POLITECNICO DI TORINO

Master degree in Automotive Engineering

Propulsion System Development

Master Degree Thesis

Test bench set up for the evaluation of the Static Transmission Error in gearbox

Supervisor:

Professor Carlo Rosso

Correlators:

Fabio Bruzzone

Tommaso Maggi

Claudio Marcellini

Candidate:

Filippo Maria Greco

S257584

Academic Year: 2020/2021

I take this opportunity to thank Professor Rosso and the engineers who contributed to my thesis work, helping me in this fantastic experience.

I thank all my family and those who unfortunately are no longer here for the support and the strength that they give to me in the most difficult moments.

I thank all my friends for making me smile all the time.

1 Abstract

The fulcrum of this thesis is the study of the Static Transmission Error of two meshed gears, in this specific case spur gears, which is the main proponent of the problems related to noise, vibrations and roughness.

Starting from the aforementioned premises, in this thesis, a careful analysis on different Static Transmission Error values was obtained through a comparison between the measurements obtained from a previously designed test bench and a dedicated software called GeDy TrAss.

First of all it is described what type of phenomenon is the STE, after which the test bench is summarized in all its characteristics describing the improvements made with respect to the first configuration.

Considering that the test bench must give truthfulness and act as an experimental counter-test to the GeDy TrAss software, some features are explained in order to better understand how it works.

At this point of the work the subject that will be analyzed comes into play, that is a particular example of straight-toothed gears used by NASA in 1996.

In order to compare the STE results of the literature with those obtained from the test bench, a MATLAB code is developed to analyze and process all the data collected by the test bench itself.

The final part of the work includes the comparison of the results between the test bench, literature and GeDy TrAss software.

Based on several tests it is possible to draw conclusions and essentially provide some possible future implementations.

2 Introduction

Nowadays the automotive companies continuously try to demand an improvement of the performance both to be able to compete with other brands and to be able to enter new emerging sectors such as purely electric handling.

Focusing on our field of study, it is important to understand how to deal with some problematics in the gear boxes, the new trend is work on the components, the only way to improve the behavior of the gears , in order to satisfy that demand is optimisation and reduction of weight.

This approach usually means increasing of deformability that presupposes running into NVH and overloads problems, at this point it is essential to analyze what STE is and the causes that entail it.

2.1 Static Transmission Error

The static transmission error in gears represents the main noise and vibration source of mechanical transmissions, both for self-excitation and for the excitation of powertrain components, gutting up the Static Transmission Error is possible to underline that is the most important cause of the aforementioned problems, that means the enormous importance of the study of this phenomenon, this kind of Error is defined as the difference between the actual angular or linear position of the output gear and where it should be if the transmission was kinematically perfect, practically difference between engaging gears and the same gears in operating conditions under load (quasi-static).

The formula is:

$$Static\ Transmission\ Error = \theta_2 - \frac{z_1}{z_2}\theta_1$$

and is possible to recognize with ϑ_1 the angular displacement of the pinion, the driving gear and ϑ_2 the same but for the driven gear, and with *z* as number of teeth for both gears.

Due to the fact that the main source of excitation in gearboxes is generated by the meshing process, it is important underline that its characteristics depend on the instantaneous situations of the meshing tooth pairs but also the applied torque and tooth micro and macro-geometry.

Under load at very low speed (case of static transmission error), these situations result from tooth deflections and manufacturing errors, it is important to understand that is measured under sufficient minimum torque to take up the backlash and at a speed low enough to render dynamic effects negligible [1].

In order to analyze the phenomenon of STE, it is possible to focus on the theory that the irregular transmission of motion can be visualized by means of the motion curves from which the difference between the maximum and minimum of STE is called Peak to Peak Transmission Error (PPTE).

The transmission error therefore has a periodic trend, therefore it will consist of various component harmonics and shows that it is amplified by the load and poor quality of the teeth and therefore strictly influenced by the tooth profile modification.

The transmission error can be understood as construction (measured empty on the single flank test) and under load (measured at a certain torque).

The PPTE under load can be shown with the following formula:

$$PPTE = \frac{F_t}{b} \cdot \frac{q}{c'} + ppte_{construction}$$

where is possible to see the specific load *Ft* on the tooth *b*, the stiffness of the tooth *c'* and *q* as the covering oscillation along the meshing line (due to the variable succession of pairs of teeth engaged along the meshing line).

$$Oscillation = \varepsilon_{\alpha max} - \varepsilon_{\alpha min}$$

and from this oscillation it is possible to define a silence factor q, that if it has high values means low noise index :

$$q = \frac{1}{\varepsilon_{\alpha min}} - \frac{1}{\varepsilon_{\alpha max}}$$

Instead of, the construction transmission error is due to:

- profile errors (generate noise components at all harmonics of the meshing frequency),
- single step errors (generate noise components at all harmonics of the rotation frequency),
- gear concentricity defects (they generate noise components at the rotation frequency),
- misalignments (generate noise components at 2 times the meshing frequency).

3 Test bench

The test bench was designed to give an experimental counterpoint to the results obtained with the internal software, in order to give a foundation of truthfulness to the results.

The main goal of the design of this test bench was to create a versatile archetype in order to work with various load situations and with various types of gears.

There were essentially two key points, first of all obtaining a so-called "open loop" circuit (it guarantee the possibility of actively intervening and therefore having greater flexibility) and secondly the recirculation of energy [2].

The bench can be divided in two macro elements weights and pulleys to generate driving and breaking torque and supports to be flexible.

Speaking in details the setup can be divided into five subgroups:

- Structural parts
- Transmission of the motion
- Measurement system
- Security system

3.1 Structural Parts

The structural parts provide the sustainment of the transmission of the motion and of those measuring equipment and in the meanwhile ensure high stiffness and flexibility at the whole system.

First of all, the bench is provided of two sets of weights that are placed at the right and left side of the support where respectively the right weights are used for the input torque, the left ones for the braking torque and vice versa in the case of a precise inversion of the sense of rotation, those weights are connected through a rope on series of pulleys to the fixed and movable supports, in order to close this loop of forces.

The upper part of the support Figure 1, has an hoist lever that guarantee the transmission of the torque from the weight support to the shaft on the movable support.

Figure 1, Stand frame

The structure located under the weight support, as it is possible to see in Figure 2, is called fixed, it is composed by rigid metal profiles where the input shaft group is located and locked.

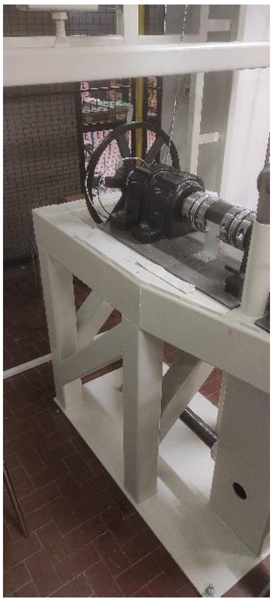

Figure 2, Fixed support

In front of the fixed part there is the movable support, Figure 3, on which there is output shaft group and thanks to electro-permanent chuck is possible to lock the support.

Figure 3, Movable support

After having mounted the two gears, one on the fixed and the other on the movable support, it is necessary to find the right distances to engage themselves.

At this point it is important to underline that the movable support is the main of the bench.

Assuming that the reference system is set having on the vertical Z axis, on the direction of approach between the two Y axis gears and on the horizontal X axis, the movable support, initially, permit and provide the ability to adjust at macro-dimensional level the driven gear to the driving one, moving with the help of a pallet truck, then is possible to ensure the micro-adjustments, that are linear displacement along Y and Z axes and at the same time rotation around X and Z axes, these kind of settings are made possible because the output shaft is mounted on a rotatable platform, in order to suppress each eventual misalignment or to see how much it can accentuate the STE value.

For what concern the back part of the movable support there is a trapezoidal threaded spindle mechanism for the vertical adjustment, that guarantee vertical shift of the gears, in order to take under control and change the centre distance between the shafts when the gears are engaged, also in this case to understand how can change the trend of STE.

This type of adjustment is allowed thanks to two threaded metal rods underneath the mobile structure, thanks to two bolts, one at the top and one at the bottom, it is possible to reduce the distance from the ground of the part in order to adjust the height of the driven gear, these bolts are of large dimensions and must be unscrewed before adjustment.

As a last step for adjustments there are four threaded pins positioned at the vertices of a rectangular vertical plate for small shifts along Y or rotation around X.

Naturally, an attempt was made to design a test bench as reconfigurable as possible and thanks to the micro and macro modifications it was possible to use any type of cylindrical and bevel gears, such as spur, helical, straight and spiral bevel gears, more in detail focusing on the shafts, if they are both parallel to the Y axis it is possible to use and test cylindrical gears, instead moving the mobile support along the X axis, always perpendicular to the Y axis, it is capable to test bevel gears.

3.2 Transmission of the motion

The transmission of motion, thanks to the 5 mm steel rope that passes inside the previously arranged 110 mm (diameter) pulleys path, allows a passage from the weights to the gears themselves, where the gear placed on the fixed support will see an input torque instead the gear mounted on the movable support will consequently see a braking torque, and this can be the exact opposite if the direction of rotation is reversed.

The loads are transferred to the fixed and movable supports, in order to guarantee the right torque that is useful during the different tests.

In our configuration we have for both the side 125 kg of mass usable to produce torque

$$Torque = Force * Arm = Mass * acceleratio * Radius$$

and in our case we have r = 0.2 m that is the radius of the bigger pulleys connected one to fixed and the other to movable support, the mass can be divided into ten plates of 5 kg each, then adding the anchoring part for a total of 125 kg.

Starting considering that the tests have to be done in quasi-static conditions, after choosing the right starting weight, that have to overcome at least the force given by the frictions of the entire system, thanks to an hoist the weight chosen to act as a tractor can be rise and it is possible to proceed with the process, the weight that provides the input torque drops while the weight that acts as a brake follows the opposite direction.

The weights can slide on two lubricated bars about two meters high, there is an effective stroke, excluding the maximum height that can be reached with the configuration that involves the use of all weights, approximately of one meter.

By doing some small calculations, thanks to gravity, when the configuration is at the maximum possible, that is 125 kg, a total force of 1250 N is obtained, but considering that an additional pulley path has already been set up during the design phase, this can be doubled value guaranteeing 2500 N of maximum force and a torque of 500 Nm.

3.2.1 Improvements

Respect to the previous configuration it has been necessary to implement a new arm profile on the right of the structure, in order to mount an electric hoist, Figure 4 , it is from Einhell and can guarantee a maximum capacity of 250 kg with the double wire configuration, perfectly abundant compared to our weight to be handled.

Figure 4, Electric hoist

This modification has been designed to make it less burdensome and to speed up the vertical movement of the weights, furthermore this variant allows to guarantee a continuity of sliding at almost constant speed, essential to make as constant as possible the torque values recorded.

During some test was possible to notice and realize that when the weight flowed on the steel bars it had an anomalous behavior going down in jerks and this led to peaks and oscillations in the values read at the output, to overcome this problem it was decided to install linear bearings, shown in the Figure 5, two on each side, they ensure greater fluidity of motion and reduce possible vibrations and further friction.

Figure 5, linear bearing

3.3 Measurement System

In order to view and compute STE values, the test bench is equipped with some measurement equipment that allows the user to interface with the analysis and to monitor the parameters involved.

Focusing on the Static Transmission Error it must be considered that it is closely related to the level of torque, for this reason it was necessary to install on the fixed support a torque meter useful to measure instantaneously the trend of the exchanged torque.

It was decided to use a torque meter T22/500Nm from HBM GmbH, Figure 6, that has the peculiarity of detects the value without any retroactive feedback on the weights, this measurement is useful for understanding the effort to which the pair of gears are subjected.

More in detail, Volt peak to peak (Vpp) signals are collected from the Torque meter, whose values range from 0 to 1024, where 0 means 512, which are then subsequently processed by an Arduino Mega2560 board, in the meantime a power supply allows to guarantee a continuous current to the board itself, after which thanks to MATLAB which reads data from Arduino it is possible to interface and transform the signal into a torque value.

Arduino for MATLAB is able to reads the value from the specified analog pin. Arduino boards contain a multichannel, 10-bit analog to digital converter. This means that it will map input voltages between 0 and the operating voltage, 5V, into integer values between 0 and 1023.

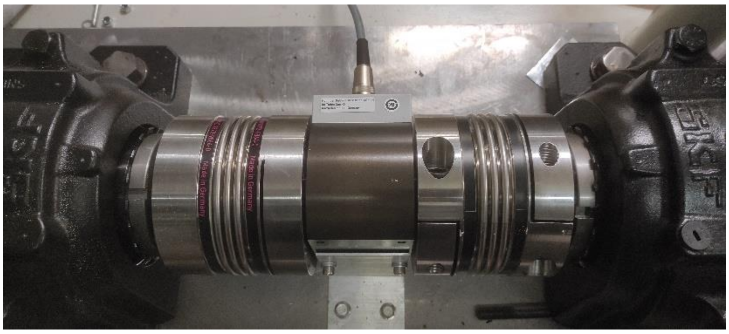

Figure 6, Torque meter

On the two sides of the torque meter there are metal bellows joints R+W BKM 1000 by R+W Italia s.r.l, instead near each gear there are respectively an Heidenhain RCN8580 encoder, Figure 7,which is able to identify its angular position in a precise and immediate way, Heidenhain has its own software that allows to obtain the angular variation of both devices and then process the data on Excel or MATLAB

Moving axially on the shaft, after the encoder there is the decoupling mechanical joints, Figure 7, this kind of mechanical joint take inspiration from quill drive, old mechanism in locomotive, it is composed by two collars, where one is connected to the only rotating part of the encoder and the other one is connected to the steel spacer on which the gear rest.

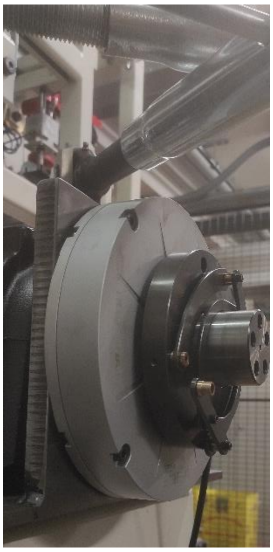

Figure 7, Encoder, quill drive and tie rods

Observing between the two collars, it is possible to note a series of elements used to unite at kinematical level the two parts.

The fixed part of the encoder is blocked on a vertical plate, welded to the steel profile of the support and the shaft passes through them and through five screws is axially fixed to the head of the gear.

Thanks to this decoupling joint the deformations exerted by the gear are decoupled, in fact the torsional deformation, that is the main of interest for the STE evaluation, is transferred from the collar on the right to the left one and can be read by the encoder and in the meanwhile the others are absorbed by the elements between the two collars, in short words, the joint decouples rotation and bending when the gears are under load

3.4 Security System

For safety reasons, during the various tests, the mobile turret is blocked by the two magnetic plates previously described which, once excited, join with the platform making the whole structure perfectly stable and fixed.

To ensure further stability to the coupling of the two toothed wheels, the use of two tie rods, Figure 7, is envisaged that connect the mobile turret to the fixed one, giving a high rigidity in the coupling phase.

To protect the entire work area, even for legal reasons, there is a cage with a safety lock that disconnects the entire machine if the door is opened. On this cage there is an external push-button panel that allows to control the main functions of the counter.

4 GeDy TrAss software

The test bench analyzed was born to make a comparison between the experimental value and the value produced by a software, all the work revolves around the comparison with the outputs generated by the software GeDy TrAss is a powerful creation for the design of mechanical transmissions.

It consists of two main tools:

- GearDraft, to be used during the transmission pre-design phase, when only the design constraints are known,
- OptiMicro, which aims to optimize the micro-geometry of the components considering static and dynamic phenomena.

4.1 Geardraft

GearDraft is the first of the two tools, it is mainly based on ISO standards and depending on the designer's need it can be used both for the verification of components already existing both for sizing. Thanks to the algorithm created, it is possible to generate an entire mechanical transmission in a short time, but above all automatically inserting a very limited number of inputs, a necessary thing to mention is the timing, it is the strong point of the software, as it can significantly reduce the calculation and computation time.

The constraint parameters currently available in the software are: torque to be transmitted, encumbrance from comply, transmission ratio and material. The code, using different modules, is able to verify and size, always according to legislation, a wide range of components, such as bearings, shafts, gear and synchronizers, thus leaving the user the freedom to choose the constraints design, generating a CAD / FEM model as output.

GearDraft allows, therefore, to save time by eliminating the often hand-made pre-design operations.

4.2 Optimicro

The second tool is called OptiMicro and has the function of optimizing the transmission components, being able, also, to integrate the results obtained by GearDraft. The optimization is based on both static and dynamic conditions of all the elements present in the gearbox, focusing more on components such as sprockets. The core of the operation is due to a patented algorithm (IT- 102018000001328), thus allowing to provide the study of the quasi-static behavior e dynamic system ensuring high accuracy.

Taking the deformed as input it is possible to extrapolate the Load Sharing Factor (LSF) or the distribution of loads between the couplings of teeth in contact and Static Transmission Error (STE) as in Figure 9.

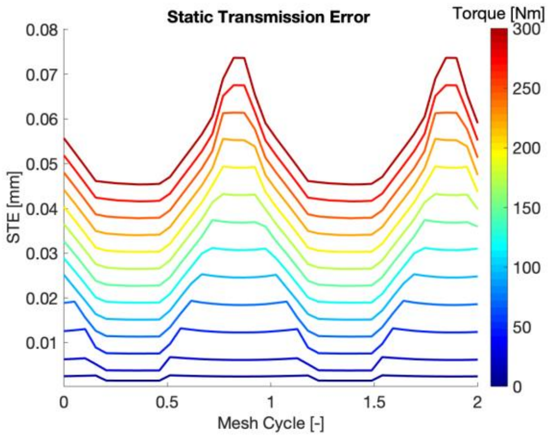

Figure 8,STE calculated with OptiMicro at different load conditions.

This last possibility allows us to create graphs with each STE for different load and consequently it is possible to compare these STE values with those obtained from the test bench, in order to be able to guarantee different comparison between calculated and experimental values.

Thanks to OptiMicro it is possible to search for the deformations of the toothing under load and contact between bodies of any shape.

The focus is on using a three-dimensional non-Hertzian contact model, which allows to abandon the simplifying hypotheses of the classical method; more in detail, in order to compute the forces exchanged between the engaged teeth of the gears and the resultant local displacements, the pressure distribution must be calculated, the way to solve this calculation is due to the fact that the software works on an own algorithm, which analyze the non-conforming contact conditions through the geometries.

The algorithm, iteratively, starts by considering whether the teeth have meshed or not, if the meshing has occurred, in the contact direction, the displacements are calculated by superimposing both the surface of the pinion teeth and that of the teeth of the driven gear, point within the designated contact area.

The displacements allow to calculate the pressure distribution of each node and the total exchanged load P divided among the meshing couples of teeth, the only thing to pay attention to is that some precautions must be made to ensure that the pressure difference, obtained from which data as input, is in a certain range.

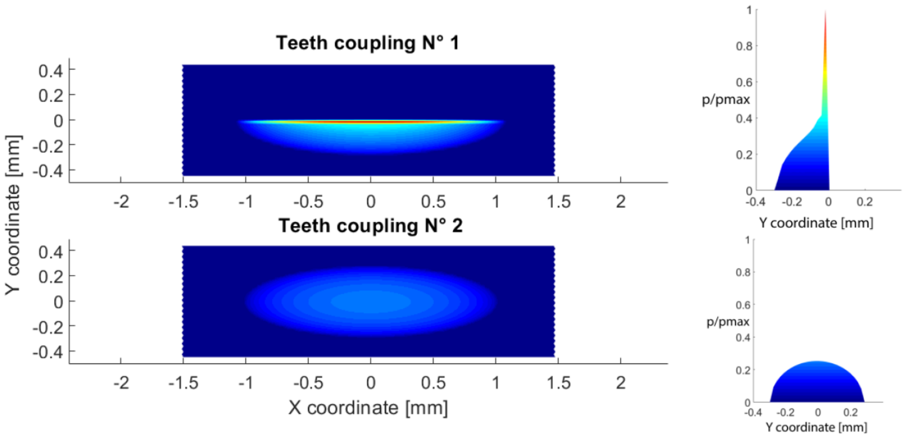

Figure 9,Pressure distribution and peak, no tip relief

In Figure 10, it is possible to observe that the teeth coupling 1 is about to leave the contact: it is noticeable that due to the absence of a tip relief modification, the tooth next to the recess condition has a peak of pressure, as it is show in the graph p/p_{max} ,all along the face-width at the tip, while in the teeth coupling 2, which are in contact, there is no pressure peak [5].

OptiMicro guarantees considerable versatility, as it is able to analyze gears with any type of profile modification, such as linear tip relief, parabolic, involute crowning or face width crowning.

Thanks to pre-existing static and dynamic analyzes, OptiMicro allows to obtain more performing components with greater durability in operating conditions, returning the best solution in terms of flexural stress at the base of the tooth and pitting due to flank contact.

5 Gears

The test bench described in the previous chapters allows you to assemble and analyze the gears that are the subject of this thesis work.

The following gears have been reproduced according to the specifications described and present in the literature.

These are gears of considerable importance, as they were the subject of a study by NASA in 1996 [3].

We are talking about standard spur gears with low contact ratio, in more detail with correction for the breaking of the cutting edge at the tip of about 0.012 '' less than the nominal value and linear discharge of the tip starting from a roll angle of 24, 5 ° with a total amount (at the tip of the tooth) of 0.0010 ''.

Figure 8 shows the subject under examination, while Table 1 collects the main characteristics.

In detail, the gears being used do not have tip relief and the gear has been drilled to be mounted on the bench shafts, so that they can be tested.

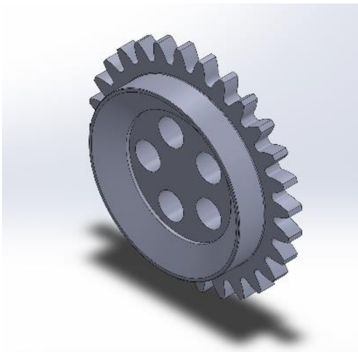

Figure 10, CAD of the gear analyzed

z		28
αn [°]		20°0'
mn [mm]		3,175
β [°]		0°0'
b [mm]		6,35
xm [mm]		0
r [mm]		0,952
Dp [mm]		88,9
Db [mm]		83,539
Df [mm]		80,962
Da [mm]		95,25
dfs [mm]		83,5487
db [mm]		83,5387
Lα [mm]		13,1
Lβ [%]		64,6
Backlash [mm]		0,1778
Degree of accuracy		4

Table 1,characteristics of the gears

The purpose of the study of such gears is different, in fact the NASA technical study had tested their gear noise system.

The gears described above, were one of three different samples that were analyzed and implemented the data on the DANST (Dynamic Analysis of Spur Gear Transmissions) software.

As regards what is sought in this thesis and similarly to what the GeDy TrAss software is able to do, the gears mounted on the test bench are used to search and evaluate how the static transmission error varies according to the input torque.

6 Code and set up

The following chapter illustrates the code that allows to use the data collected by the test bench measurement tools directly on MATLAB.

First of all, it is essential to shift the weight with the hoist, on the fixed support side, to give movement, ergo torque to the different pulleys and therefore to the gears.

First of all, thanks to an add-on you can interface instantly and autonomously to the torque meter through the Arduino board , which provides us voltages, which are then converted into torque values by means of the resolution of the instrument.

```
%% Record of the Torque
%%%Torque from the torquemeter
%a = arduino('COM3');
clear r Time Torque V var ts
close all
clc
Tot=10000;
f1=figure(1);
Time = zeros(1,Tot);
V = zeros(1,Tot);
var = 0;
Torque = zeros(1,Tot);


for r = 2:10000
          tic
          V(1,r) = readVoltage(a , 'A0');
          Torque(1,r) = V(1,r)*500/5;
          var = toc;
          Time(1,r)= Time(1,r-1) + var;
%           hold on,plot(Time,Torque,'b')
end
hold on,plot(Time,Torque,'b');
ylabel('Torque [Nm]');
xlabel('Time [s]');
title('Torque');
```

Assuming that the torque and the angular variations of the gears are collected with two different instruments, an important factor is the synchronization of the times, first of all, from the torque we have to eliminate the values that are not needed in order to have a range of data suitable for our purposes, case by case, basically it is useful to take as the starting point the one in which the constant signal begins to increase,

```
%% Torque data adjustment
TTy=Torque;
figure
plot(TTy);
Ctorque1=85;
Ctorque2=2314;
x1=Time(1,Ctorque1);
Timet=(Time-x1);
Timet(1:1:Ctorque1-1)=[];
Timet(Ctorque2-Ctorque1+2:1:end)=[];
TT=TTy(Ctorque1:Ctorque2);
f3=figure(3);
plot(Timet,TT);
ylabel('Torque [Nm]');
xlabel('Time [s]');
title('Torque');
```

For what concern the angular variations of the two encoders, the dedicated software of the parent company produces a .csv file as output, thanks to the fact that the software saves the necessary data in the same columns every time, through MATLAB it is possible to process the data from this worksheet in order to analytically calculate the value of the Static Transmission Error.

```
%% Take values from the encoder
% read from Excel angular variation and time
[displ]=xlsread('D:\POLI\EIB_74x\Tests\\.csv','E:E');
[Timeen]=xlsread('D:\POLI\EIB_74x\Tests\\.csv','K:K');
dg=displ(2:2:end);
p=displ(1:2:end);
timeen=Timeen(1:2:end)*10^-6;
```

At this point it is possible to obtain a raw STE pattern, after making the angular measurement continuous, avoiding a tooth chart, (which jumps from 360 ° to 0 °).

It is important to underline that the STE values must also be taken in the synchronization range between the encoder and the torque meter

```
%% Unwrap, make the reading of angular variations continuous
Qp = unwrap(p(:,1)*pi/180-p(1,1)*pi/180);
Qdg = unwrap(dg(:,1)*pi/180-dg(1,1)*pi/180);

%% calculation of STE
zp=28;
zdg=28;
STE = Qdg-(zp/zdg)*Qp;
f5=figure(5);
plot(STE*180/pi);
%%%remember to press brush button and save Cste from figure above
```

```
%% Time modification, remember to press brush button and save Cste from figure
above
x2=timeen(Cste(1,1),1);
tenc=(timeen-x2);
tenc(1:1:Cste(1,1)-1)=[];
tenc(Cste(2,1)-Cste(1,1)+2:1:end)=[];
STEn=STE(Cste(1,1):Cste(2,1));
f6=figure(6);
plot(tenc,STEn*180/pi);
xlabel('Time [s]');
ylabel('STE [°]');
title('Raw Static Transmission Error');
```

Another important step is to try to relate all the data collected, the torque meter collects a certain amount of data, much lower than what the encoder collects, due to different resolutions, but by interpolating it is possible to use the same base to be able to relate torque and transmission error.

```
%% interpolation
TTenc = interp1(Timet,TT,tenc);

%% plot of the STE and Torque
f7=figure(7);
yyaxis left
plot(tenc,STEn*180/pi);
ylabel('STE [°]');
xlabel('TIME [s]');
yyaxis right
plot(tenc,TTenc);
ylabel('Torque [Nm]');
title('STE-Torque');

%% overlap and synchronization
%%% take Csov1 e Csov2 from the plot
figure
plot(TTenc);%%% Csov1
figure
plot(STEn);%%% Csov2
dif=Csov2(1,1)-Csov1(1,1);
T2=TTenc(1:end-dif);
STE2=STEn(dif+1:end);
TIME2=tenc(1:end-dif);


f10=figure(10);
yyaxis left
plot(TIME2,STE2*180/pi);
ylabel('STE [°]');
xlabel('Time [s]');
yyaxis right
plot(TIME2,T2);
ylabel('Torque [Nm]');
```

```
xlabel('Time [s]');
title('STE-Torque');
```

In the raw STE mentioned earlier, the term raw indicates that the trend of the values obtained has not been minimally modified or filtered. During the numerous tests it was possible to notice how the signal obtained and processed is, in other words, dirty and altered.

Despite the changes made to the test bench, some problems have been found that disturb the output.

It has been realized that the pulley-strand structure could create problems and repercussions on the final values, which will make us think about a possible modification of the type of wire to be used.

As regards the modifications at gears level, it is possible to reduce the run-out phenomenon, that is the error due to the eccentricity of the gears, first of all, checking the eccentricity through a comparator, which gives as output a value that can be reduced gradually making several small adjustments on the on the screws and on the correct meshing of the gears themselves.

At this point of the set-up, the last trick to be done on the gears comes into play, that is to have the right interaxis calculated as the distance between the centers between the gears plus twice the external radius.

In detail, however, an attempt was made to correct and filter the signal in order to optimize it and get it as close to the truth as possible.

First of all, for each test, an ascent and a descent phase are obtained, for this reason the whole graph is divided into two parts treated differently and average torque on both phases is sought, this helps to free the raw STE signal from any oscillations due to the torque variation.

The next step to clean the signal is to work on the output of the no-load case and then subtract it from the aforementioned STE.

The no-load case arises when the whole structure and the gears were moved by hand without the use of weights, in order to capture only the STE due to the meshing of the gears.

The no-load case being moved by hand allows to obtain different rotations of the pinion and driven wheel, at this point the fulcrum is to find in the rotations of the no-load case the actual angular quantity carried out by the case under load, in this way the actual segment of the STE is obtained which will then be subtracted.

Moving forward in the code, it is necessary to subdivide graphically the STE into mesh cycles, producing an asterisk on the STE graph at each designated location.

```
%% Filtering of torque ( cut of variations due to the torque)
%%% take Valm1 for right side and Valm2 for left side

figure
plot(T2);%%% Ct cut in two part STE
STE2l=STE2(1:Ct(1,1));
STE2r=STE2(Ct(2,1):end);
T2l=T2(1:Ct(1,1));
T2r=T2(Ct(2,1):end);
figure
plot(T2r);
figure
plot(T2l);
Vmt=mean(T2r(Valmr(1,1):Valmr(2,1),1));
Vmta=mean(T2l(Valml(1,1):Valml(2,1),1));
Vmster=mean(STE2r(Valmr(1,1):Valmr(2,1),1));
Vmstel=mean(STE2l(Valml(1,1):Valml(2,1),1));
deltamr=Vmster/Vmt;
deltaml=Vmstel/Vmta;
STE_cr=STE2r-deltamr*T2r;
STE_cl=STE2l-deltaml*T2l;
figure
plot(STE_cr*180/pi); %%%take Cload
figure
plot(STE_cl*180/pi); %%%take Cload2

%% working on the pinion
%%% in order to find the corresponding angular
%%% variation on the no-load case
%%% searching values of p1 in pnoload (we find several applicants
%%% extremes and the most comfortable one must be chosen in order to then
%%%subtract the STE of the no-load)

p1=p(Cload(1,1)+Cste(1,1):Cload(2,1)+Cste(1,1));
p2=p(Cload2(1,1)+Cste(1,1):Cload2(2,1)+Cste(1,1));

%%% right part
[LIA,fLocB] = ismembertol(pnoload,p1(1,1),1e-4); %%% initial point of p1
LocB=find(fLocB);
figure
plot(pnoload)
hold on, plot([1,length(pnoload)],[p1(1,1),p1(1,1)],'r'),
plot(LocB,pnoload(LocB),'*');
figure
plot(NSTE);
hold on;
plot(LocB,NSTE(LocB),'*')
[LIA1,fLocB1] = ismembertol(pnoload,p1(end),1e-4); %%% final point of p1
LocB1=find(fLocB1);
figure
plot(pnoload)
hold on, plot(LocB1,pnoload(LocB1),'*');
figure
plot(NSTE);
```

```
hold on;
plot(LocB1,NSTE(LocB1),'*');
%%% both
figure
plot(pnoload)
hold on, plot(LocB,pnoload(LocB),'*'),plot(LocB1,pnoload(LocB1),'x');
figure
plot(NSTE);
hold on;
plot(LocB,NSTE(LocB),'*',LocB1,NSTE(LocB1),'x')


%%% left part


[LIA2,fLocB2] = ismembertol(pnoload,p2(1,1),1e-4); %%% final point of p2
LocB2=find(fLocB2);
figure
plot(pnoload)
hold on, plot([1,length(pnoload)],[p2(1,1),p2(1,1)],'r'),
plot(LocB2,pnoload(LocB2),'*');
figure
plot(NSTE);
hold on;
plot(LocB2,NSTE(LocB2),'*');
[LIA3,fLocB3] = ismembertol(pnoload,p2(end),1e-4); %%% initial point of p2
LocB3=find(fLocB3);
figure
plot(pnoload);
hold on, plot(LocB3,pnoload(LocB3),'*');
figure
plot(NSTE);
hold on;
plot(LocB3,NSTE(LocB3),'*');
%%% both
figure
plot(pnoload);
hold on, plot(LocB3,pnoload(LocB3),'*'),plot(LocB2,pnoload(LocB2),'x');
figure
plot(NSTE);
hold on;
plot(LocB3,NSTE(LocB3),'*',LocB2,NSTE(LocB2),'x') %%% remember to read from right
to left from first*


%% Cut of No-load
%%% values of  Locb and LocB1 change case by case from part above
%%% right
Xspace=linspace(0,1,LocB1(7)-LocB(3)+1);
Yspace=linspace(0,1,(Cload(2,1)-Cload(1,1)+1));
NLste = interp1(Xspace,NSTE(LocB(3):LocB1(7)),Yspace);
Ste_cr=STE_cr(Cload(1,1):Cload(2,1));
Ste_pcr=Ste_cr-NLste';
figure
```

```
plot(Ste_pcr*180/pi);


%%% left
Xlspace=linspace(0,1,LocB3(5)-LocB2(3)+1);
Ylspace=linspace(0,1,(Cload2(2,1)-Cload2(1,1)+1));
NLstel = interp1(Xlspace,NSTE(LocB2(3):LocB3(5)),Ylspace);
Ste_cl=STE_cl(Cload2(1,1):Cload2(2,1));
Ste_pcl=Ste_cl-NLstel';
figure
plot(Ste_pcl*180/pi);


%% mesh cycle


nloc=0;
for i=[0:19,26:28];
    tol=1e-5;
  mc(i+1)=i*360/zp;
  [LIA4,LocB4] = ismembertol(p,mc(i+1),tol);
 fLocB4=find(LocB4);
  if isempty(fLocB4)
      while isempty(fLocB4)
          tol=tol*1.1;
  [LIA4,LocB4] = ismembertol(p,mc(i+1),tol);
 fLocB4=find(LocB4);
      end
  end
  nloc(end+1:end+length(fLocB4))=fLocB4;
end


%%% right
figure
  plot(p);
  hold on;
  plot(nloc(2:end),p(nloc(2:end)),'*k');
  nloc1=nloc-Cste(1,1);
  nloc1(1)=[];
  nloc2r=nloc1(nloc1>Ct(2,1))-Ct(2,1);

figure %%% absolute mesh cycle angle increment
plot(nloc2r,Ste_pcr(nloc2r)*180/pi,'*k');
hold on;
plot(Ste_pcr*180/pi);


%%% left
  nloc3=nloc-Cste(1,1);
  nloc3(1)=[];
  nloc2s=nloc3(nloc3<Cload2(2,1))-Cload2(1,1);

figure %%% absolute mesh cycle angle increment
```

```
plot(nloc2s,Ste_pcl(nloc2s)*180/pi,'*k');
hold on;
plot(Ste_pcl*180/pi);


%% unique mesh cycle value
nr=uniquetol(nloc2r,0.01);
nl=uniquetol(nloc2s,0.01);
%% save


%%% right
STEn_case_Nm=Ste_pcr;
Vmt_case_=Vmt;
%%% left
STEn_case_Nm=Ste_pcl;
Vmta_case_=Vmta;
nr__=nr;
nl__=nl;
```

The last part of the code transforms the graphic division in mesh cycles into an actual division on the abscissa axis to obtain graphs in STE [°] Mesh cycle [-].

```
%% Creation of a space vector that will be the x axis in the graphs, already
divided into mesh cycles
S=STEn_case_Nm(nl(1):nl(end)-1);
n=nl-nl(1);
l=length(n);
x=[];
for z=[1:l-1]
    if z==1
        lg=(n(z+1)-n(z));
   spa=linspace(0,z,lg);
   x=[spa];
    else
        lg=(n(z+1)-n(z));
        spa=linspace(z-1,z,lg);
    x=[x,spa];
    end
end
figure
plot(x,S*180/pi);
set(gca, 'XTick', 0 : 1 : l-1);
```

7 Test

7.1 Test 1, preliminary proof

7.1.1 Output of the code and transformation of the STE

In this sub-chapter it will show how the code works and which types of outputs are produced.

For this chapter a specific test was taken as a sample, where at the fixed support side we had set 70 kg of weight and the mobile turret side 15 kg.

First of all, the torque trend is collected through the Arduino board, obtaining Figure 11, it is important to mention, at this point, the average torques of both sides are mentioned, 60 Nm on the left and 28Nm on the right.

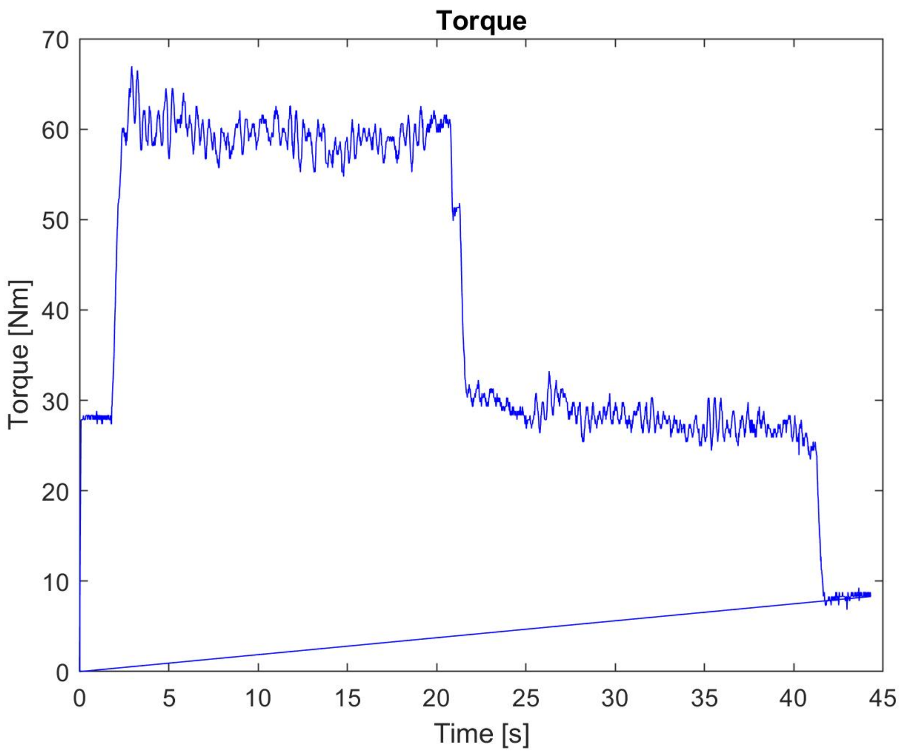

Figure 11, Torque

In the meanwhile, the sections of the torque that are not used or useful for the purpose are eliminated and cut as is shown in Figure 12.

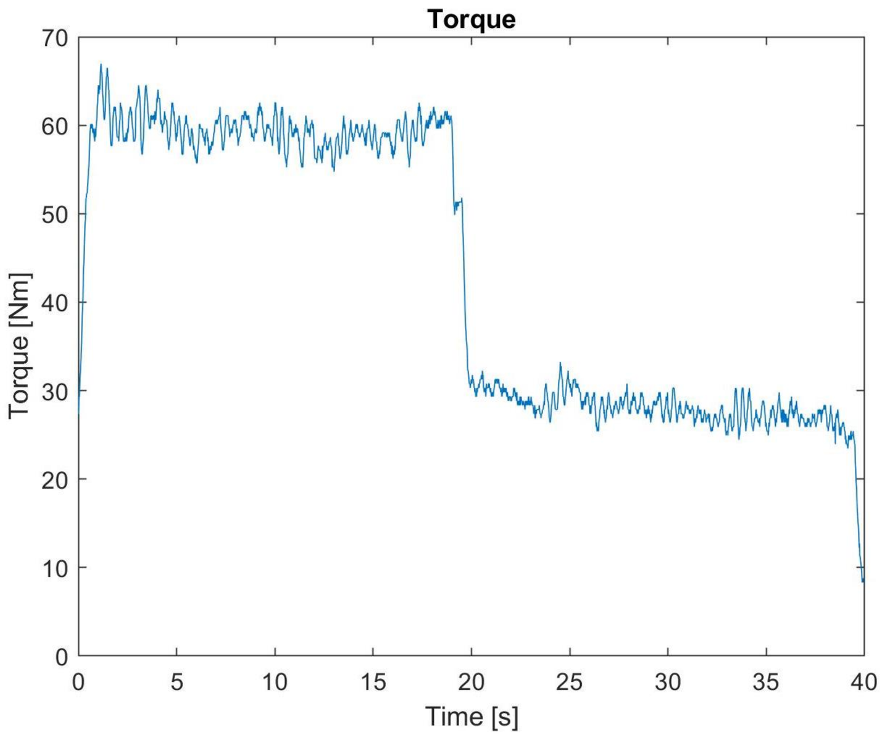

Figure 12, Torque used in the sample

After working on the torque, the encoder data must be collected and processed, in the top side of the Figure 13, it is possible to see the angular variation with a non-continuous trend, instead in the bottom side of Figure 13, it is possible to see a continuous trend, the bottom part of the figure allows us to easily understand how many revolutions our gears have made, naturally the driven gear and the pinion make the same number of revolutions having a unitary transmission ratio.

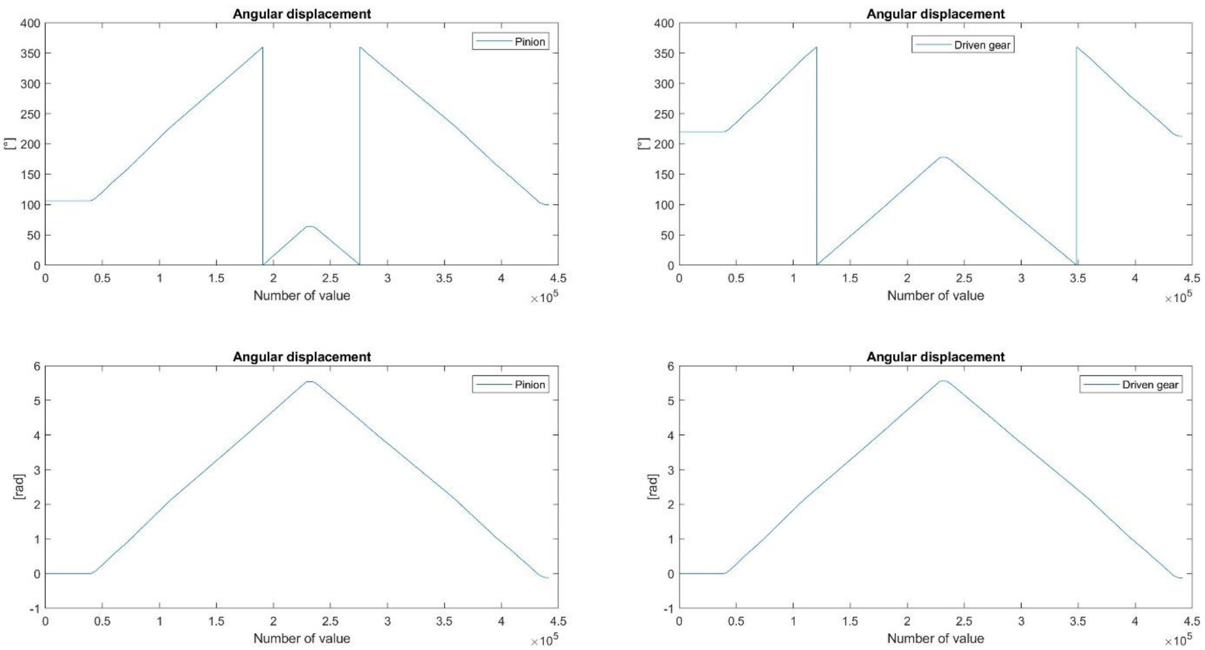

Figure 13,Pinion and Driven gear

Once the torque and angular variation have been collected, a raw STE can be calculated, after which it is advantageous to proceed step by step, starting from the elimination of the unused part of the signal, passing through the comparison between STE and torque, ending with the filtering of the variations due to the torque, thus obtaining an STE almost clean of any interference.

Figure 14, shows the raw STE,

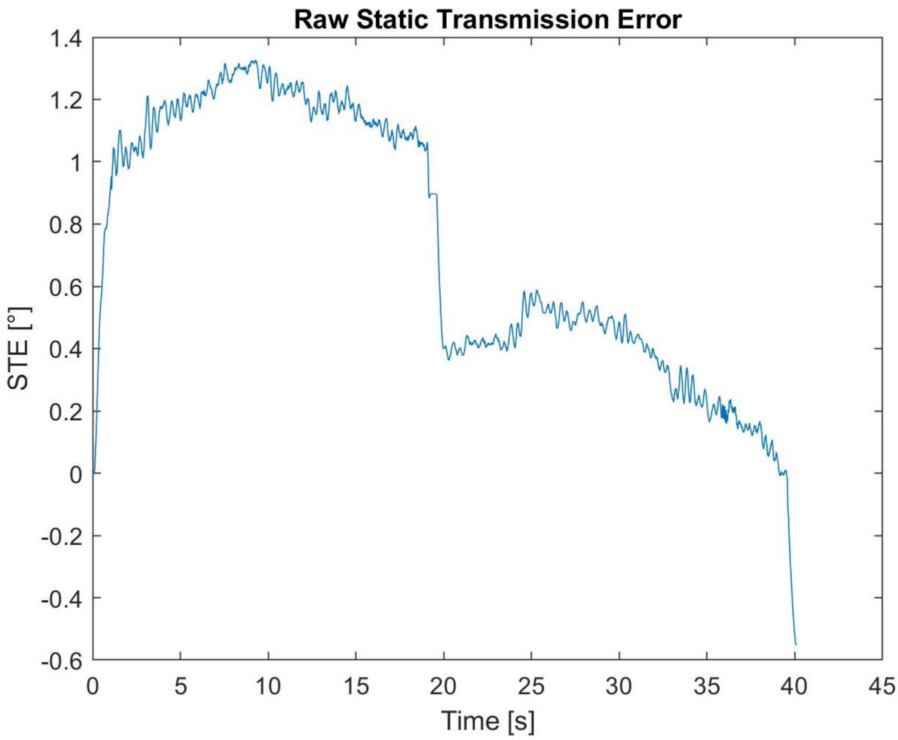

Figure 14, Preliminary STE

instead, Figure 15, shows the comparison

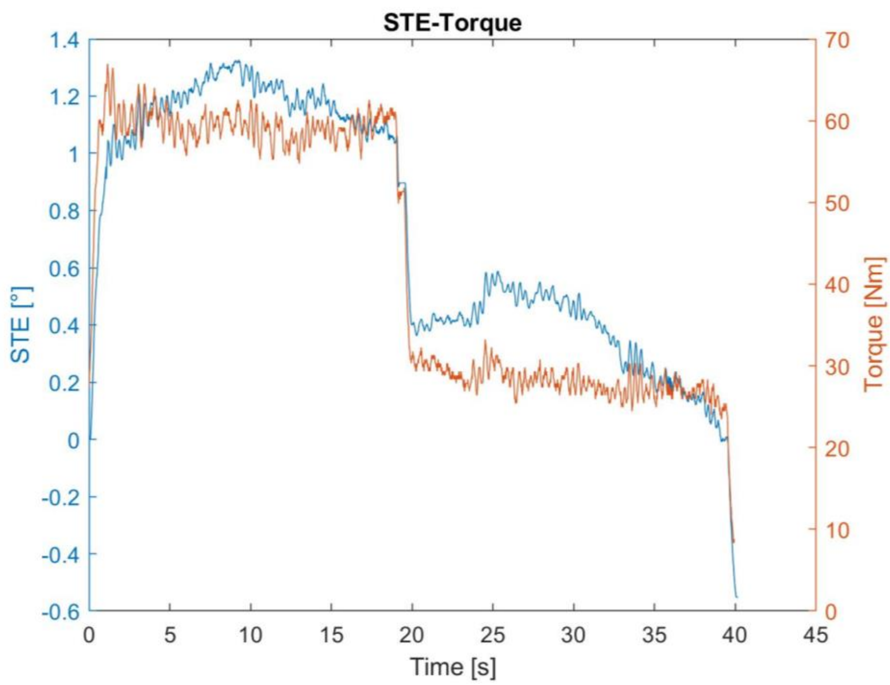

Figure 15, comparison between STE and Torque and as last step, Figure 16, shows an almost completely filtered STE.

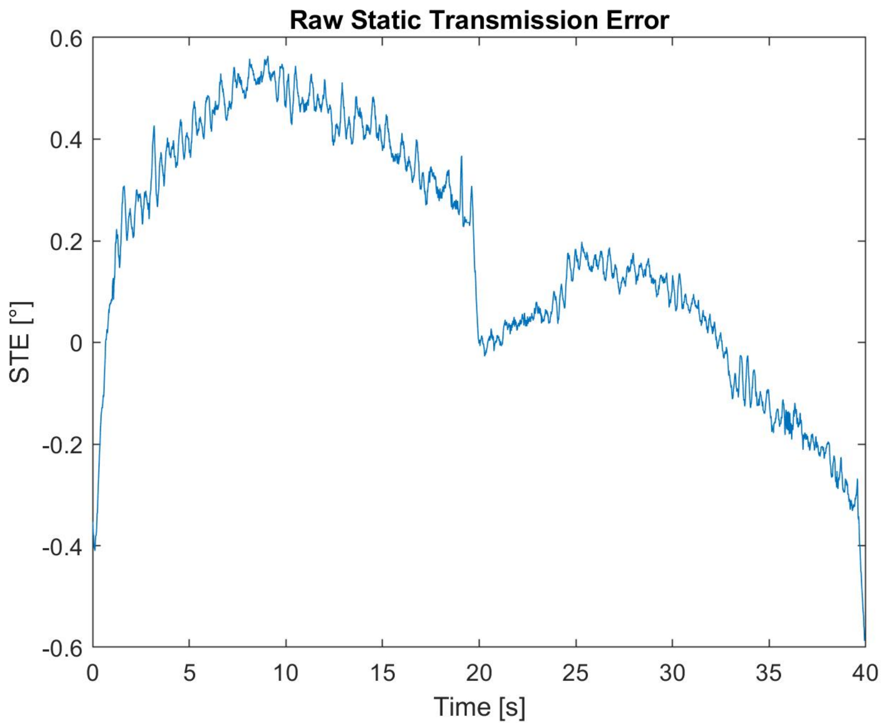

Figure 16, STE after cut of torque variation.

Looking at the torque trend, it can be seen that it has two main parts, the left side with higher torque and the right side with lower torque values, due to the fact that the left side corresponds to the time when the hoist accompanies the weight to descend, while the part on the right is obtained when the hoist helps the weight to rise.

For this reason the two parts are treated separately and it is possible to filter the effect given by the same gears but in the case of no-load.

For this purpose, Figure 17, shows the trend of the STE in the case of no-load, which it will be subtracted in both left and right STE, as it is possible to see in the graph there are several waves, this trend is due to the fact that, as described in the previous chapter, it has been collected many more turns of the gears, given that it is taken by hand.

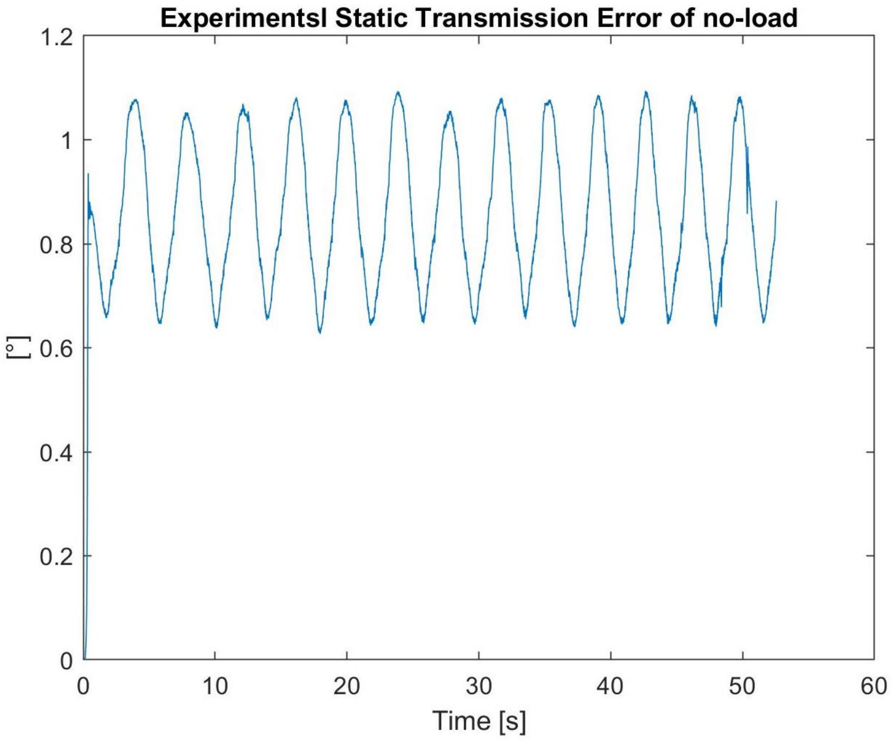

Figure 17, STE no-load case

Figure 18, will show the final trend of STE of both sides.

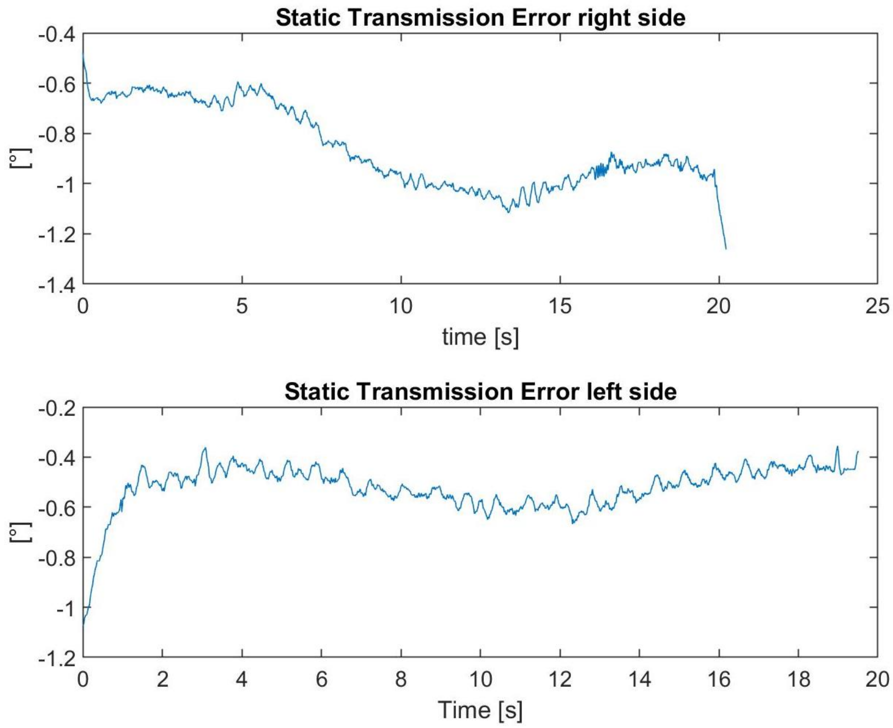

Figure 18, Right and left side of the sample STE

As for the Figure 19, it takes up the previous graph, but pointers have been added that will help to divide the entire graph into mesh cycles.

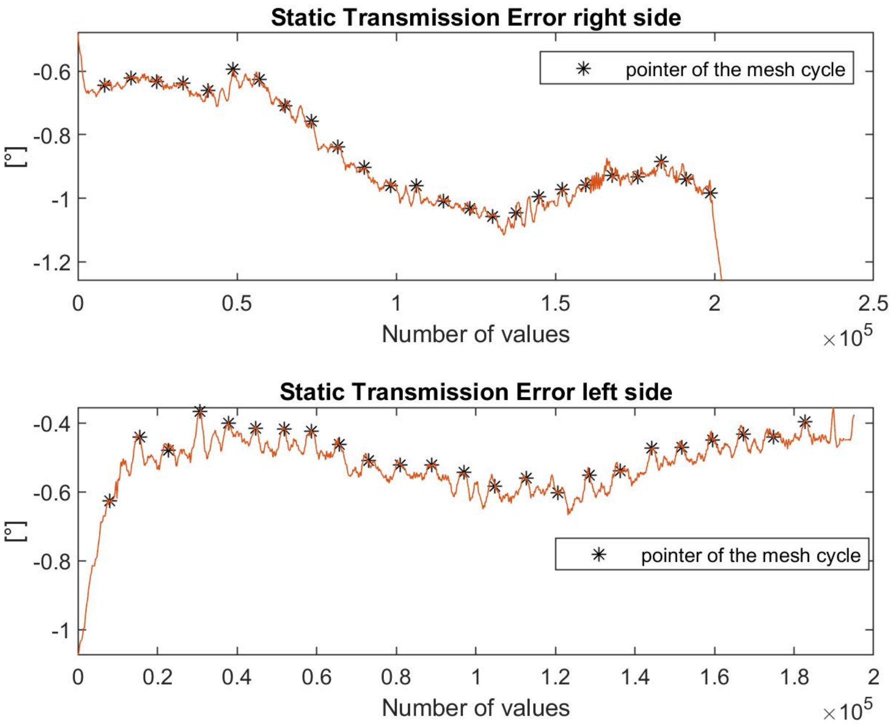

Figure 19, pointer of the mesh cycles

7.1.2 Comparison with literature and Software

7.1.2.1 Software and Test bench

Thanks to the previously developed code, it is possible to proceed with the first comparison.

In this case, in a preliminary manner, the STE at three different levels of torque was researched on the sample gears.

The test in question was a first attempt to observe the phenomenon, in fact a small number of proofs and therefore data was collected, in fact, for this preliminary comparison, the torque 13, 28, 35, 60, 110 Nm have been chosen as examples, a very important detail not to be overlooked is the fact that these gears can withstand a maximum of 100 Nm of torque, practically a load equal to 50 kg, taking into account the size of our pulley, before the teeth are affected by this stress and can, consequently, break.

The choice to also show the case with 110 Nm serves to illustrate how the value of PPTE is significantly higher than that of the other cases, when the limit of endurance is exceeded.

First of all it will be shown a comparison between the different values of STE in relation to different torque values, in Figure 20 and Figure 21.

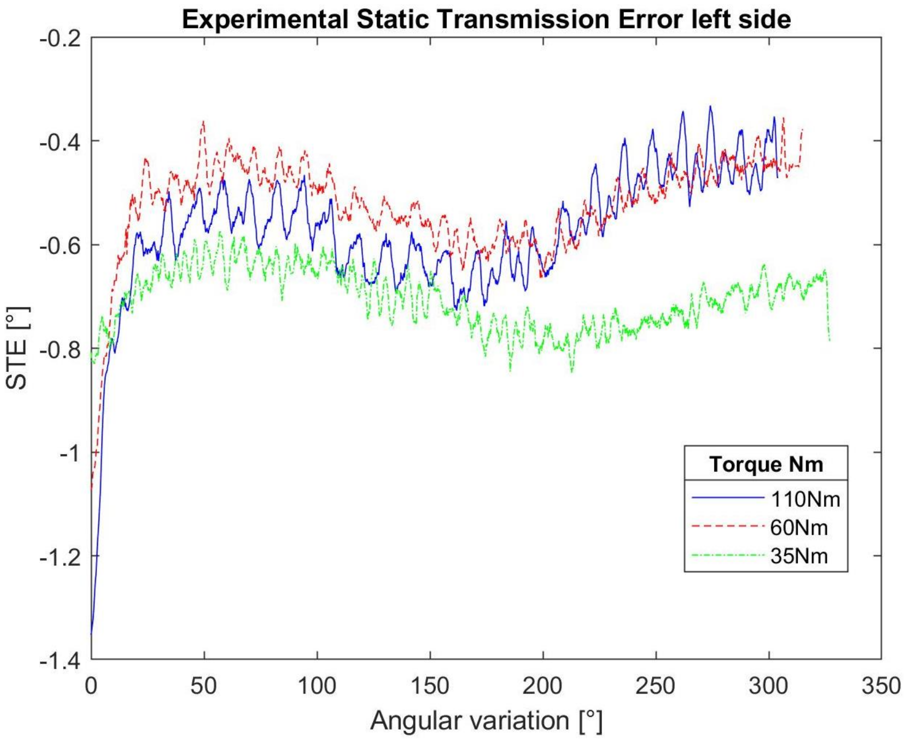

Figure 20, left side of experimental STE

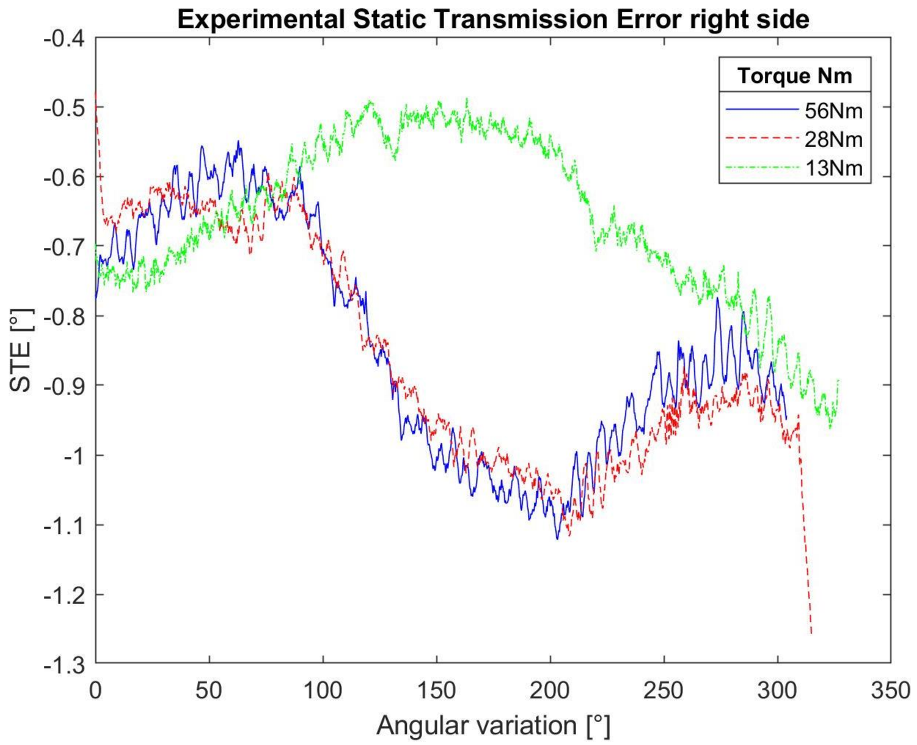

Figure 21, right side of experimental STE

This test is compared, first of all with the outputs produced by the Software GeDy TrAss .

The different comparisons are then shown,

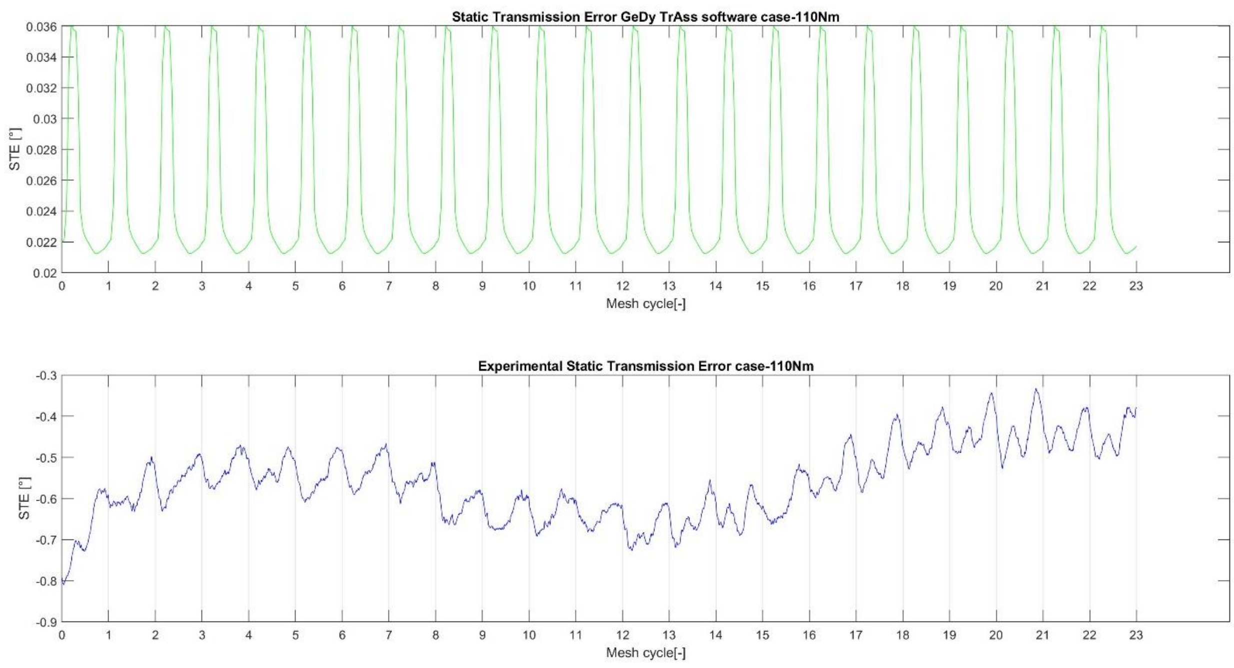

Figure 22, Comparison at 110 Nm

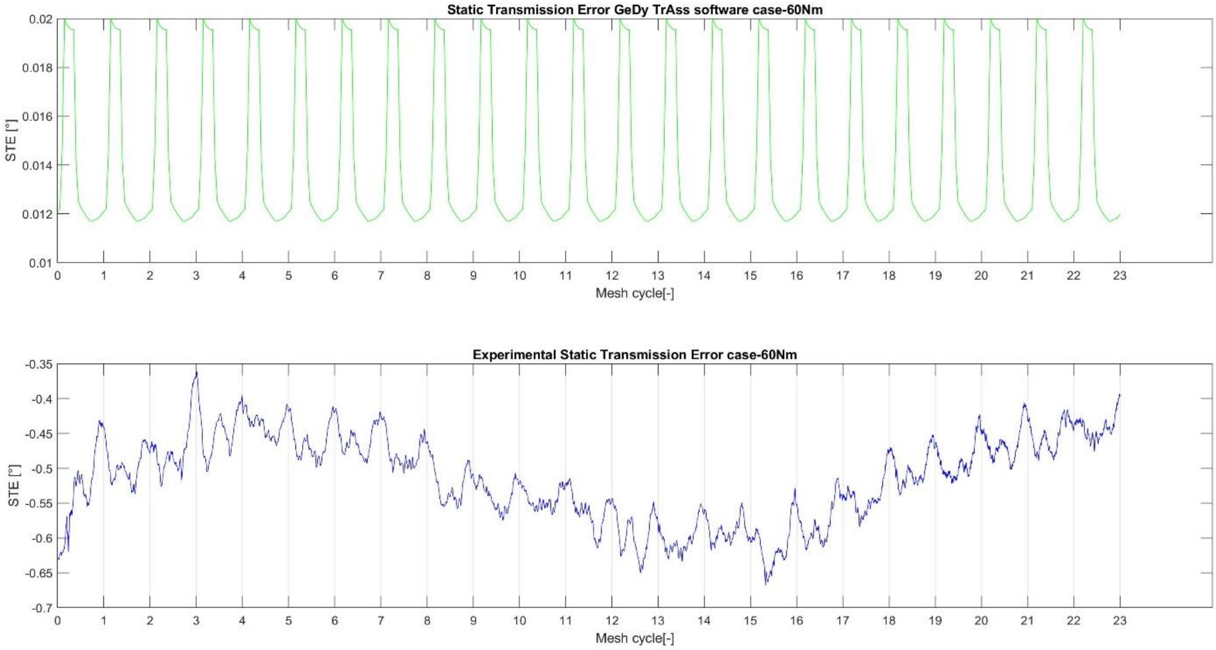

Figure 23, Comparison at 60 Nm

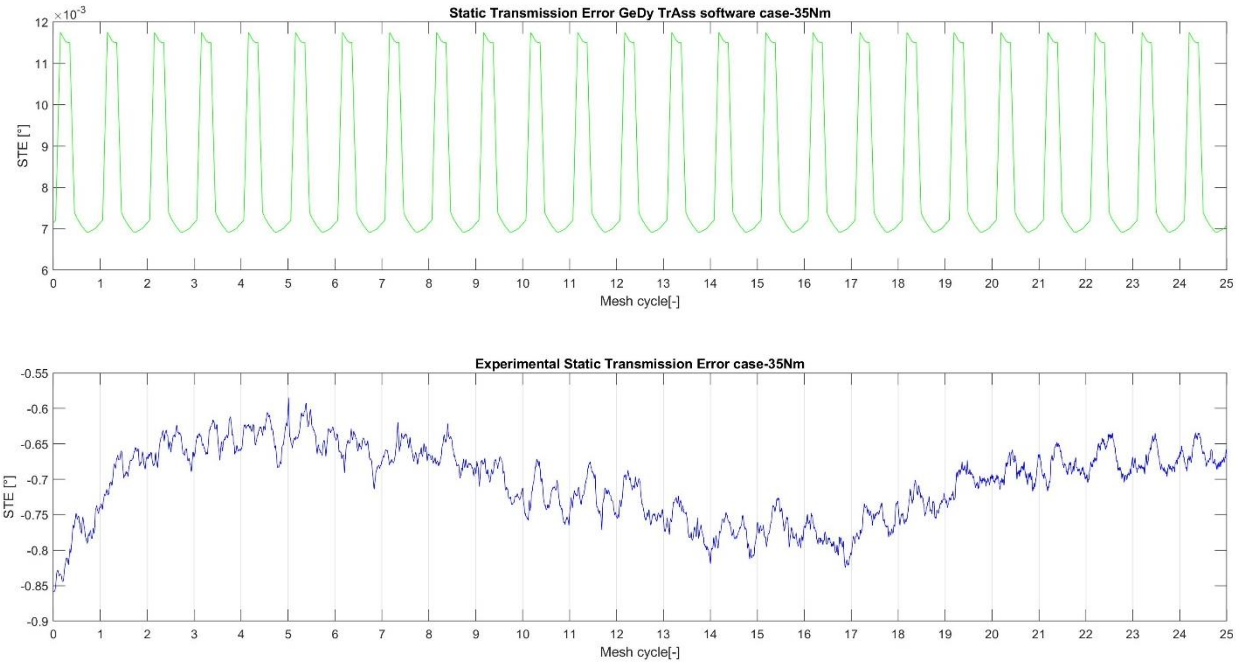

Figure 24, Comparison at 35 Nm

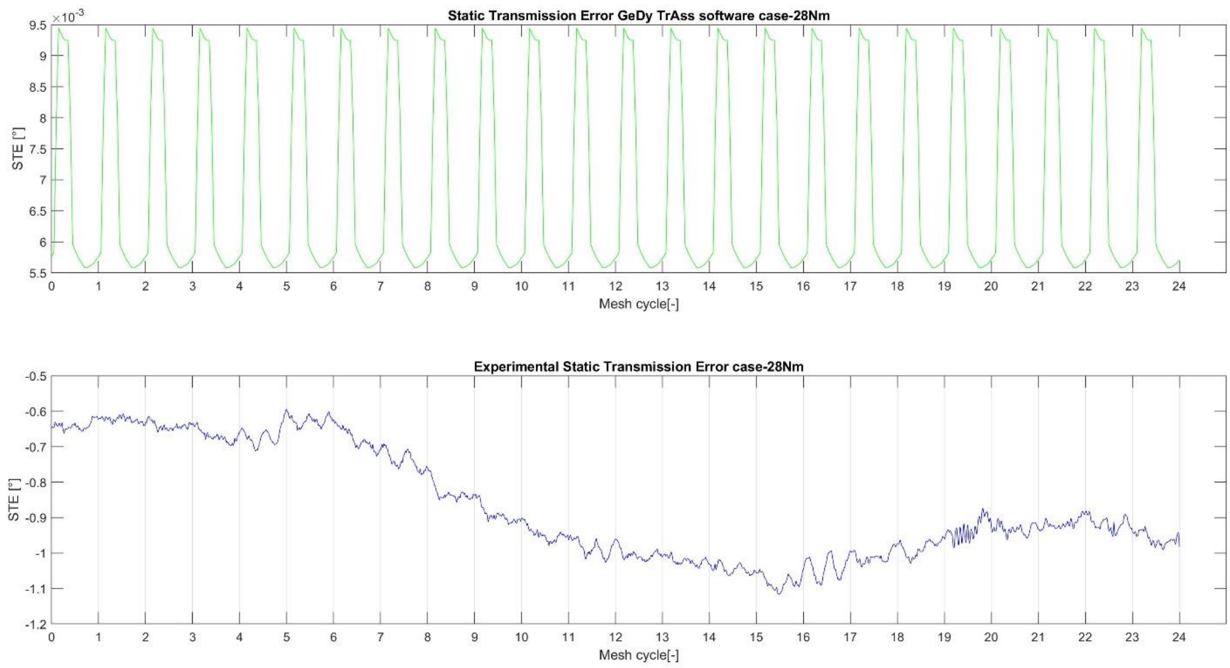

Figure 25, Comparison at 28 Nm

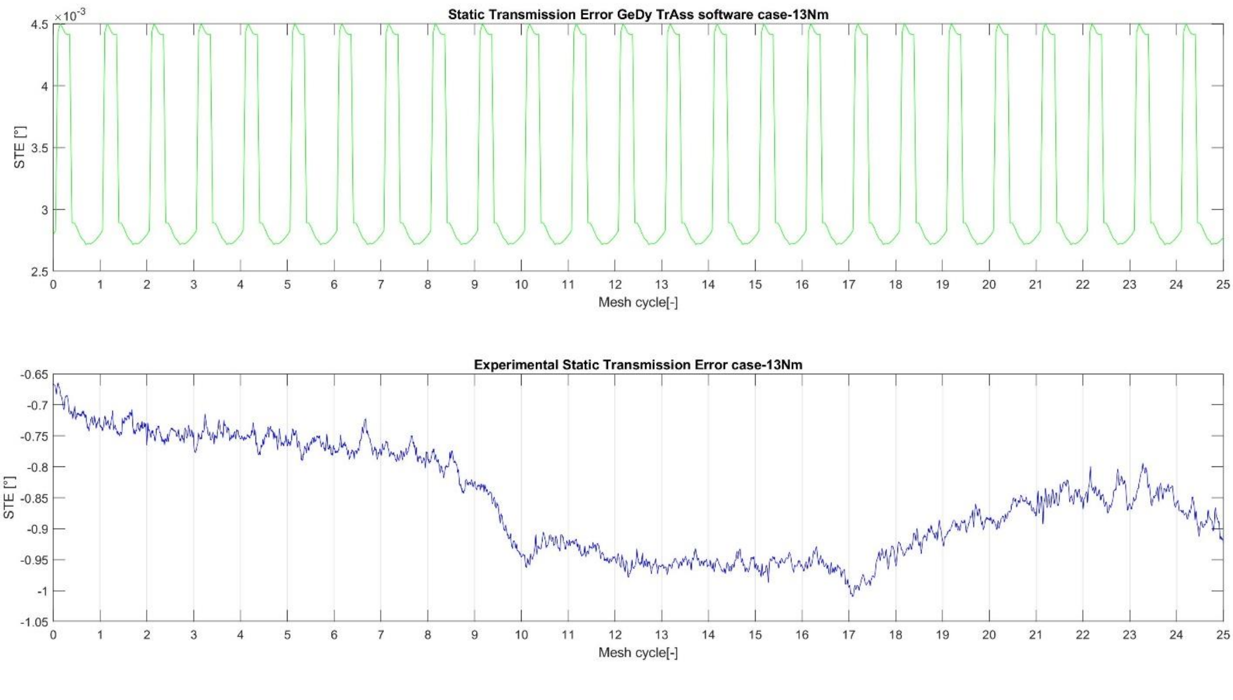

Figure 26, Comparison at 13 Nm

7.1.2.2 Software, Test bench and Literature

After having seen the comparison graphs between STE obtained from the test bench with those of the GeDy TrAss software, it is convenient to compare the results also with those relating to the NASA literature, in this case we go to compare the PPTE values in order to understand if what we obtain has a true trend and can be taken into consideration.

A small note is reported, in fact as can be seen from the NASA literature the tests were carried out at 11, 33, 56, 102 Nm of torque, in practice values slightly lower than those found with the test bench.

To create the comparison, the value obtained experimentally was kept as a sample because having few values the difference gap is difficult to bridge since our set-up guarantees us values that are quite close to literature values, but not reachable

The same is not true for the GeDy TrAss Software instead, it guarantees outputs at any value.

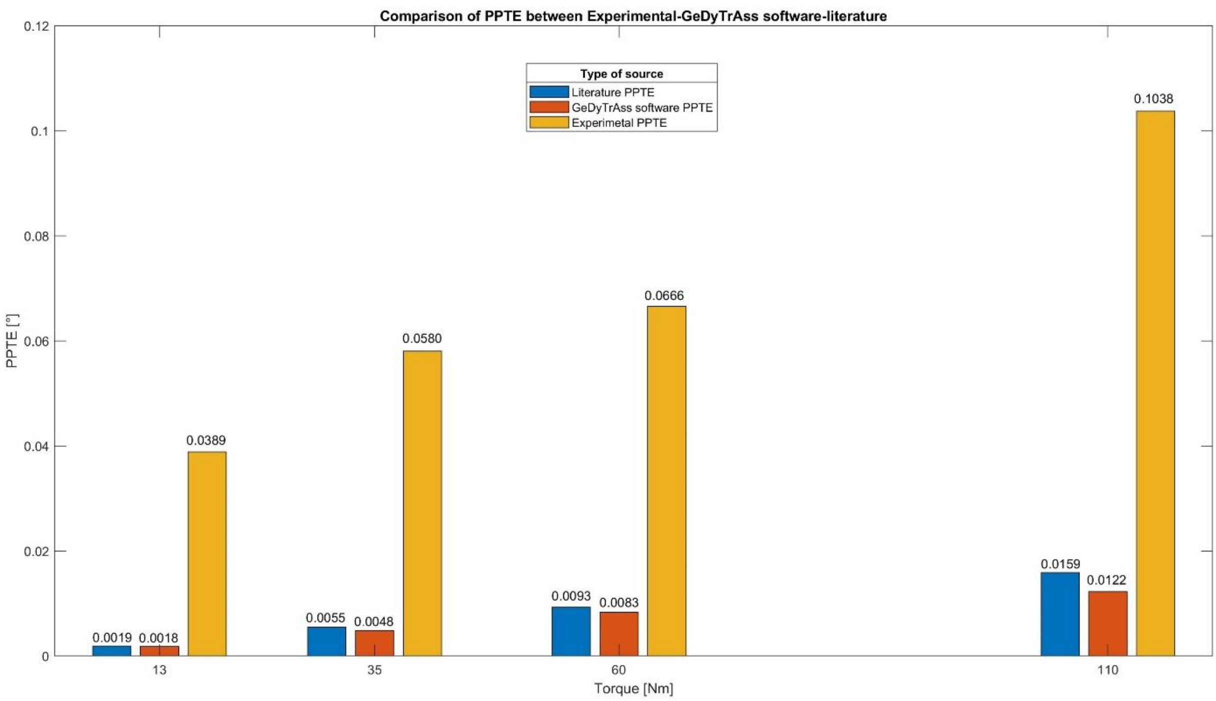

Figure 27, Comparison between PPTE

It can be seen from Figure 27 that the values obtained from the test bench is markedly different from the others, we are talking about a value of PPTE one order of magnitude greater than what it is compared with.

As a counter-proof, the PPTE values obtained have been interpolated with a linear trend line in order to highlight the trend of the points, it is evident that in the case of the test bench there is an evident offset compared to the software and literature cases, the points in fact seem follow a trend that is not even linear.

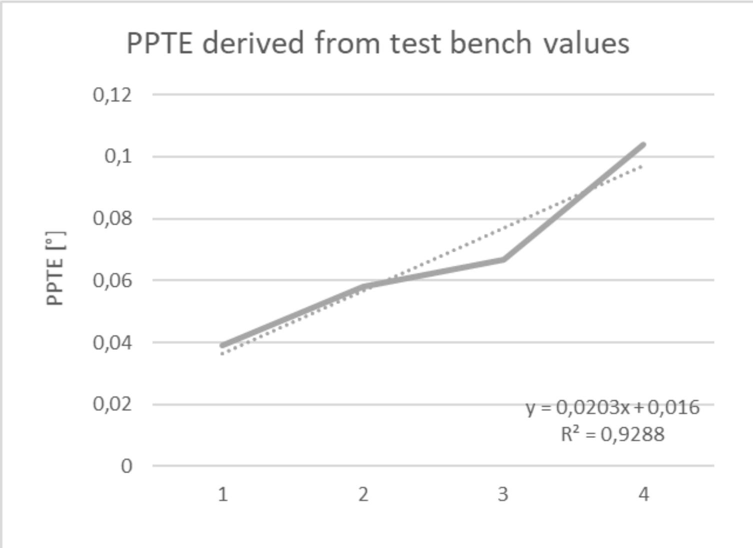

Figure 28, PPTE test bench

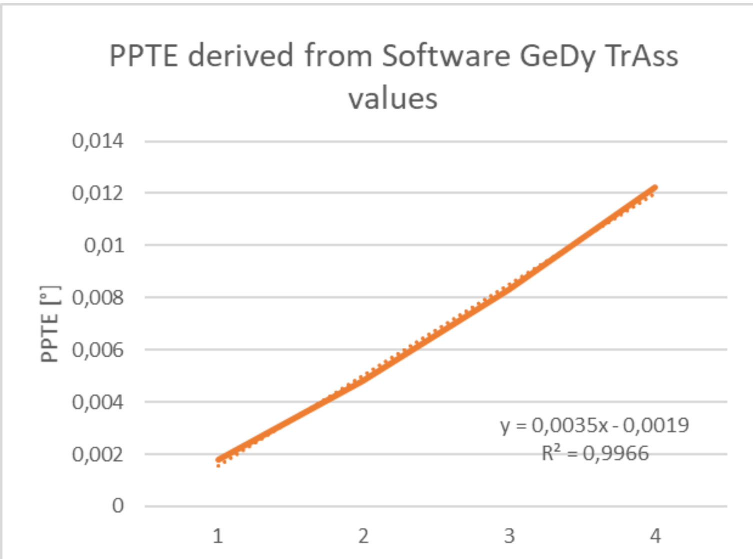

Figure 29, PPTE GeDy TrAss

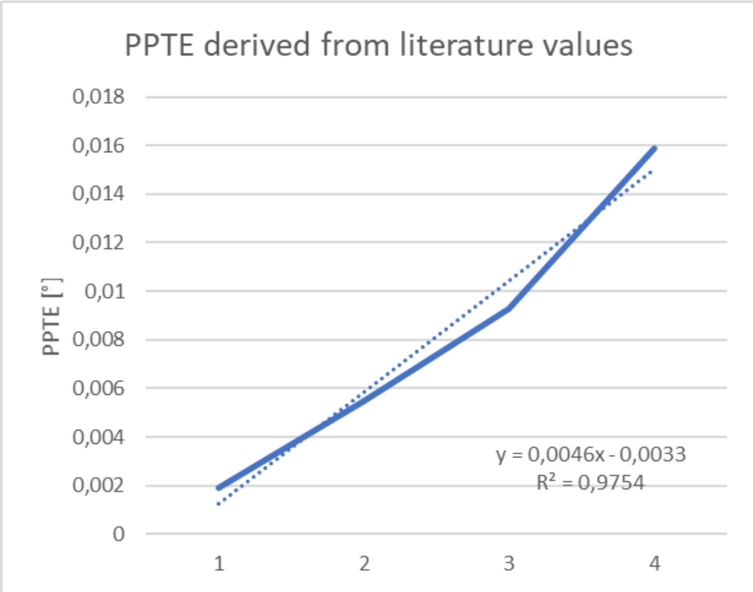

Figure 30, PPTE literature

Figure 28, 29, 30 highlight the three different patterns of the cases compared.

This discrepancy has forced us to take into consideration the fact that our measurements are offset by some component of the test bench, despite the proper fixing of the gears and the filtering implemented by means of the code itself, the output needs to be improved.

Our attention went back to the test bench set up, we realized that the quill drive, that is the decoupling joint, is the only component in play that can creates problems, due to its compliance.

For this reason, we have proceeded to modify the component in order to then carry out new tests and make sure of any improvement.

7.2 Test 2

7.2.1 Test bench modifications

From the point of view of the setup it was decided to carry out the entire test with the magnets on, which better anchor the mobile support of the test bench to the ground, and as anticipated in the previous sub-chapter, it was decided to modify the joint between the gear and the encoder, called quill drive.

The internal arms system was removed, replacing it with bushings to obtain greater rigidity and therefore eliminate internal compliance, but at the same time the new structure must decouple shaft and encoder, ensuring that gear and encoder rotate together, thus continuing to evaluate the angular variation.

In Figure 31 it possible to see the new joint.

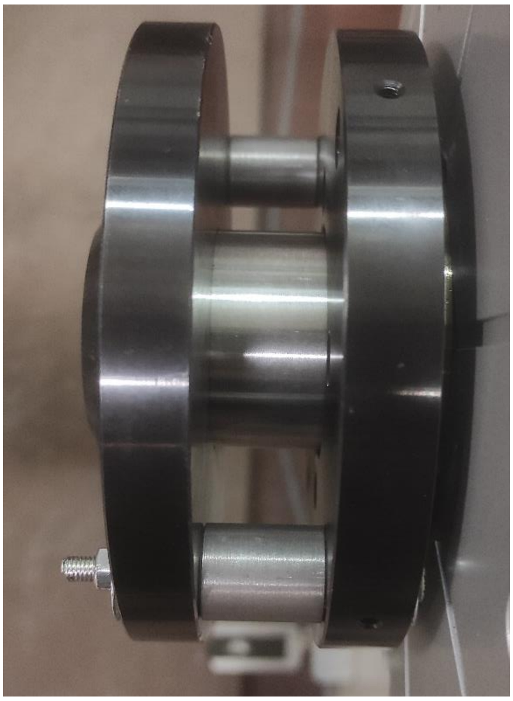

Figure 31, Coupling that replaces the quill drive

Before going into the description of the outputs of Test 2, it should be emphasized that in this case, unlike the previous test, we tried to populate more the data collection, carrying out more and more proofs, always changing the input torques.

Compared to Test 1, after the modifications the trend of the no-load case was again obtained, this time by making the gears run fewer laps, but for the purpose of post-processing the number of laps made has no relevance.

In Figure 32 is possible to see the trend of the no load.

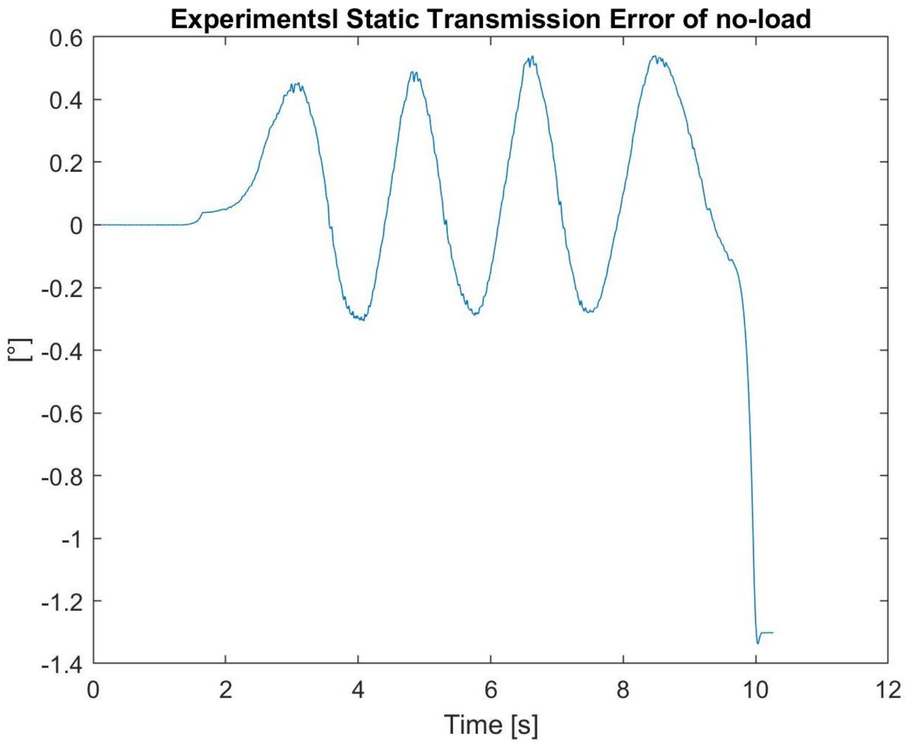

Figure 32, no-load trend

7.2.2 Comparison with Software GeDy TrAss and Literature

7.2.2.1 Software and Test bench

The steps for producing the outputs are the same as in the previous chapter described, so in this sub-chapter, first of all, we move directly to the comparison between the STEs obtained from the test bench with those extrapolated from the GeDy TrAss code, from the smallest to the highest torque, which in this test is 122 Nm, a number considerably higher than the 100 Nm limit, used only to highlight the net increase in the value of STE and therefore also of PPTE.

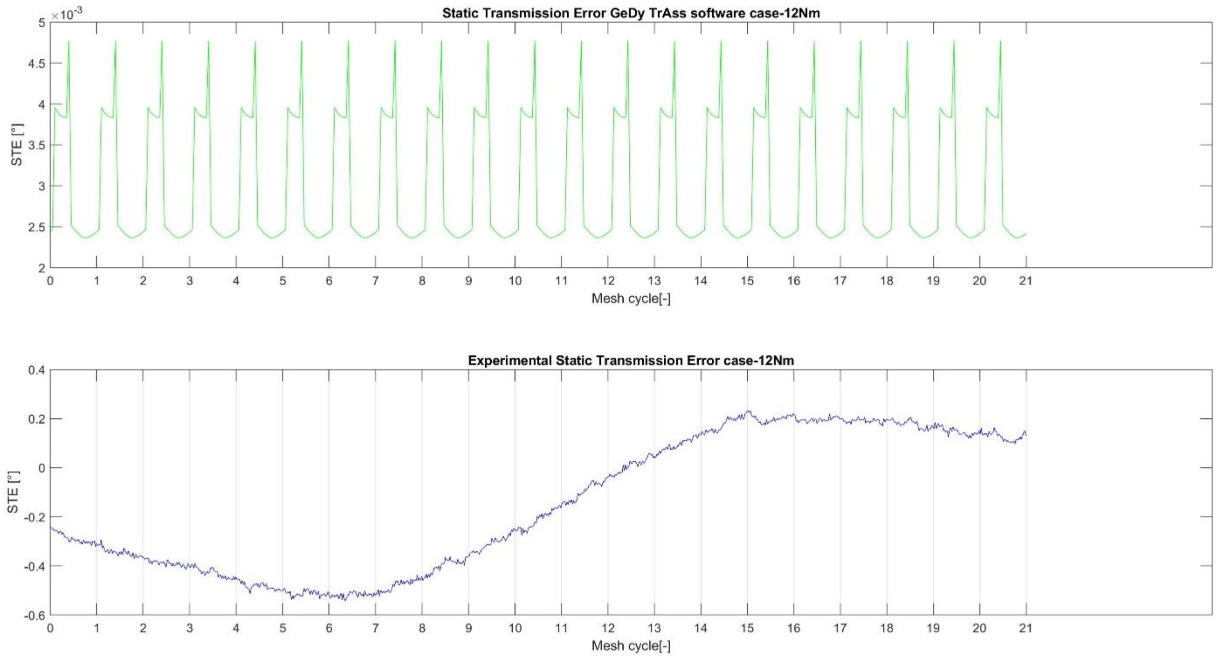

Figure 33, Comparison at 12 Nm

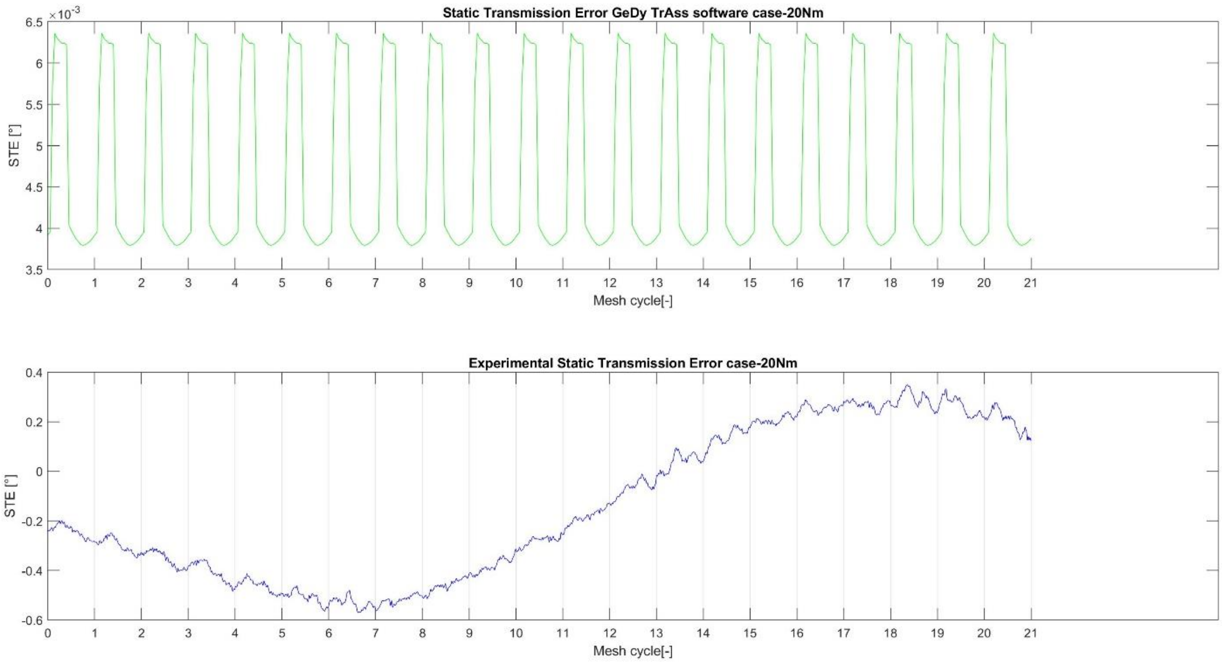

Figure 34,Comparison at 20 Nm

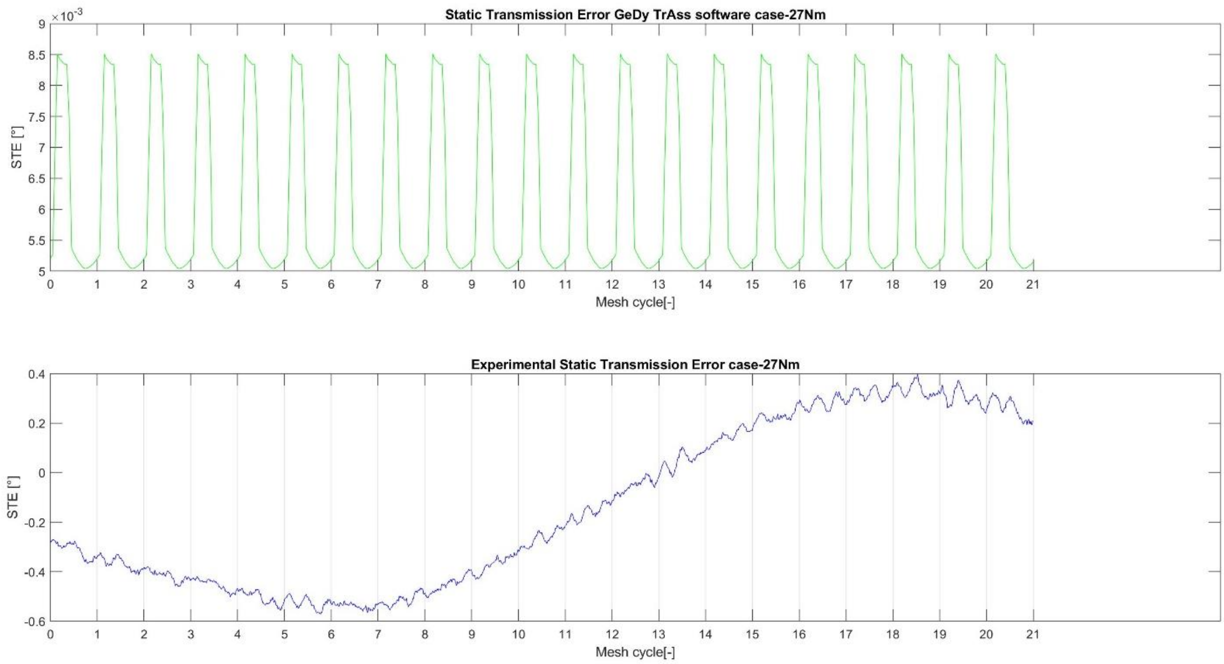

Figure 35,Comparison at 27 Nm

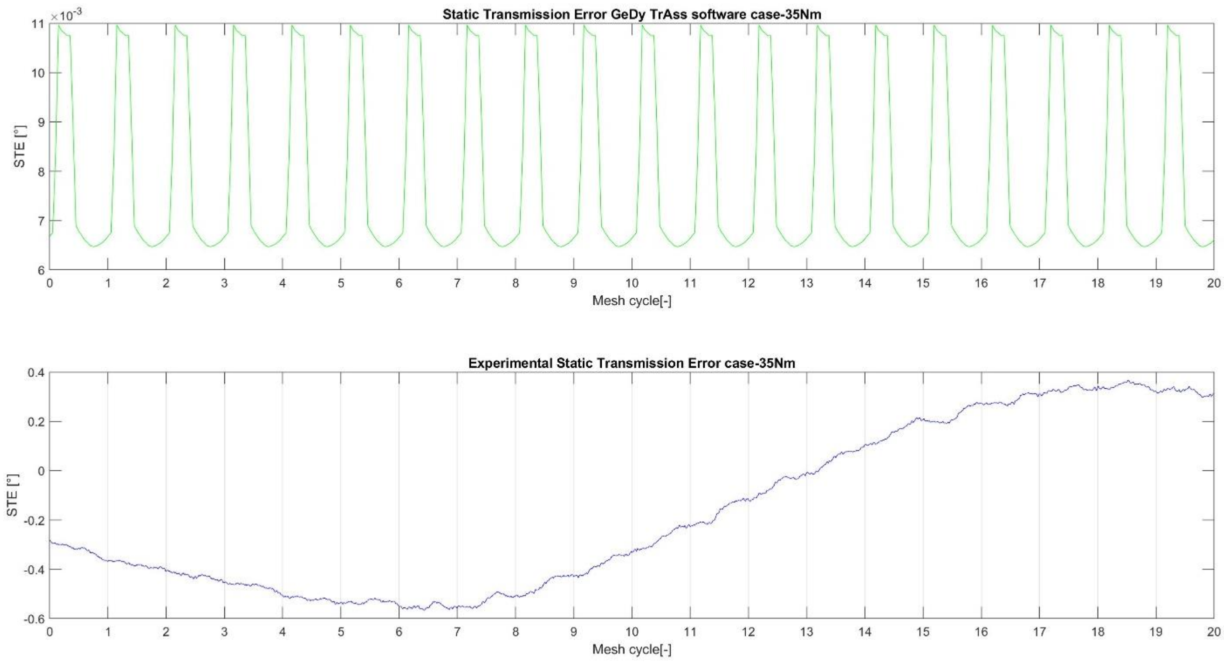

Figure 36,Comparison at 35 Nm

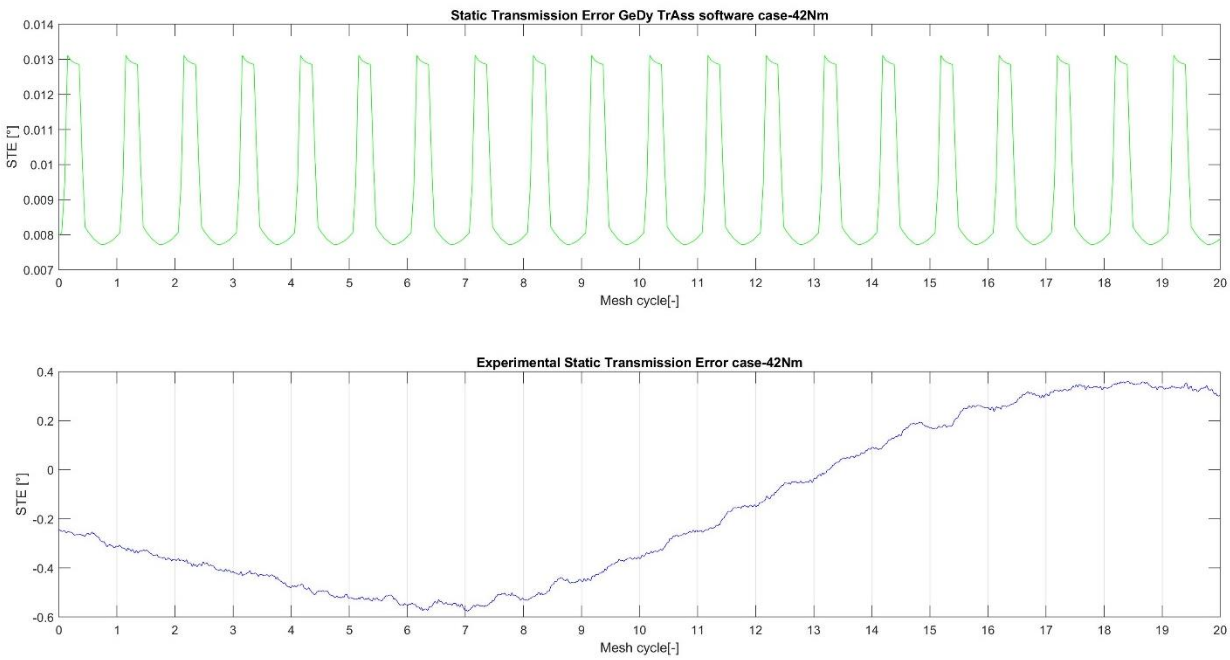

Figure 37,Comparison at 42 Nm

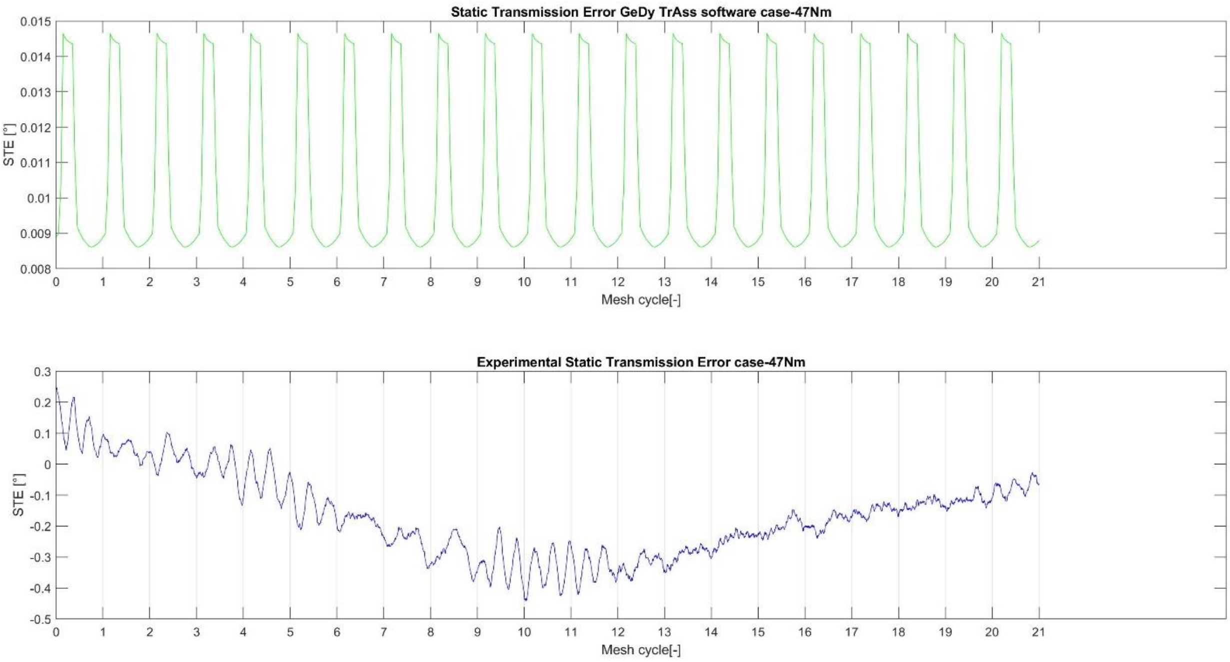

Figure 38,Comparison at 47 Nm

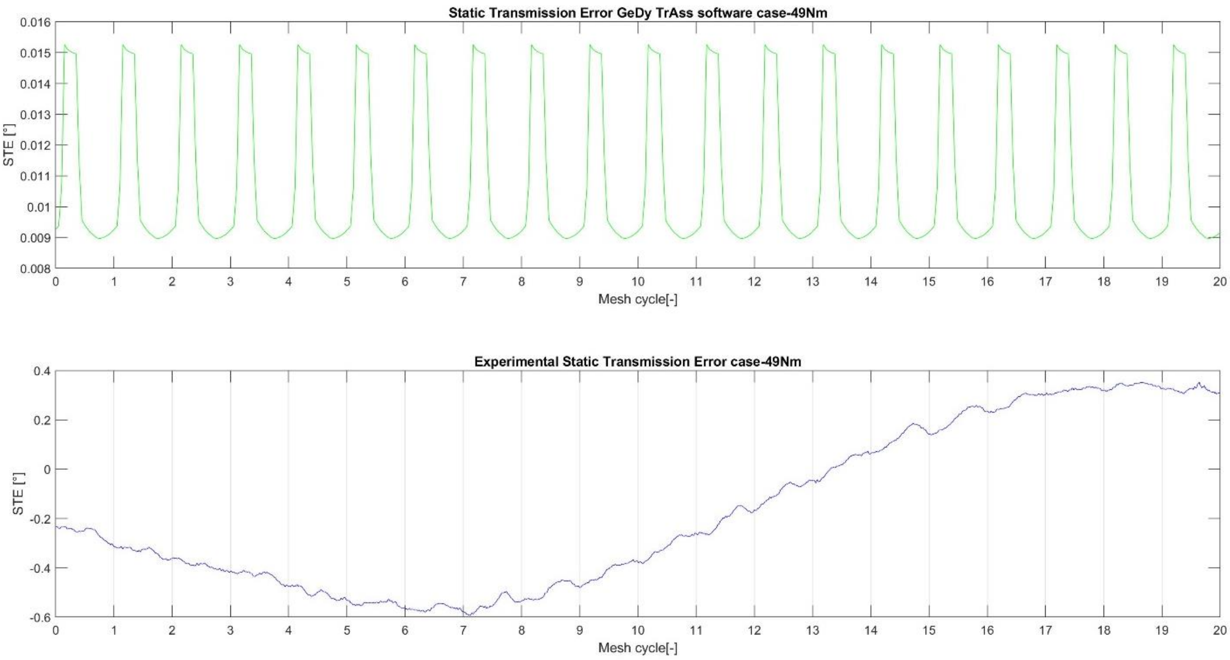

Figure 39,Comparison at 49 Nm

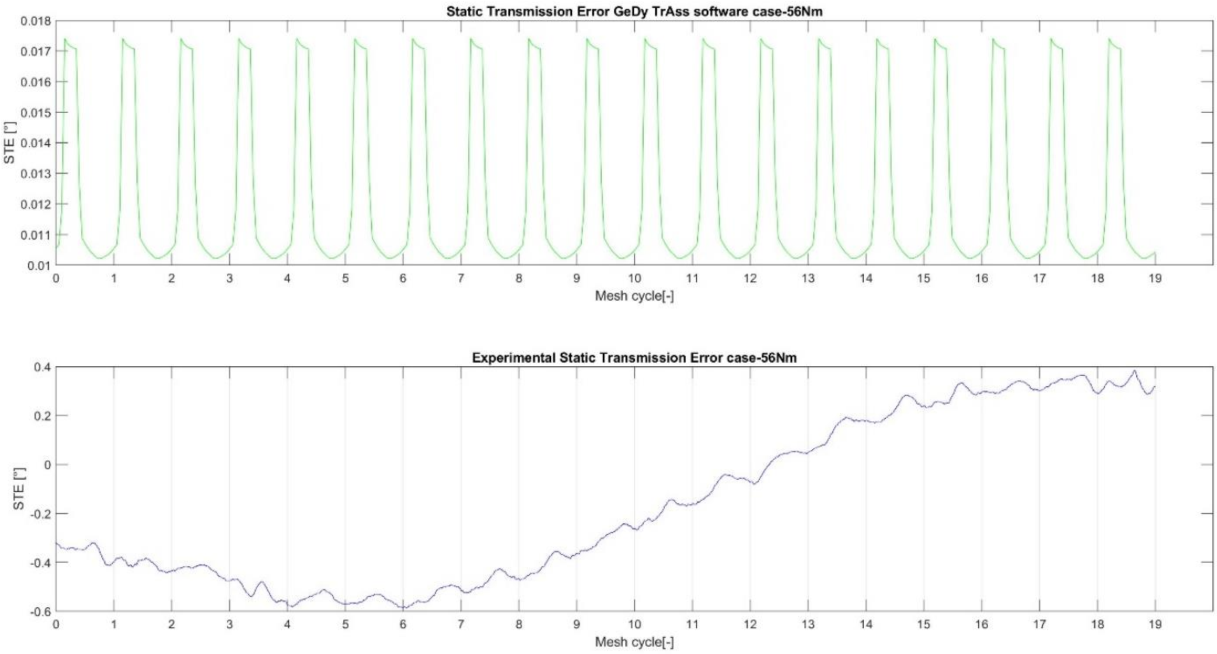

Figure 40,Comparison at 56 Nm

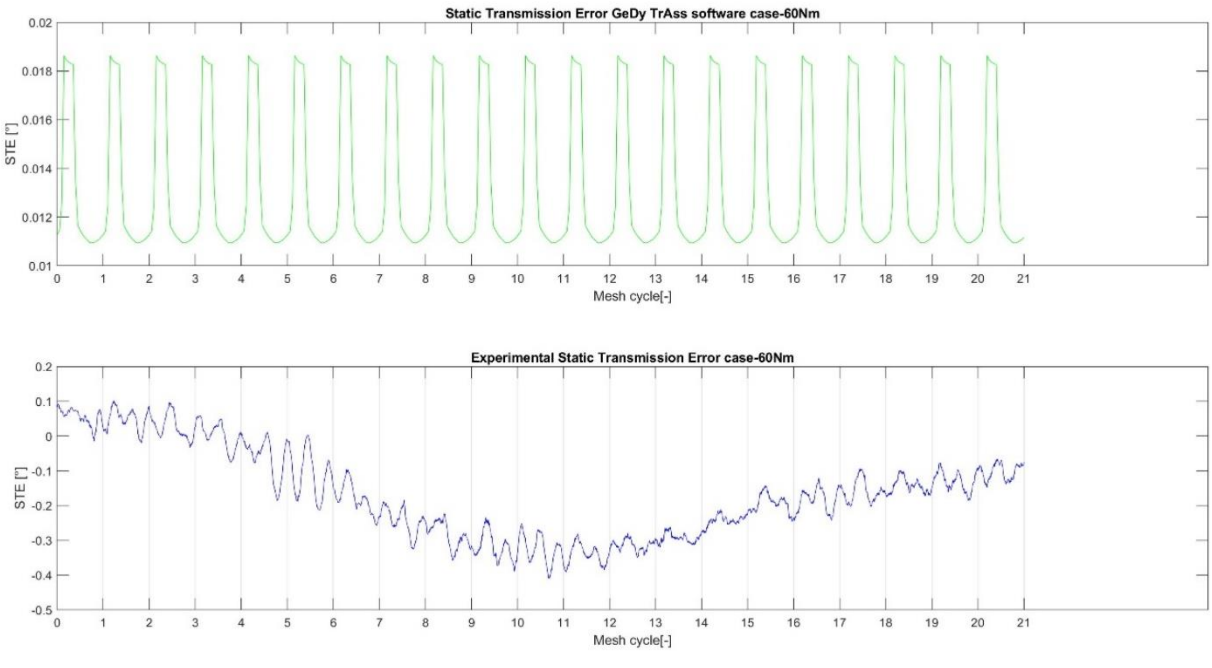

Figure 41,Comparison at 60Nm

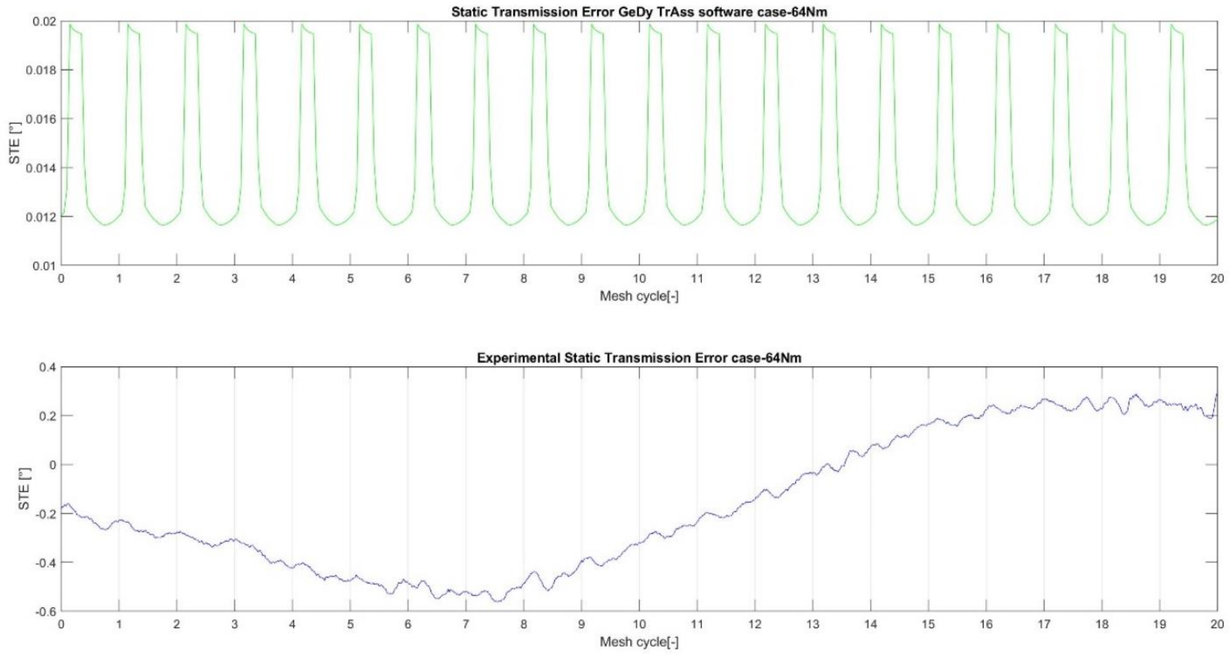

Figure 42,Comparison at 64 Nm

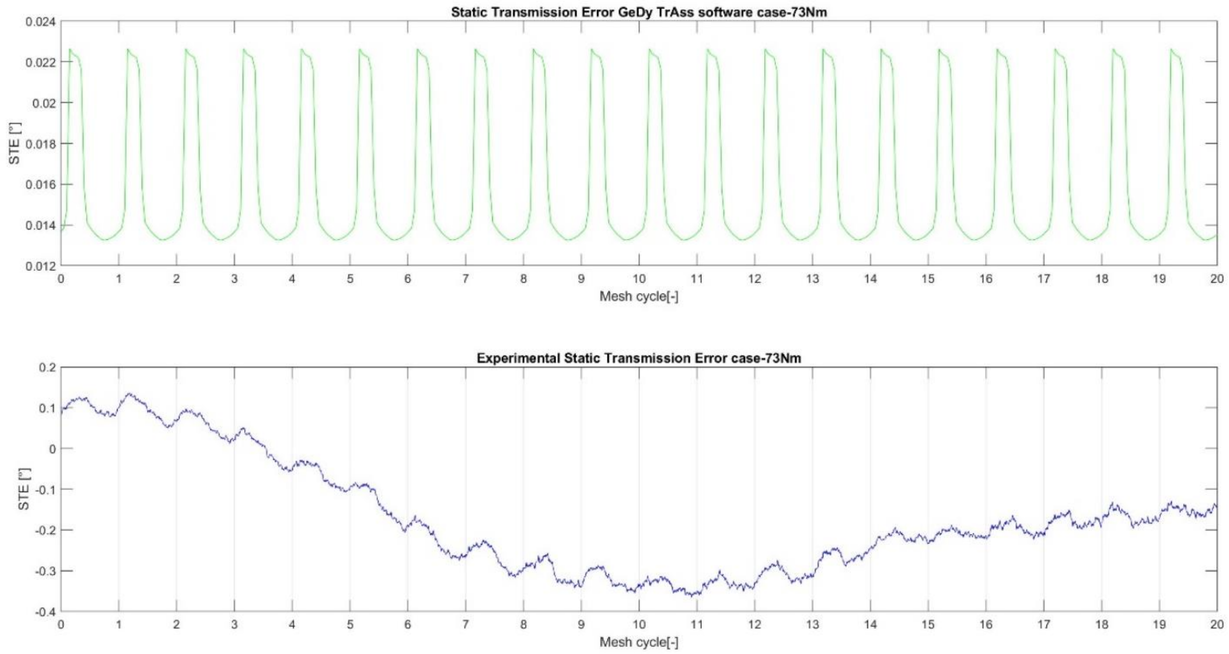

Figure 43,Comparison at 73 Nm

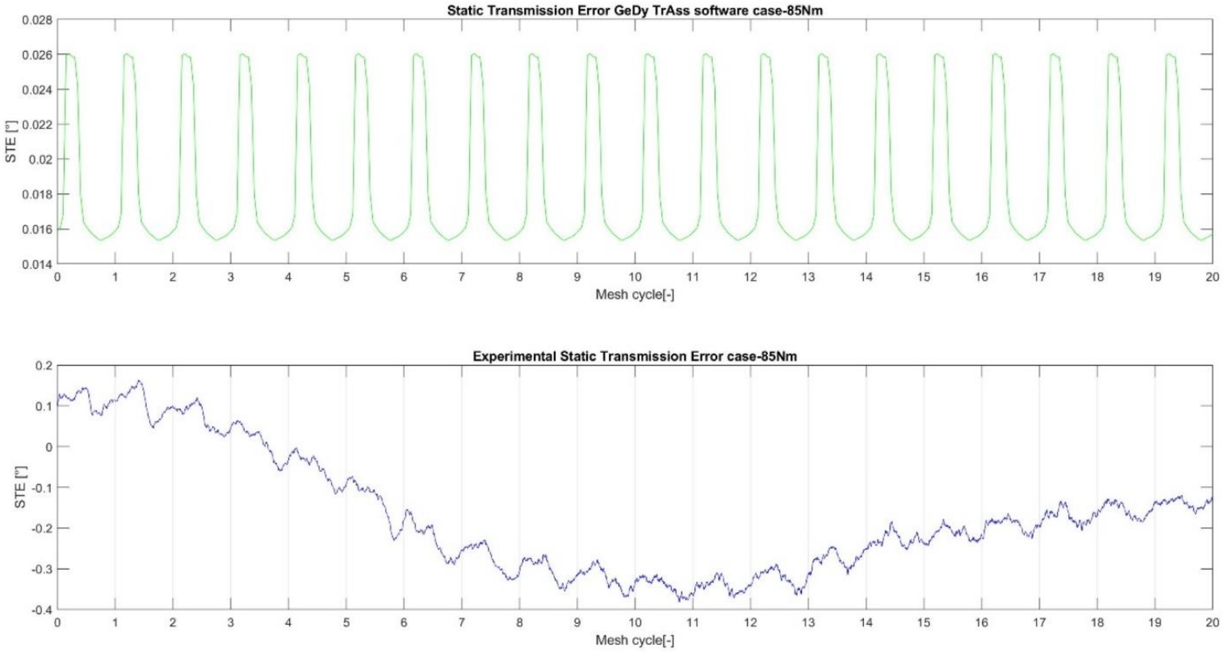

Figure 44,Comparison at 85 Nm

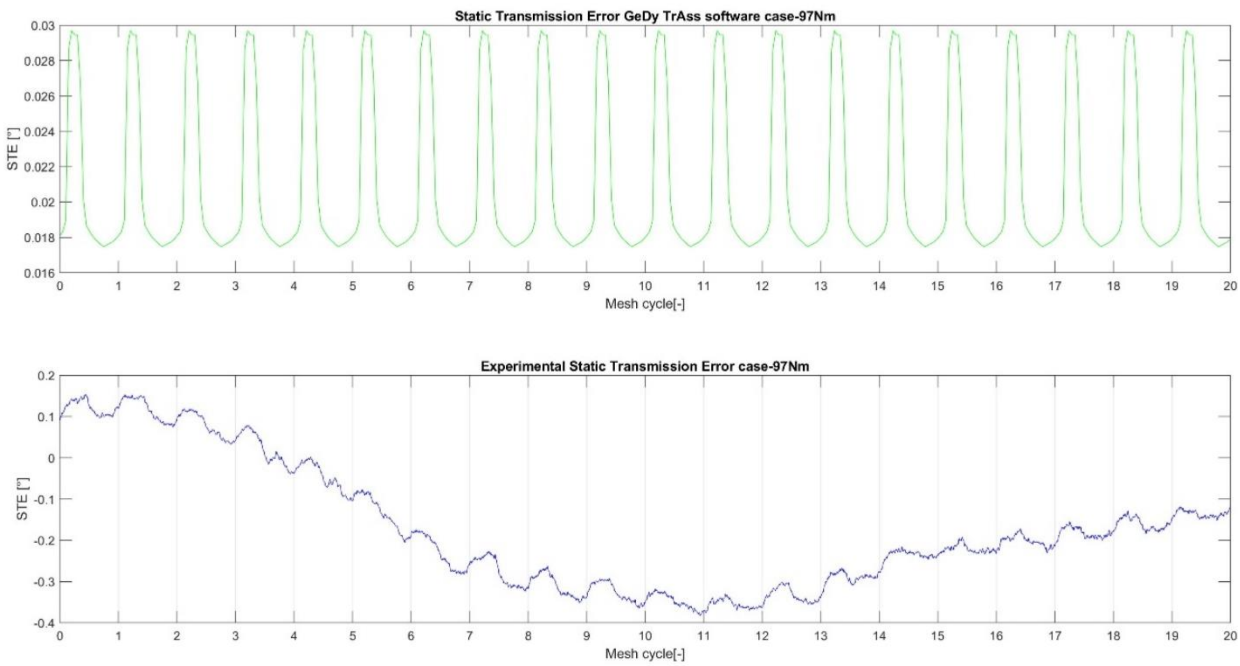

Figure 45,Comparison at 97 Nm

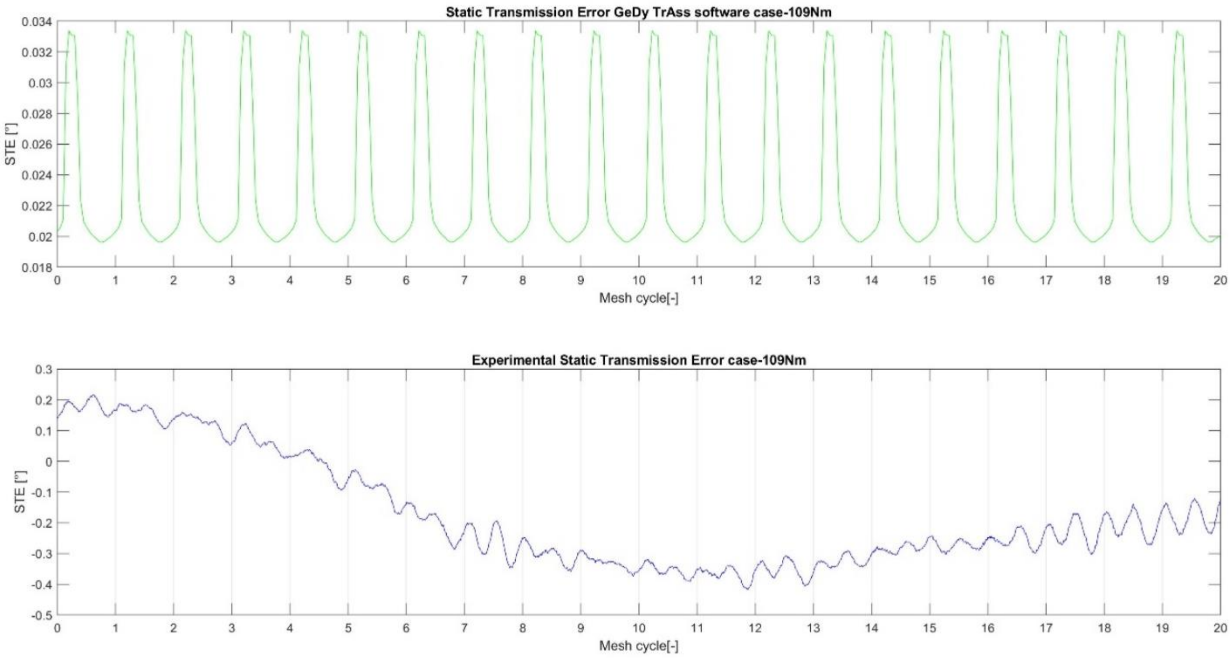

Figure 46,Comparison at 109 Nm

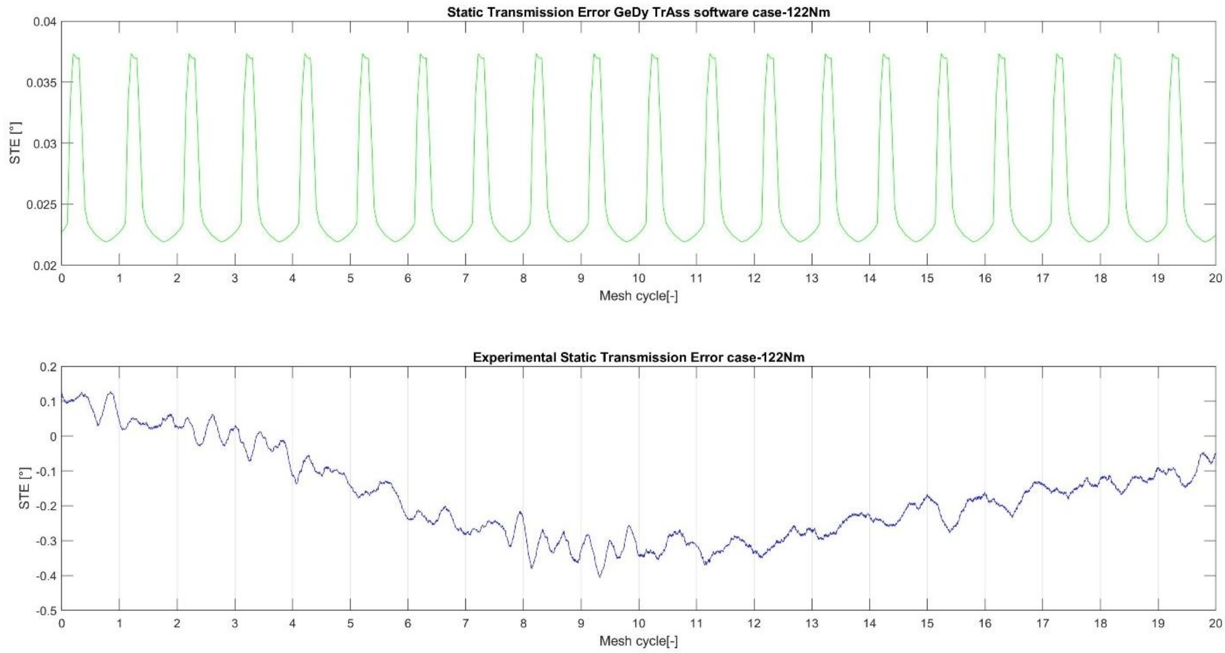

Figure 47,Comparison at 122 N

Once the comparison between the cases was shown, the various PPTEs were inserted in a graph in order to understand whether or not they follow similar trends.

As in Test 1, the data extrapolated from the software maintain and follow a linear trend, while those obtained from the test bench differ considerably from this trend.

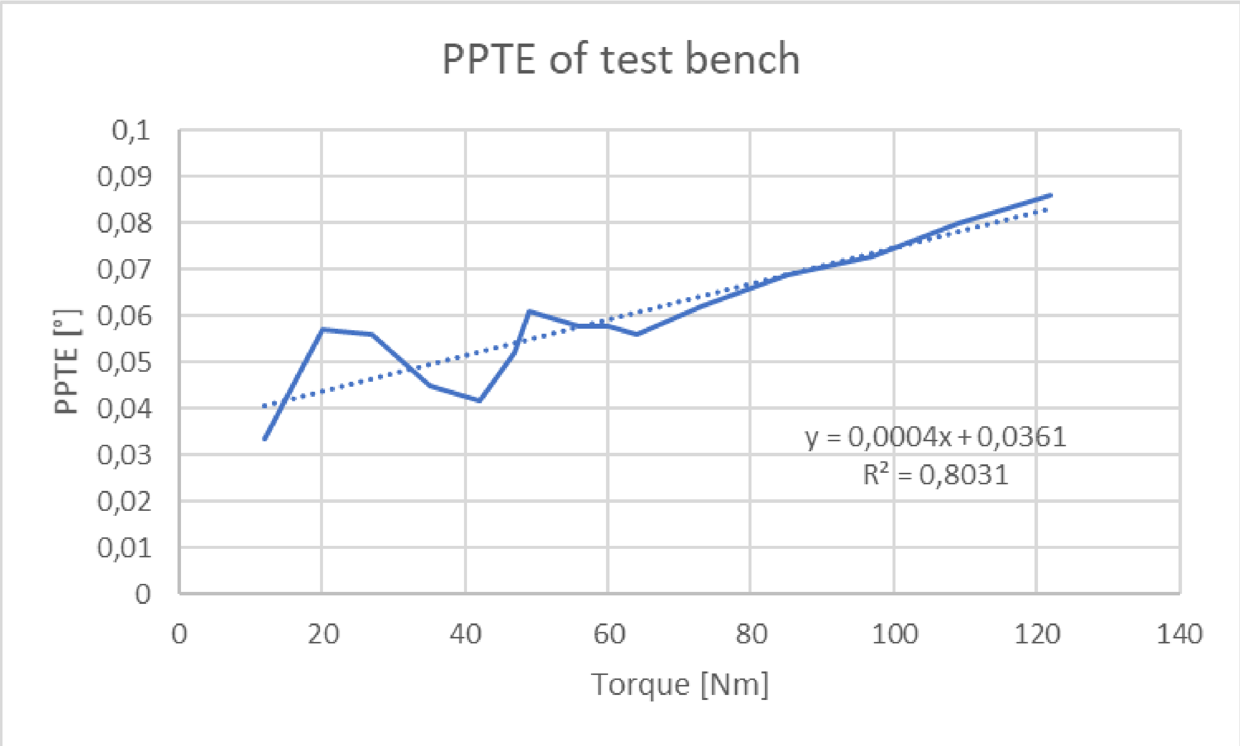

Figure 48,Trend of test bench values

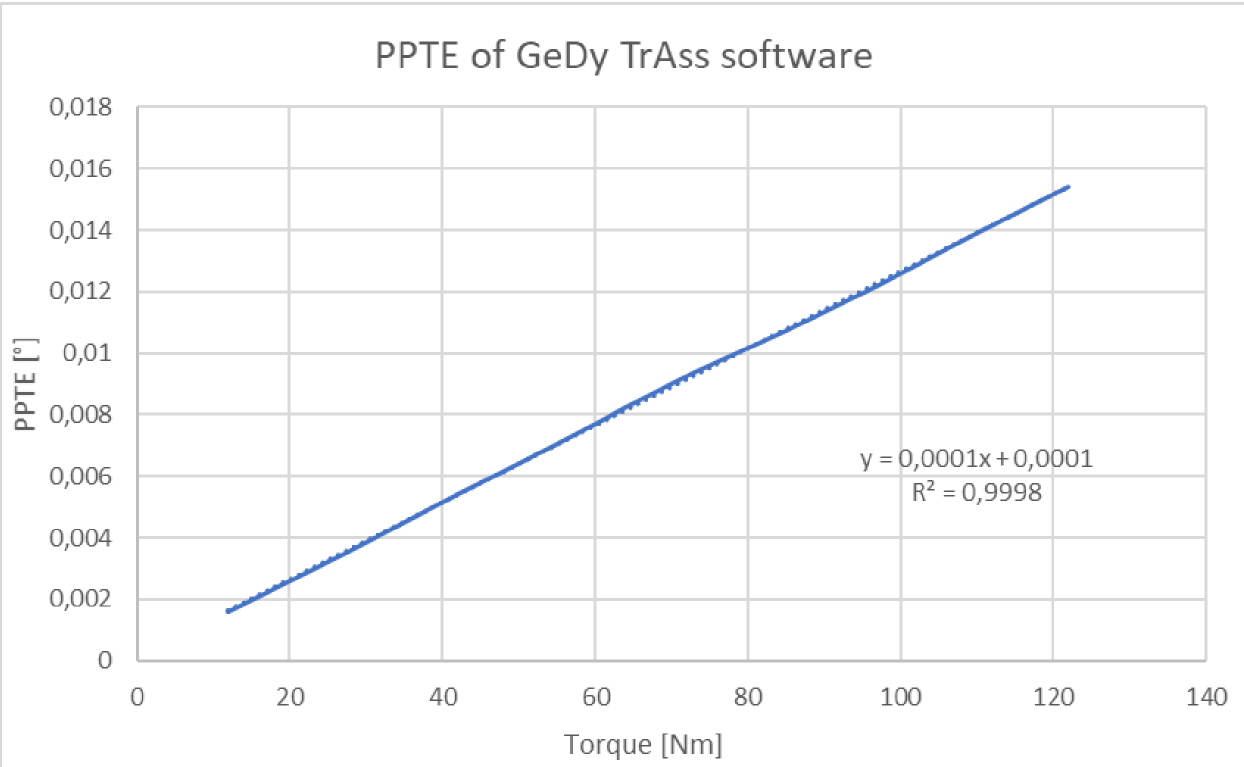

Figure 49,Trend of software values

7.2.2.2 Software, Test bench and Literature

In the comparison between the values of literature, test bench and GeDy TrAss software, as regards the new Test, having a large number of torques permitted to get closer to those that NASA used as a sample, underlining however without reaching them perfectly, but reaching a more accurate comparison respect to Test1.

The reduction of the value gap has made possible the comparison also at a value near of 79 Nm that is in our case 73Nm, as it is shown in Figure 50

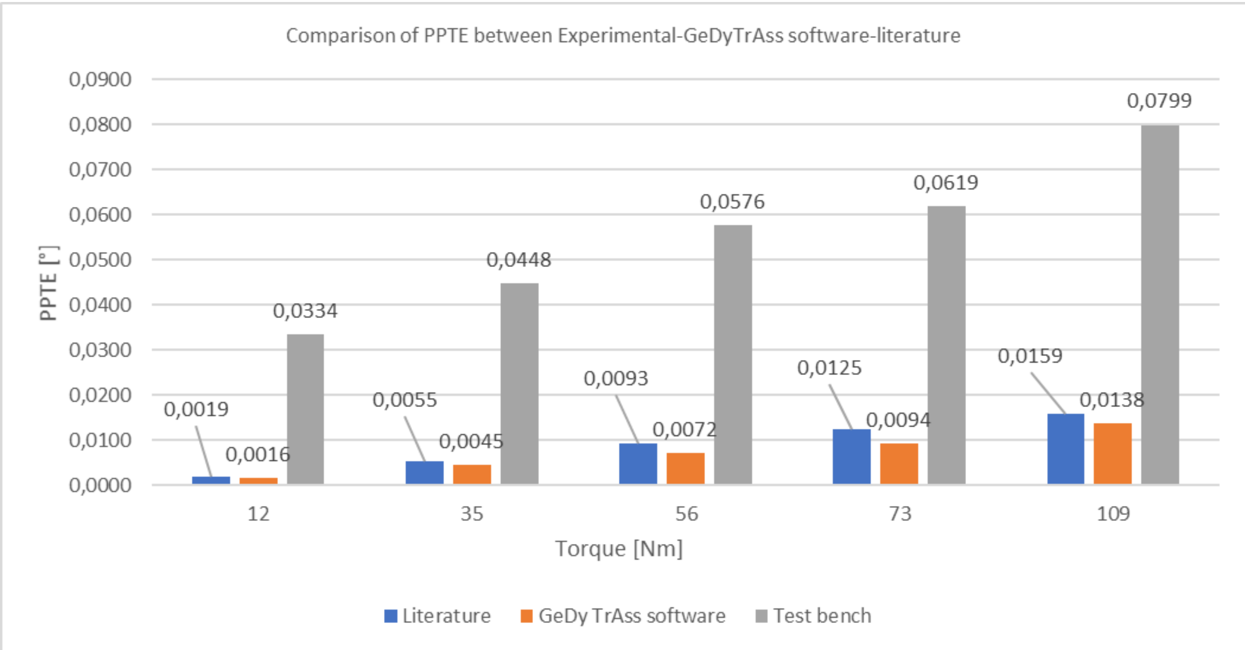

Figure 50, Comparison of PPTE

Also in this second Test the values obtained from the test bench are significantly higher than those compared with, however it is evident that the discrepancy has been reduced.

8 New procedure for the development of the Static Transmission Error

8.1 Code and description

Having processed the two tests described above, it was possible to notice how the trends of the STEs found and consequently the values obtained are still clearly far from the corresponding ones obtained with the GeDy TrAss software.

From the previous tests it was realized how the torque trend greatly influenced the STE, the continuous oscillations given by the hoist, we are talking about intrinsic vibrations of the hoist due to his motor, involves dirtying the signal sought, making it discontinuous and different from the standard models.

For the aforementioned reason, the surveys for Test 3 were made only in the ascent phase, towards traction in which the hoist guarantees better behavior, distorting the signal less.

All these considerations forced us to change our modus operandi, that is to filter the signal produced of the possible disturbances produced by the test bench, thus going to search only the actual STE.

First of all the STE signal is cut in such a way that the starting and ending points are at the same ordinate, in order to be repeatable.

After that, we go to search within the signal itself all the frequencies that compose it, trying to bring out only and exclusively the STE signal and not all the disturbances.

8.2 Steps of the procedure

The first step is to calculate the discrete Fourier transform (DFT) using a fast Fourier transform (FFT) algorithm, at the same time, thanks to the MATLAB "hann" command, a Hann Window is created, which reduces the truncation effect due to the observation time.

It should be emphasized that to speed up the FFT calculation, it is possible to transform a signal into base 2, when its length is not a power exact of 2.

At this point it is essential to obtain the characteristic frequency of the gears studied, to be found in the Fourier spectrum of the signal, having the angular variation, the time of this variation and the number of teeth.

$$f = \left(\frac{\frac{\Delta\vartheta}{\Delta Time}}{2\pi} \right) * n°of\ theeth$$

At this point of the discussion of the different steps Figure 51 represents a bar graph of the first frequencies that the signal studied has inside it, where it is possible to search for the actual characteristic frequency of the teeth, a numerical example of a possible frequency is given on the chart.

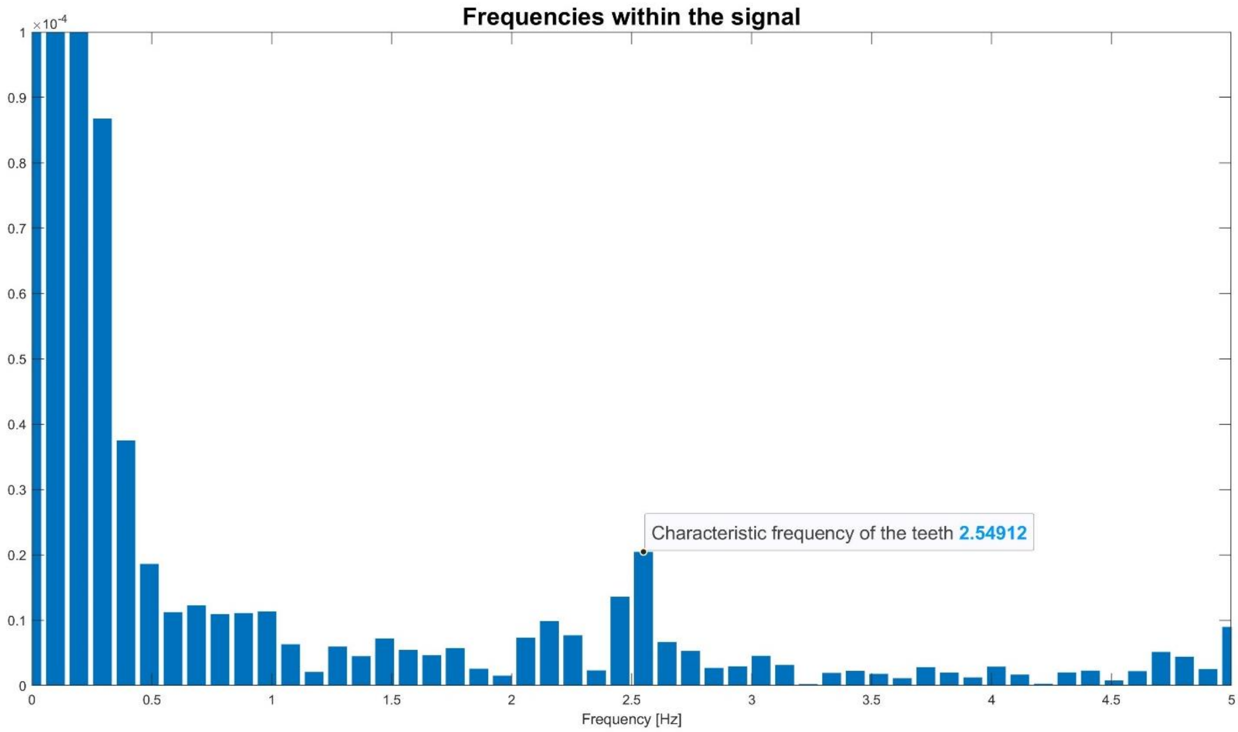

Figure 51, Frequencies of the signal

Finally, the inverse of the Fourier transform is performed to reconstruct only the STE signal sought.

As regards the reconstruction of the signal, only the first 10 multiples of that characteristic of the tooth and consequently the last ten were taken into consideration in the frequency mapping, since the spectrum is double.

```
%% calculation of STE
zp=28;
zdg=28;
STE = Qdg-(zp/zdg)*Qp;
f5=figure(5);
plot(timeen,STE*180/pi);
figure
plot(timeen,Qp)
%%% calculation of the characteristic frequency of the teeth
freq=abs((((Ch(2,2)-Ch(1,2))/(Ch(2,1)-Ch(1,1)))/(2*pi))*28);
%% elaboration of the signal and 2^17 it is our new base of the signal
P=nextpow2(length(STEn));
lzz=length(STEn);
l17=linspace(0,1,(2^17));
sten=interp1(linspace(0,1,lzz),STEn,l17);
```

```
TMP=interp1(linspace(0,tenc(end),length(tenc)),tenc,linspace(0,tenc(end),2^17));
figure
plot(sten)
window=hann(2^17);
ger=fft(sten.*window');
ger1=zeros(size(ger,2),1);
ger1(1)=ger(1)/size(ger,2);
ger1(2:end)=ger(2:end)/(2*size(ger,2));
num3=round(size(ger1,1));
f3=linspace(0,1/TMP(2),num3);
figure
plot(f3,abs(ger1));
xlim([0 5])
ylim([0 0.001])


arm=round(find(f3==Carm(1,1)));
mapg=arm*linspace(1,10,10);
ger2=zeros(size(ger,2),1);
mapg=[1 mapg size(ger2,1)+2-arm*linspace(1,10,10)];
ger2(mapg)=ger(mapg);
STE=ifft(ger2);
figure
plot(STE*180/pi);
```

8.3 Test 3

8.3.1 Software and Test bench

The Figures below show the outputs comparisons between the signal calculated by GeDy TrAss and the signal measured experimentally through the bench, 9 proofs were performed, increasing each time by 5 kg, stopping at a weight value that guarantees not to exceed too much the value of 100 Nm.

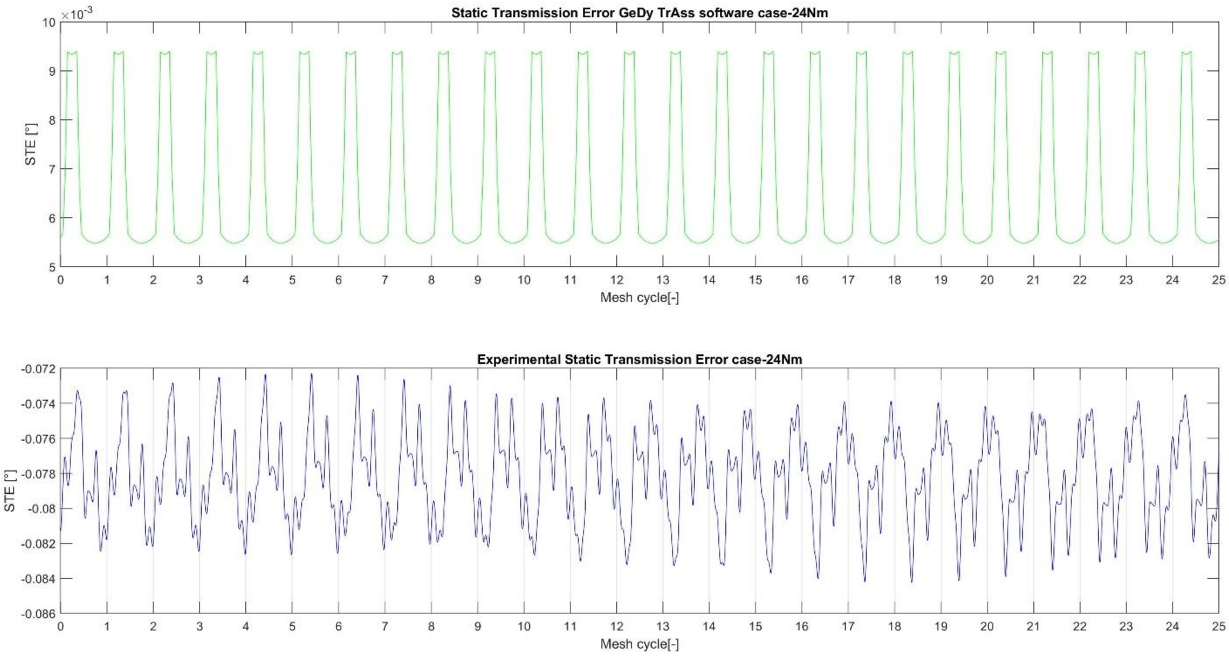

Figure 52, Comparison at 24 Nm

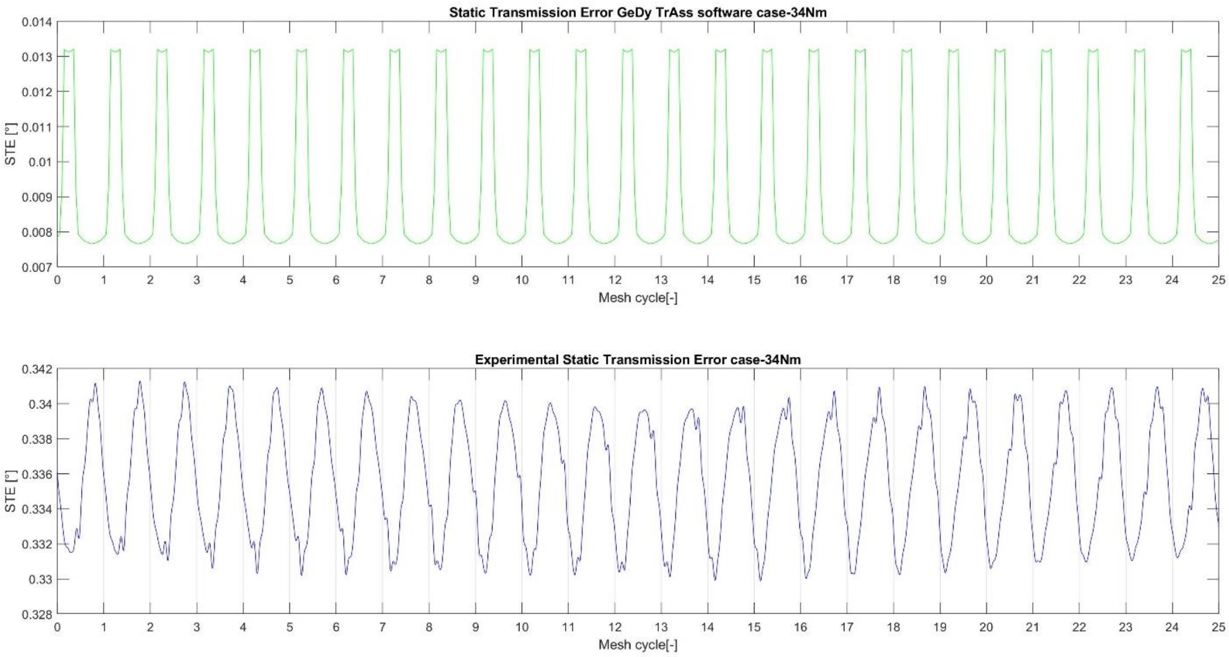

Figure 53,Comparison at 34 Nm

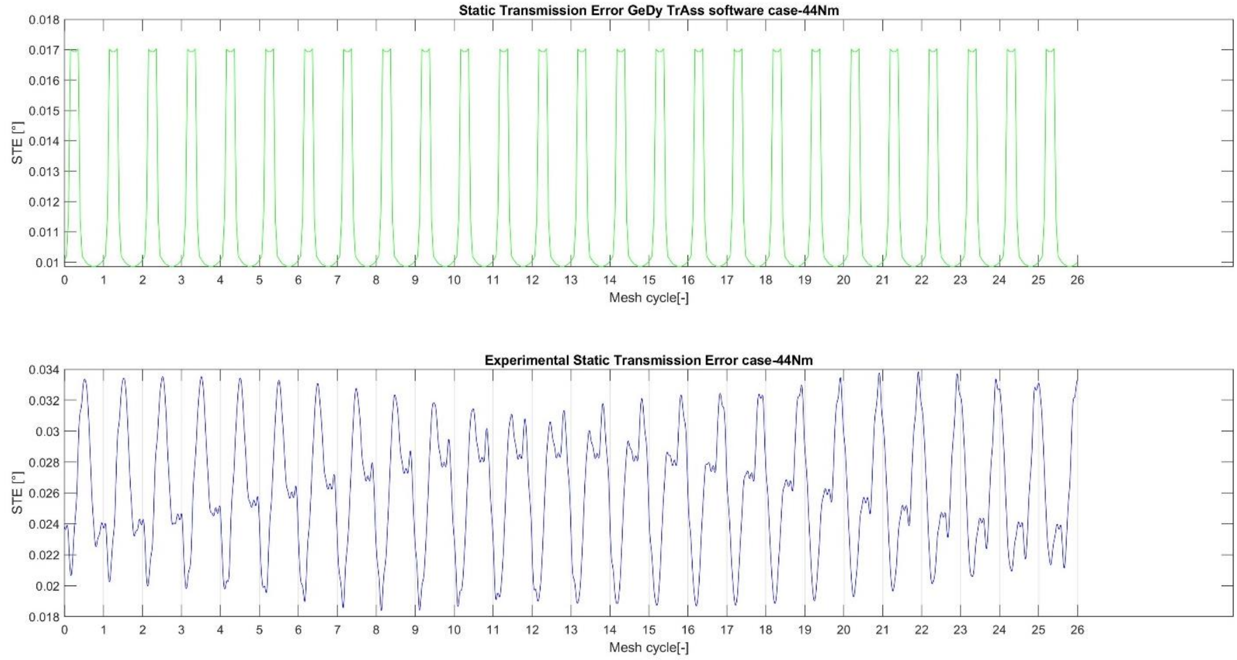

Figure 54,Comparison at 44 Nm

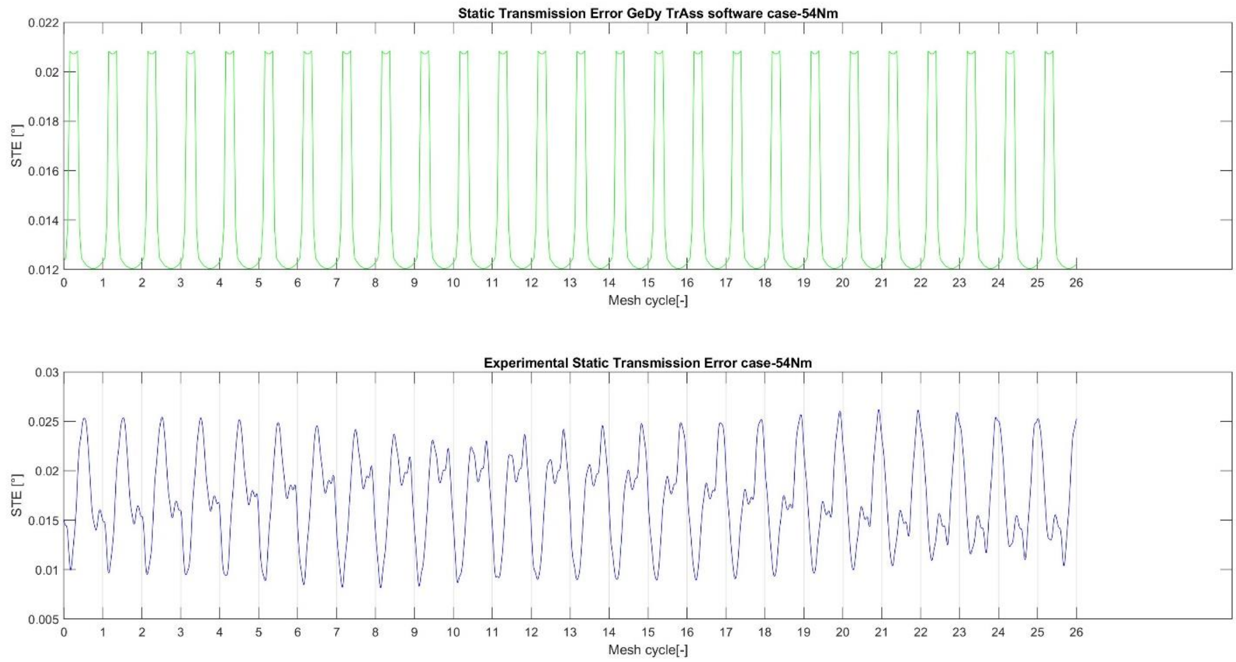

Figure 55,Comparison at 54 Nm

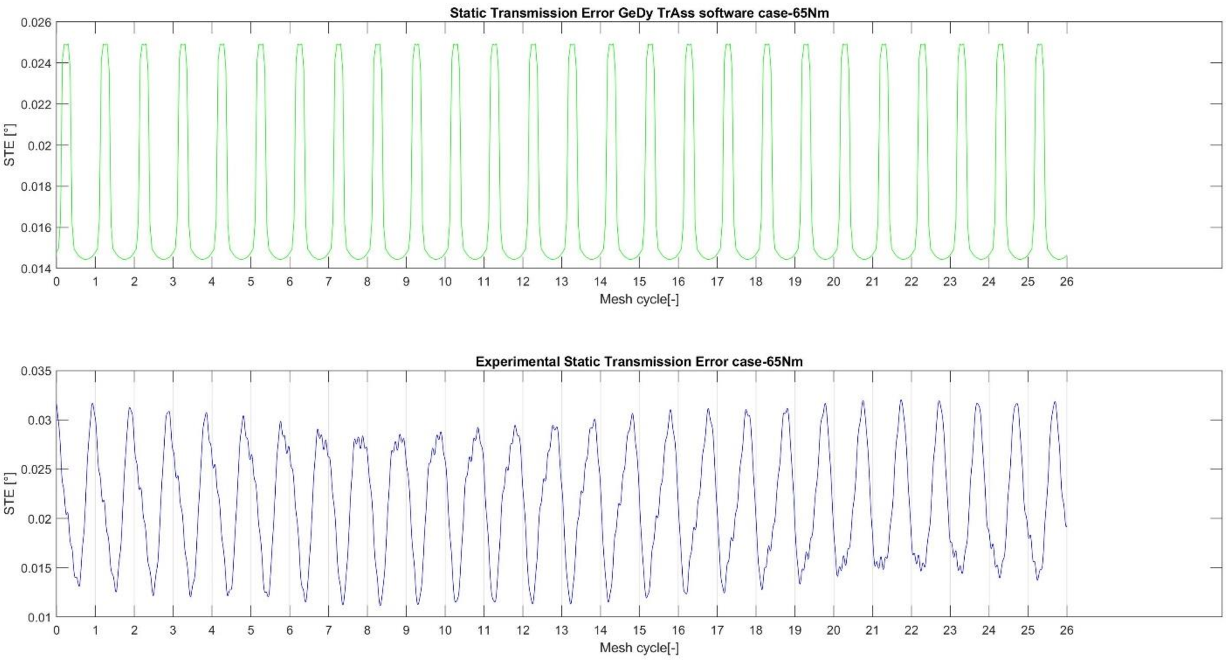

Figure 56,Comparison at 65 Nm

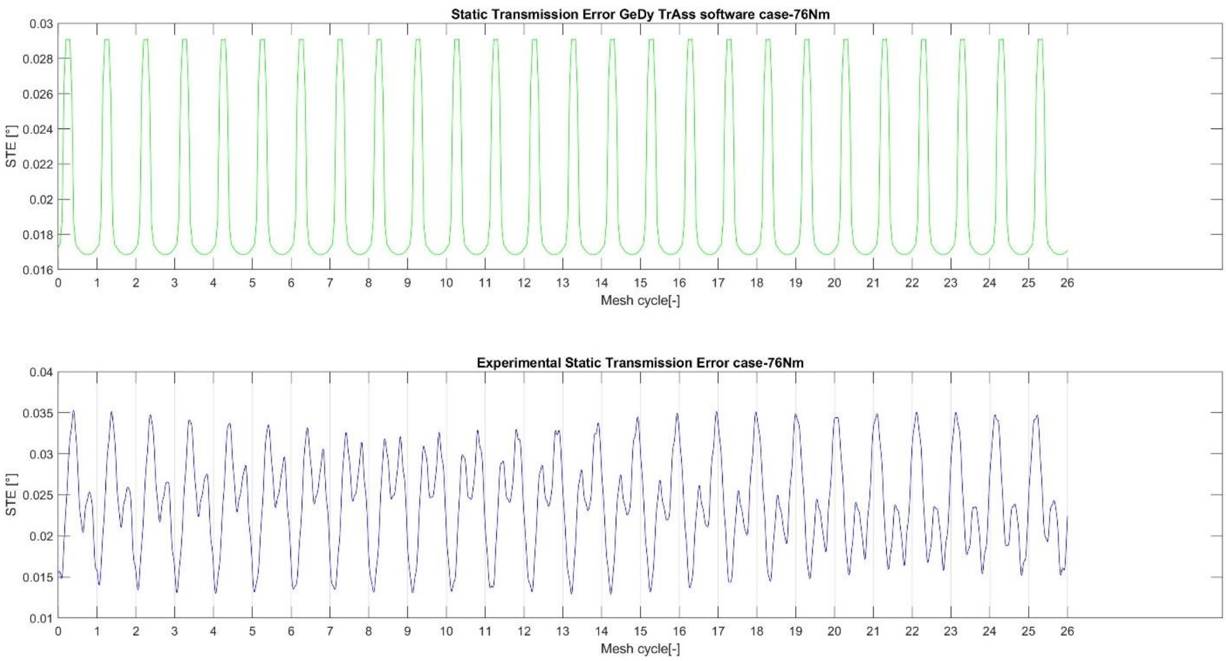

Figure 57,Comparison at 76 Nm

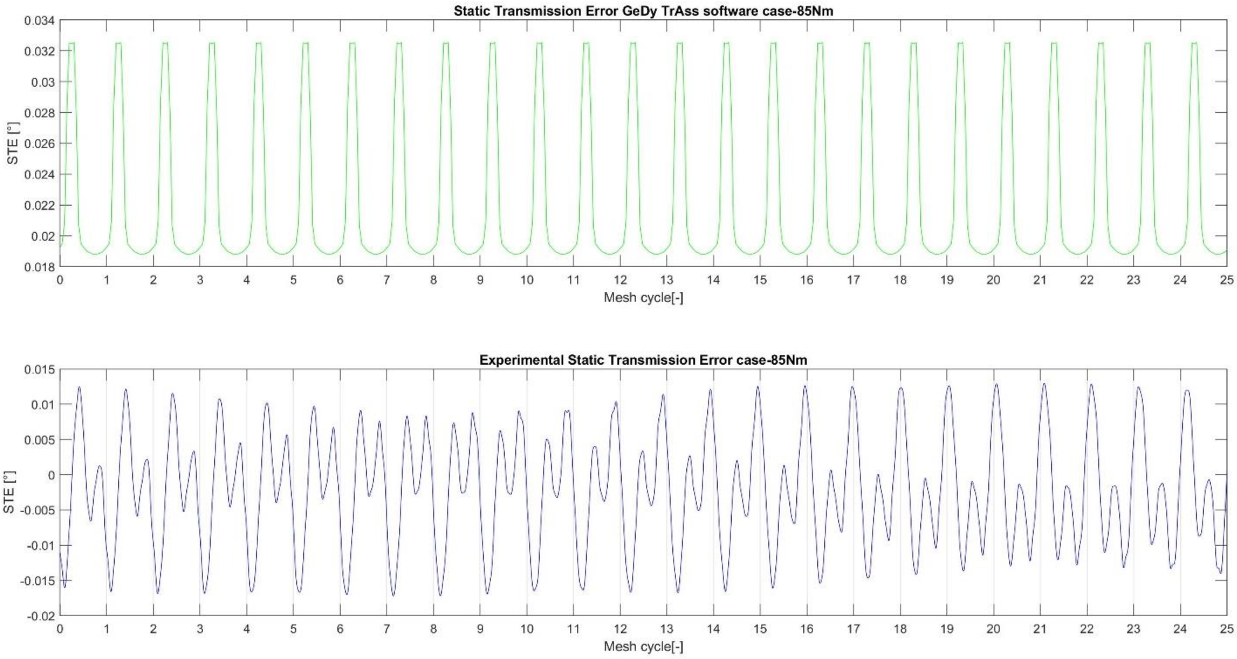

Figure 58,Comparison at 85 Nm

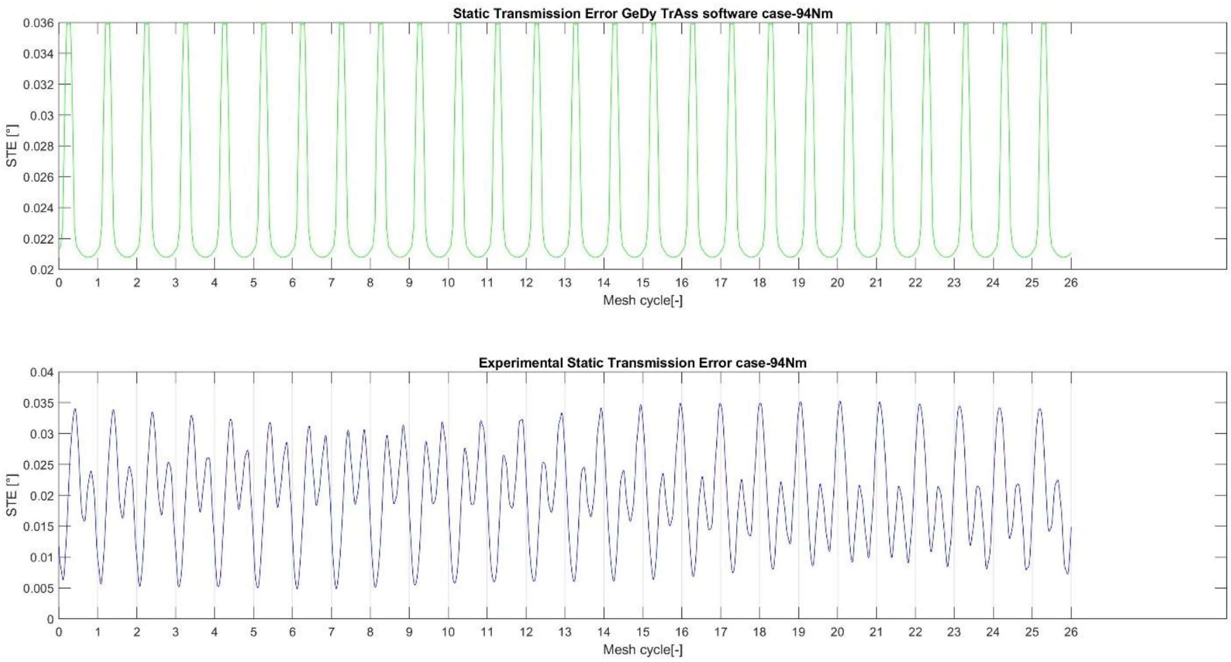

Figure 59,Comparison at 94 Nm

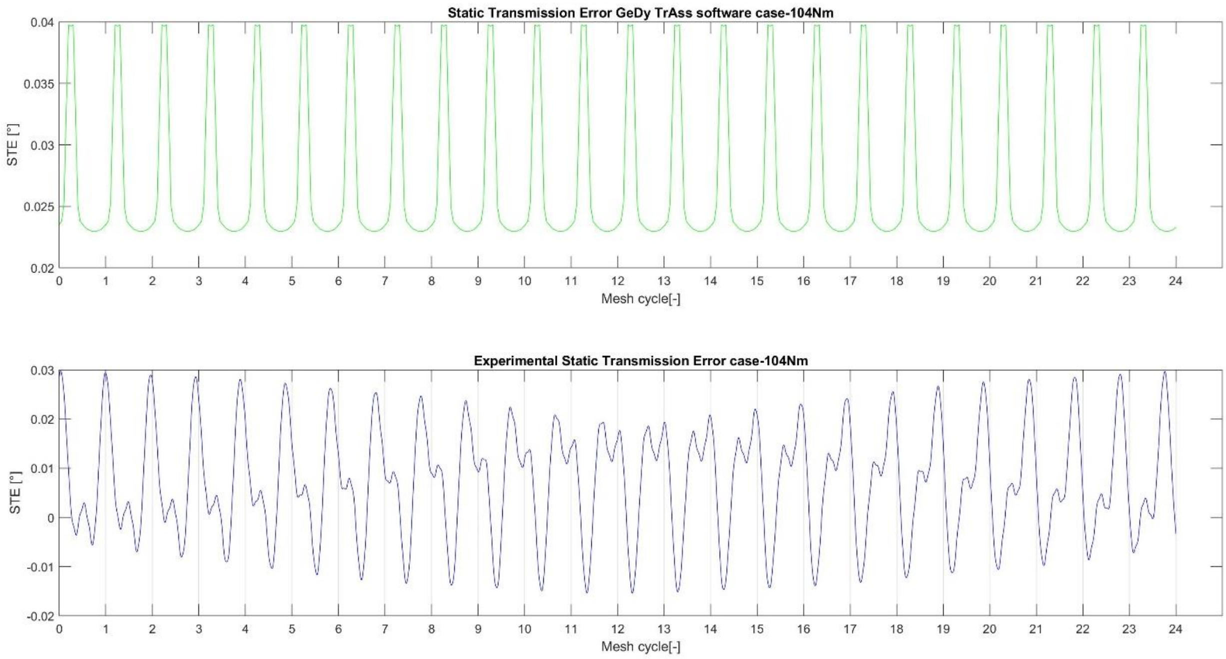

Figure 60,Comparison at 104 Nm

As can be seen from the Figures, compared to the previous tests, thanks to the new procedure the STEs produced show a very precise pattern, sometimes even repetitive and not just a jagged line as in Test 2, underlining how the true STE signal has been extrapolated from the total.

At this point, a comparison is made between the PPTE values obtained from the bench and those obtained from the software, Figure 61.

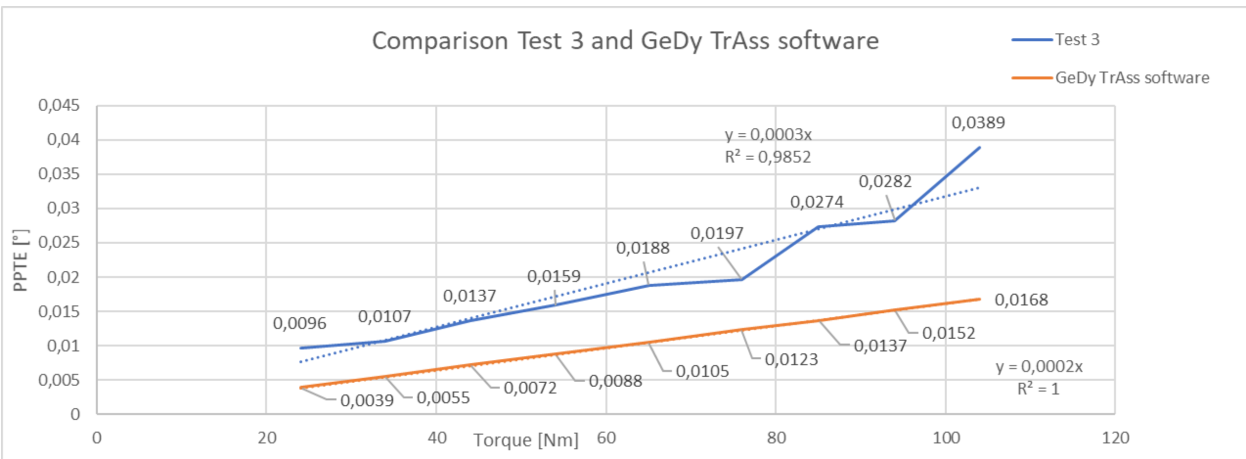

Figure 61, Test 3 and software PPTE

In order to understand better the graphs of PPTE, it has been reported, also in this case, the trend lines and it is worth focusing on the meaning of the R^2.

It represents an indicator that starting from the regression line, summarizes in a single value how much the analyzed quantity deviates on average from this line, the more the value approaches 1 and the more there is a perfect linear relationship between the phenomenon analyzed and its regression line.

The trend line was forced to pass through the origin to better understand the trend of the experimental PPTE values.

It is noteworthy that thanks to the new test carried out, it was possible to achieve a clear improvement in terms of numerical results, in this case it is possible to note how the values of R^2 reached are remarkably close to the value 1, this indicates that the population of our example follows a linear trend, like that of the PPTE obtained by means of the software, it should also be noted that the experimental data show a slight offset.

A reason for this gap could be given by the fact that the values of GeDy TrAss with which the comparison was made do not take into account the theory of Hertzian contact.

In order to give truth to the new procedure it was decided to re-elaborate the data of Test 2, always taking only the ascent phase.

First of all the PPTE values of the test bench and software are compared in Figure 62,

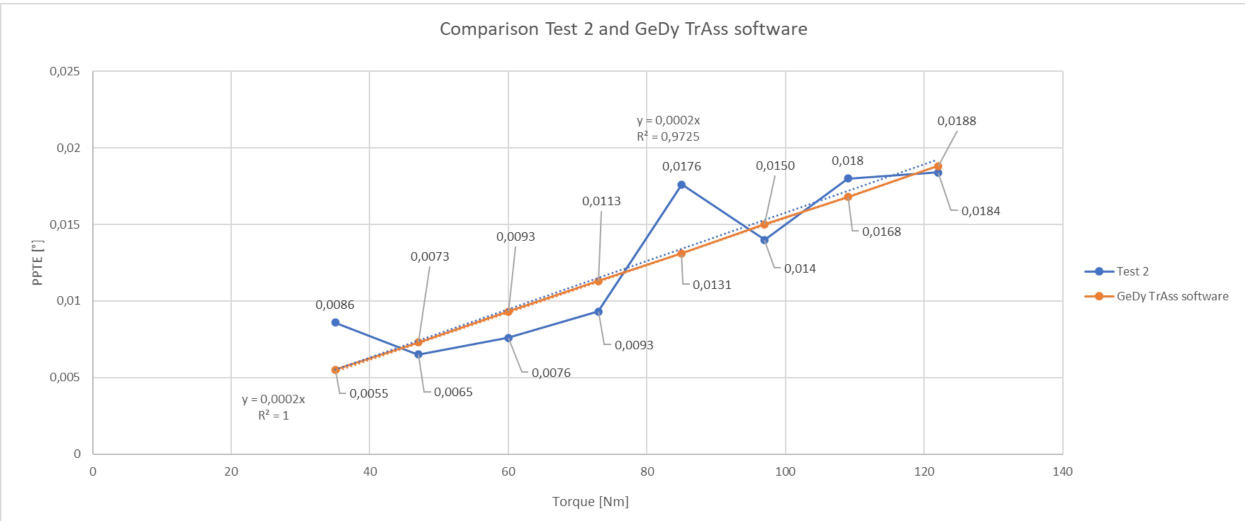

Figure 62, PPTE Test 2 and software

With Test 2, a trend similar to that of the software is achieved, it is noted that the trend fluctuates, but in the same time reaches a very high value of R^2 index.

then the Figure 63, shows the comparison of values between the two different procedures,

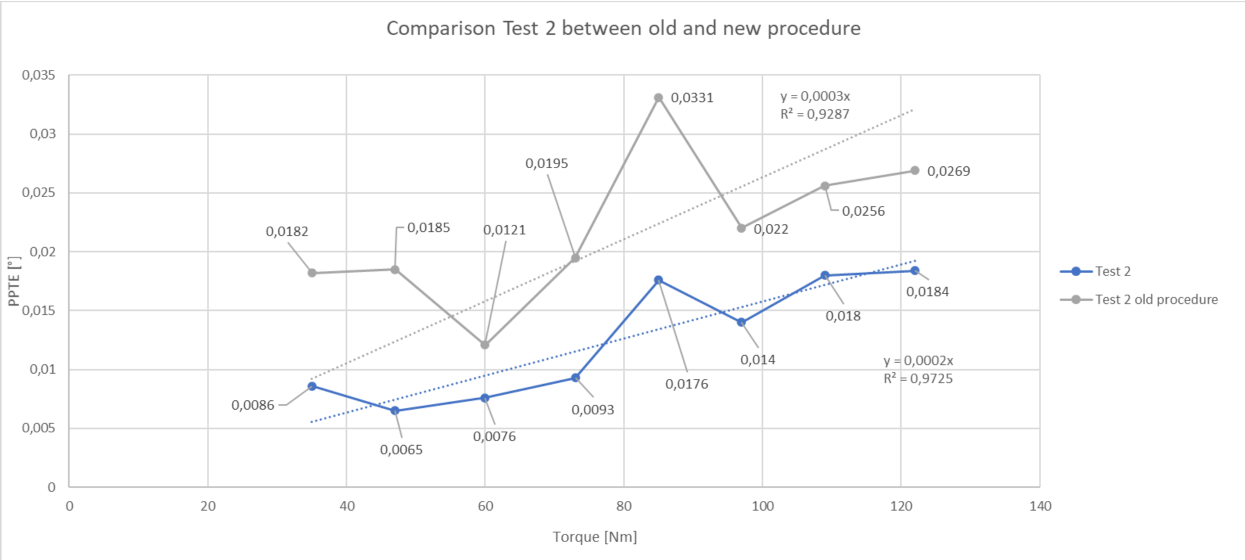

Figure 63, Old procedure and new procedure

from the Figure it is clear that the new procedure obtains a higher R coefficient, therefore a more linear trend than the old procedure.

9 Conclusions

This experimental thesis work was born from the need to derive the Static Transmission Error of two coupled gears from a test bench.

During the different tests, setup problems were encountered, which forced the team to change its mind on the modus operandi resulting in subsequent changes.

The most obvious problems were the reduction of friction on the pulleys or the correct positioning of the mobile turret compared to the fixed one, as well as all the adjustments for the correct positioning of the gears.

All precautions that have allowed the correct movement of the line and of the gears.

Focusing on the essential modifications of the test bench, a motorization of the bench was needed, but with a tool that maintained the static condition, which would have speeded up and helped the lifting and lowering maneuvers, so a hoist was used.

The hoist guaranteed reduced speeds and excellent uphill behavior, but irregular downhill behavior, as it was not designed to act as a brake.

After assembling the sample gears and creating the MATLAB code, the first tests were carried out, compared both with the GeDy TrAss home software and then also with the literature deriving from NASA tests.

The results obtained differed by orders of magnitude from the respective calculations, although the experimental values trends tend to be different, it was easy to realize that the way of operating on the collected data had to be changed.

The procedure used was therefore replaced by a discourse more focused on the signal, it presented numerous dirt and disturbance, given by friction and noise, the hoist itself changes the signal.

After analyzing the Transmission Error in all the tests carried out, it was decided to focus the study only on the harmonics concerned, managing to reconstruct the exact trend of the error sought.

The values of PPTE obtained were of the order of magnitude of those calculated and compared and also the trends, thanks to the value of R^2, follow a linear behavior, a result that highlights how the values obtained have a logical and sensible meaning.

The experimental's results still differ from those of the software, in fact, the latter do not take into account the effect of the Hertz Theory on the contact between the teeth of the coupled gears, which was instead considered experimentally.

As for future developments and improvements, looking at some graphs of the comparisons between STEs it can be seen how the trends are different from what it is compared with, there is still a small presence of run out, certainly some studies on the behavior of the tooth when it is excited will give more explanations, but also the possibility of implementing two electric motors on the test bench, one motor and one brake could help the correct carrying out of the tests, and finally it could also decide to replace the type of wire used with another that guarantees less deformation when it is in traction.

Having said that, it is possible to say with certainty that the desired result has been achieved.

10 Code for graphic outputs

Below is the code that it was possible to produce the graphic outputs of the values collected and compared, it is the code executed on Test 1, adaptable to any type of test, but this case was reported, which includes less data, only for the purpose to show how it works and not burden the writing of the thesis.

```
%% Plot pinion and driven gear
tiledlayout(2,2)
nexttile
plot(p)
xlabel('Number of value');
ylabel('[°]');
title('Angular displacement');
legend('Pinion');
nexttile
plot(dg)
xlabel('Number of value');
ylabel('[°]');
title('Angular displacement');
legend('Drived gear');
nexttile
plot(Qp)
xlabel('Number of value');
ylabel('[rad]');
title('Angular displacement');
legend('Pinion');
nexttile
plot(Qdg)
xlabel('Number of value');
ylabel('[rad]');
title('Angular displacement');
legend('Driven gear');


%% variation due to torque
figure
plot(tenc,STE_c*180/pi);
xlabel('Time [s]');
ylabel('STE [°]');
title('Raw Static Transmission Error');


%% without noload right side
%%% take Ctimer and Ctimel from the grapf below
timer=tenc((Cload1:Cload2)-Cste1)-15.9103;
timel=tenc(Cload4:Cload5)-0.0937;
figure
tiledlayout(2,1)
nexttile
plot(timer,Ste_pc*180/pi);
xlabel('time [s]');
```

```
ylabel('[°]');
title('Static Transmission Error right side');




%%% without noload left side
nexttile
plot(timel,Ste_pcl*180/pi);
xlabel('Time [s]');
ylabel('[°]');
title('Static Transmission Error left side');

%% no-load case
figure
plot(timenoload,NSTE*180/pi);
xlabel('Time [s]');
ylabel('[°]');
title('Experimentsl Static Transmission Error of no-load');


%% pointers of the possibly mesh cycles
%%% right side
figure
tiledlayout(2,1)
nexttile
plot(nr,Ste_pc(nr)*180/pi,'*k');
hold on;
plot(Ste_pc*180/pi);
xlabel('Number of values');
ylabel('[°]');
title('Static Transmission Error right side');
legend('pointer of the mesh cycle');


%%% left side
nexttile
plot(nl,Ste_pcl(nl)*180/pi,'*k');
hold on;
plot(Ste_pcl*180/pi);
ylabel('[°]');
xlabel('Number of values');
title('Static Transmission Error left side');
legend('pointer of the mesh cycle');


%% Comparison of STE between different cases, divided into left and right side

%%% interpolation right side
x=length(STEn_case_56Nm);
y=length(STEn_case_28Nm);
z=length(STEn_case_13Nm);
Xspace=linspace(0,1,x);
Yspace=linspace(0,1,y);
```

```
Zspace=linspace(0,1,z);
% STE_28Nm = interp1(Yspace,STE_case_28Nm,Xspace);
% STE_13Nm = interp1(Zspace,STE_case_13Nm,Xspace);
% STE_56Nm=STE_case_56Nm;

%%% 13
angle13=(0:32.7:327);
xangle13=length(angle13);
xspaceangle13=linspace(0,1,xangle13);
Angle13=interp1(xspaceangle13,angle13,Zspace);

%%% 28
angle28=(0:31.5:315);
xangle28=length(angle28);
xspaceangle28=linspace(0,1,xangle28);
Angle28=interp1(xspaceangle28,angle28,Yspace);

%%% 56
angle56=(0:30.4:304);
xangle56=length(angle56);
xspaceangle56=linspace(0,1,xangle56);
Angle56=interp1(xspaceangle56,angle56,Xspace);

%% interpolation left side

x1=length(STEn_case_110Nm);
y1=length(STEn_case_60Nm);
z1=length(STEn_case_35Nm);
X1space=linspace(0,1,x1);
Y1space=linspace(0,1,y1);
Z1space=linspace(0,1,z1);
% STE_60Nm = interp1(Y1space,STE_case_60Nm,X1space);
% STE_35Nm = interp1(Z1space,STE_case_35Nm,X1space);
% STE_110Nm=STE_case_110Nm;

%%% 35
angle35=(0:32.7:327);
xangle35=length(angle35);
xspaceangle35=linspace(0,1,xangle35);
Angle35=interp1(xspaceangle35,angle35,Z1space);

%%% 60
angle60=(0:31.5:315);
xangle60=length(angle60);
xspaceangle60=linspace(0,1,xangle60);
Angle60=interp1(xspaceangle60,angle60,Y1space);

%%% 110
angle110=(0:30.4:304);
xangle110=length(angle110);
xspaceangle110=linspace(0,1,xangle110);
Angle110=interp1(xspaceangle110,angle110,X1space);

%% STE right side
```

```
figure
plot(Angle56,STEn_case_56Nm*180/pi,'b');
hold on;
plot(Angle28,STEn_case_28Nm*180/pi,'r--');
plot(Angle13,STEn_case_13Nm*180/pi,'g-.');
ylabel('STE [°]');
xlabel('Angular variation [°]');
title('Experimental Static Transmission Error right side');
leg = legend('56Nm','28Nm','13Nm')
title(leg,'Torque Nm');

%% STE left side
figure
plot(Angle110,STEn_case_110Nm*180/pi,'b');
hold on;
plot(Angle60,STEn_case_60Nm*180/pi,'r--');
plot(Angle35,STEn_case_35Nm*180/pi,'g-.');
ylabel('STE [°]');
xlabel('Angular variation [°]');
title('Experimental Static Transmission Error left side');
leg = legend('110Nm','60Nm','35Nm')
title(leg,'Torque Nm');

%% Comparison between STE from test bench and GeDy TrAss software

%% interpolazione right side
% x=length(STEn_case_56Nm);
y=length(S28);
z=length(S13);
% Xspace=linspace(0,1,x);
Yspace=linspace(0,1,y);
Zspace=linspace(0,1,z);

%% interpolazione left side
x1=length(S110);
y1=length(S60);
z1=length(S35);
X1space=linspace(0,1,x1);
Y1space=linspace(0,1,y1);
Z1space=linspace(0,1,z1);

%% take from software
%%% from plot
%%%70-5 (13/35)  25/25
%%%70-15   (60/28)  23/24
%%%100-35  (110/56) 23
p110=(l110-1)*560/28;
p60=(l60-1)*560/28;
p35=(l35-1)*560/28;
p28=(l28-1)*560/28;
p13=(l13-1)*560/28;
STE13s=STE(1,1:p13);
STE28s=STE(2,1:p28);
STE35s=STE(3,1:p35);
```

```
STE60s=STE(4,1:p60);
STE110s=STE(5,1:p110);

%% interpolation software-code

%%%110
l_110=length(STE110s);
l_110space=linspace(0,1,l_110);
STEs_110Nm = interp1(l_110space,STE110s,X1space);
a110=(0:2.3:23);
al110=length(a110);
alspace110=linspace(0,1,al110);
X110=interp1(alspace110,a110,X1space);
tiledlayout(2,1)
f1=nexttile
plot(f1,X110,STEs_110Nm,'g-');
set(gca, 'XTick', 0 : 1 : l110-1);
ylabel('STE [°]');
xlabel('Mesh cycle[-]');
title('Static Transmission Error GeDy TrAss software case-110Nm');
f2=nexttile
plot(f2,x110,S110*180/pi,'b');
set(gca, 'XTick', 0 : 1 : l110-1);
ylabel('STE [°]');
xlabel('Mesh cycle[-]');
title('Experimental Static Transmission Error case-110Nm');
grid(f2)
set(gca, 'YGrid', 'off');

%%%60
l_60=length(STE60s);
l_60space=linspace(0,1,l_60);
STEs_60Nm = interp1(l_60space,STE60s,Y1space);
a60=(0:2.3:23);
al60=length(a60);
alspace60=linspace(0,1,al60);
X60=interp1(alspace60,a60,Y1space);


tiledlayout(2,1)
f3=nexttile
plot(f3,X60,STEs_60Nm,'g-');
set(gca, 'XTick', 0 : 1 : l60-1);
ylabel('STE [°]');
xlabel('Mesh cycle[-]');
title('Static Transmission Error GeDy TrAss software case-60Nm');
f4=nexttile
plot(f4,x60,S60*180/pi,'b');
set(gca, 'XTick', 0 : 1 : l60-1);
ylabel('STE [°]');
xlabel('Mesh cycle[-]');
title(' Experimental Static Transmission Error case-60Nm ');
grid(f4)
set(gca, 'YGrid', 'off');
```

```
%%%35
l_35=length(STE35s);
l_35space=linspace(0,1,l_35);
STEs_35Nm = interp1(l_35space,STE35s,Z1space);
a35=(0:2.5:25);
al35=length(a35);
alspace35=linspace(0,1,al35);
X35=interp1(alspace35,a35,Z1space);
tiledlayout(2,1)
f5=nexttile
plot(f5,X35,STEs_35Nm,'g-');
set(gca, 'XTick', 0 : 1 : l35-1);
ylabel('STE [°]');
xlabel('Mesh cycle[-]');
title('Static Transmission Error GeDy TrAss software case-35Nm');
f6=nexttile
plot(f6,x35,S35*180/pi,'b');
set(gca, 'XTick', 0 : 1 : l35-1);
ylabel('STE [°]');
xlabel('Mesh cycle[-]');
title(' Experimental Static Transmission Error case-35Nm ');
grid(f6)
set(gca, 'YGrid', 'off');

%%%28
l_28=length(STE28s);
l_28space=linspace(0,1,l_28);
STEs_28Nm = interp1(l_28space,STE28s,Yspace);
a28=(0:2.4:24);
al28=length(a28);
alspace28=linspace(0,1,al28);
X28=interp1(alspace28,a28,Yspace);
tiledlayout(2,1)
f7=nexttile
plot(f7,X28,STEs_28Nm,'g-');
set(gca, 'XTick', 0 : 1 : l28-1);
ylabel('STE [°]');
xlabel('Mesh cycle[-]');
title('Static Transmission Error GeDy TrAss software case-28Nm');
f8=nexttile
plot(f8,x28,S28*180/pi,'b');
set(gca, 'XTick', 0 : 1 : l28-1);
ylabel('STE [°]');
xlabel('Mesh cycle[-]');
title(' Experimental Static Transmission Error case-28Nm ');
grid(f8)
set(gca, 'YGrid', 'off');


%%%13
l_13=length(STE13s);
l_13space=linspace(0,1,l_13);
```

```
STEs_13Nm = interp1(l_13space,STE13s,Zspace);
a13=(0:2.5:25);
al13=length(a28);
alspace13=linspace(0,1,al13);
X13=interp1(alspace13,a13,Zspace);
tiledlayout(2,1)
f9=nexttile
plot(f9,X13,STEs_13Nm,'g-');
set(gca, 'XTick', 0 : 1 : l13-1);
ylabel('STE [°]');
xlabel('Mesh cycle[-]');
title('Static Transmission Error GeDy TrAss software case-13Nm');
f10=nexttile
plot(f10,x13,S13*180/pi,'b');
set(gca, 'XTick', 0 : 1 : l13-1);
ylabel('STE [°]');
xlabel('Mesh cycle[-]');
title(' Experimental Static Transmission Error case-13Nm ');
grid(f10)
set(gca, 'YGrid', 'off');

%% histogram
rb=41.7693;
%%% software
v13s=STE(1,4)-STE(1,15);
v28s=STE(2,4)-STE(2,16);
v35s=STE(3,4)-STE(3,15);
v60s=STE(4,4)-STE(4,16);
v110s=STE(5,4)-STE(5,15);

%%% code
v13=abs(Cc13(2,2)-Cc13(1,2));
v28=abs(Cc28(2,2)-Cc28(1,2));
v35=abs(Cc35(2,2)-Cc35(1,2));
v60=abs(Cc60(2,2)-Cc60(1,2));
v110=abs(Cc110(2,2)-Cc110(1,2));


%%% literature
rb=41.7693;
v11p=atand((Cl11(2,2)-Cl11(1,2))*25.4/rb);
v34p=atand((Cl34(2,2)-Cl34(1,2))*25.4/rb);
v56p=atand((Cl56(2,2)-Cl56(1,2))*25.4/rb);
v102p=atand((Cl102(2,2)-Cl102(1,2))*25.4/rb);


%%% 2
coppie=[13 35 60 110];
errore=[v11p v13s v13; v34p v35s v35; v56p v60s v60 ;v102p v110s v110];
grafico2=bar(coppie,errore);


xtips1 = grafico2(1).XEndPoints;
ytips1 = grafico2(1).YEndPoints;
labels1 = string(grafico2(1).YData);
```

```
text(xtips1,ytips1,labels1)
% set(gca,text ,'HorizontalAlignment');


xtips2 = grafico2(2).XEndPoints;
ytips2 = grafico2(2).YEndPoints;
labels2 = string(grafico2(2).YData);
text(xtips2,ytips2,labels2)


xtips3 = grafico2(3).XEndPoints;
ytips3 = grafico2(3).YEndPoints;
labels3 = string(grafico2(3).YData);
text(xtips3,ytips3,labels3)
title('Comparison of PPTE between Experimental-GeDyTrAss software-literature');
xlabel('Torque [Nm]');
ylabel('PPTE [°]');
leg = legend('Literature PPTE','GeDyTrAss software PPTE','Experimetal PPTE')
title(leg,'Type of source');
```

11 References

[1] Tharmakulasingham, R & Alfano, Giulio & Atherton, Mark. (2008). Reduction of gear pair transmission error with tooth profile modification. 23rd International Conference on Noise and Vibration Engineering 2008, ISMA 2008. 2.

[2] Rosso, C., Maggi, T., Marcellini, C., and Bruzzone, F., "Test Bench for Static Transmission Error Evaluation in Gears", SAE Technical Paper 2020-01-1324, 2020, doi:10.4271/2020-01-1324.

[3] Fred B. Oswald, Hsiang Hsi Iin, Irebert R. Delgado, Dynamic Analysis of Spur Gear Transmissions (DANST), NASA Lewis Research Center Cleveland, Ohio 44135-3191 and Vehicle Propulsion Directorate U.S. Army Research Laboratory Cleveland, Ohio 44135-3191

[4] Bruzzone, F., Maggi, T., Marcellini, C., Rosso, C., Delprete, C., " Proposal of a novel approach for 3D tooth contact analysis and calculation of the static transmission error in loaded gears", AIAS 2019 International Conference on Stress Analysis

[5] Bruzzone, F., Maggi, T., Marcellini, C., Rosso, C., "Materiale informative GeDy TrAss"