at 65 dB. Planetary set has a maximum at 72 dB. Furthermore, it can be noticed a difference in the trend of the Sound Level pressure: according to Masuda formula, level pressure is mainly dependent on the power transmitted through gears meshing but also the contribution of vibration in fixed axes gives a contribution that cause a trend which is not linear. The following figure illustrate dimensions of the gearbox that have been sized.

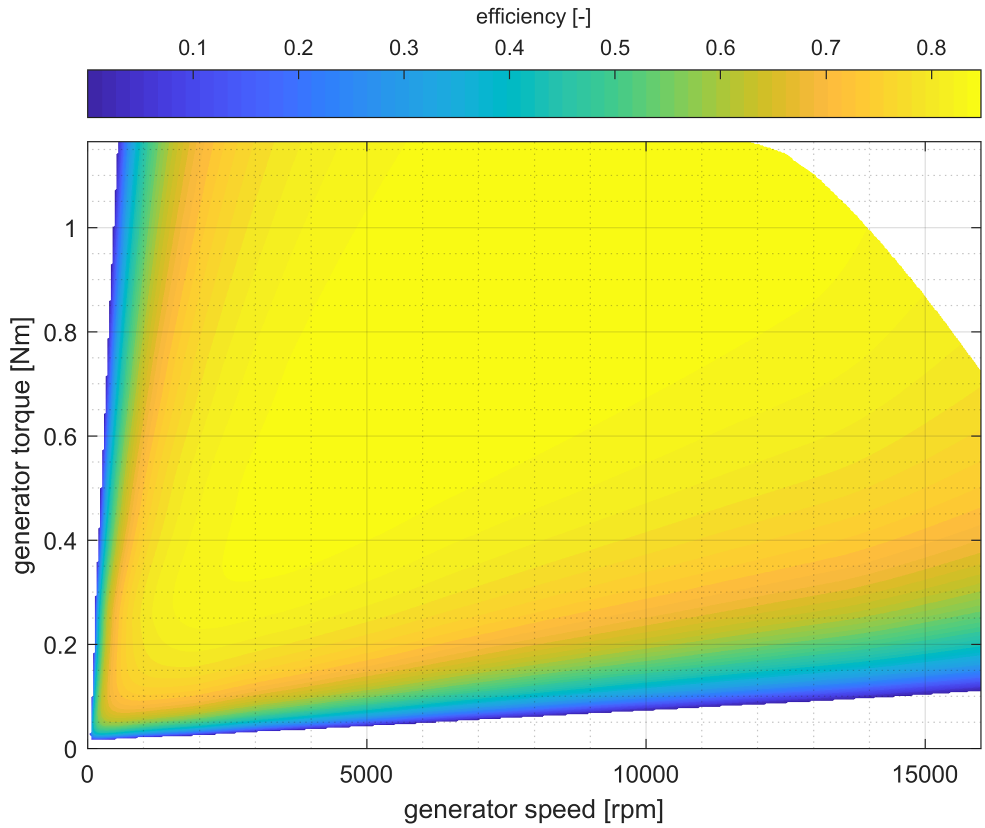
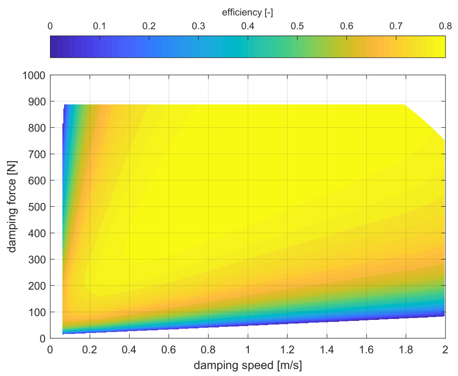


Figure 3.16: Electric motor efficiency map

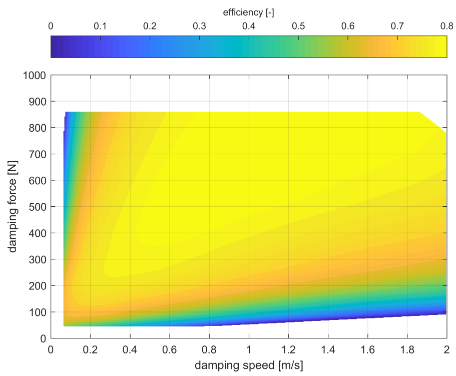
From this electric motor efficiency illustrated in Fig. 3.16 map is possible to calculate the total efficiency of the system multiplying the efficiency of the gearbox by the efficiency of the electric motor map. The results are illustrated in the following plots. At this stage in the overall efficiency of the system the leverage efficiency is neglected.

It is clear that the efficiency of the overall system is mainly affected by the efficiency of the generator. The overall efficiency of the system is similar for both configurations: fixed axes and planetary set. The value of the total efficiency is close to 0.8. Considering previous figure it is possible to see that the area at maximum efficiency is wider for fixed axes configuration.
The overall dimensions of the two configurations are illustrated in the following figure.

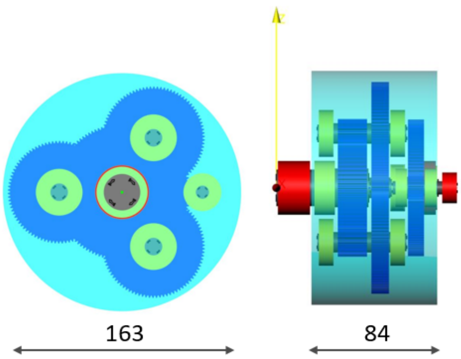
The dimensions illustrated in previous figures are in *mm*. The difference between two configuration is important considering diameter dimension. For fixed axes configuration the total diameter is 163mm, whereas for planetary configuration the overall diameter is 136mm. The length in Y direction is similar for both cases: 84mm for fixed axes and 83mm for planetary configuration. The length for planetary configuration is mainly affected by carrier width, in Y direction. In order to reduce deflections due to torque transmission by carrier shafts this width is manda-



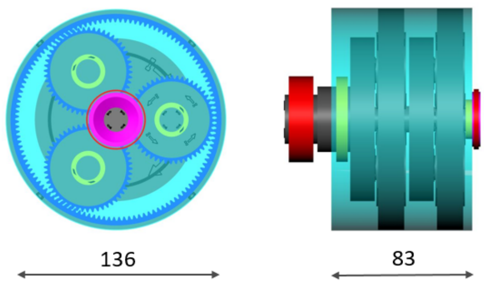
(a) *Total efficiency parallel Axes*



(b) *Total efficiency planetary Axes*



(a) *Overall dimension parallel Axes*



(b) *Overall dimension planetary Axes*

tory. However, this depends mainly on carrier geometry. In fact, as illustrated in previous section has been considered a carrier that supports planets only from one side. The deflection has to be taken into account in order to guarantee constant working conditions, considering that for this kind of application the gearbox is load with load spectrum in which the torque changes direction continuously. For these reason, the deformation can be reduced increasing width in Y direction of the gearbox.

Chapter 4

Conclusions

The objective of this work was to carry out a preliminary trade off analysis regarding *Electromechanical Rotary Shock Absorber*. The focus was the mechanical part of this kind of damper. In particular *Leverage Arrangement* and *Gearbox multiplier*. Four different solutions regarding leverage arrangements have been investigated. First two solutions can be considered the simpliest solutions, the drawback regards the very high transmission ratio required for gearbox. Third solution is the best considered from transmission ratio required from gearbox. However, this big advantage bring a practical problem due to the arrangement of spring element. For this reason, has been decided to consider fourth solution. This solution results as a good compromise between gearbox ratio required and modifications on suspension scheme. It is important to underline that the aim of this work is to figure out the potential of this type of shock absorber. It is clear that if experimental tests carried out on a prototype are positive, the suspension has to be design from the starting point. As consequence of these considerations, fourth solution has been considered for gearbox design. Initially, different type of gearbox have been considered. Simple parallel axes configuration has been discarded because of high value of transmission ratio required, that would have cause high dimensions. In order to reach compactness has been taken into account power split. This configuration has double advantage: reduction of module requested because the torque transmitted is divided throughout fixed planets and circular symmetry of the system. This fact is an advantage for what concern vibration and dynamic response of the system. However, also in this case has been necessary to add one last stage with one gear meshing.
Then planetary configuration has been considered. An example of speed multiplier gearbox is given by wind turbine gearbox. Different configurations are used typically the fist 2 stages are planetary and then is added a third single parallel axes stage. The objective in wind gearbox is to increase speed, from turbine to generator, reducing noise as much as possible. Taking into account these information, has been decided to investigate 2 stage planetary gearbox.
Gear pairs, shafts and bearings has been sized for both configurations according to ISO standards. Then efficiency analysis have been carried out considering power losses due to according to standards: gears meshing, bearings and lubrication. The

efficiency calculation pointed out minimum difference between fixed axes and planetary set. For both configurations, considering the also the efficiency of the generator, is close to 0.8. The efficiency of the leverage system has been considered reasonably equal to 1. In leverage system power losses are due to deformations in leverage and friction in uniball joints. In this work this contribution has not been taken in consideration.
Preliminary evaluation of noise generated by two gearbox has been carried out. Masuda formula is a fast method to figure out which is Sound Level Pressure generated by meshing, comparing two configurations. The results shows that fixed axes is quieter than planetary set, in particular for a wider area of the the map Torque-Speed. However, it is important to underline the fact that this method does not take into account complex phenomena that affect noise that regard lubrication. The results point out that planetary set is slightly noisier compared to fixed axes configuration. Equivalent mass has been computed for both configurations. It is interesting to noticed the difference of two configuration. Planetary set has higher Equivalent mass due mainly to revolving contribute of planets. A possible to avoid this problem is to keep fixed carrier and to connect to the input the ring gear. In this way the revolving contribute will be equal to zero. On the other hand the transmission ratio will be different so the total transmission ratio, keeping same gears sizing, will not be guaranteed. Equivalent mass has to be as small as possible, in order to have constant damping characteristic in frequency. In this work dynamic analysis have not been considered. However, from the results illustrated before fixed axes have this advantage.
Finally, planetary set is advantageous from dimension point of view. However, dimensions is not the only parameter to consider. Considering the small potential amount of energy that it could be possible to harvest, efficiency is the first objective to consider. This preliminary work pointed out that both configurations have advantages that have to be analyse in details in future in order to proceed with the realisation of a prototype and carry out experimentally the real energy harvesting potential from road roughness.

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