
Constant Mesh Stiffness Gears

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Abstract

Gearboxes are the most used power transmission mechanisms. Nowadays, the demand for a gear with low vibration and sound levels as well as high efficiency is very high, specially considering the growing use of lightweight materials for mechanical components. The gear mesh stiffness is related to the dynamic transmission error which in turn is closely related to gear noise and vibration. That being said, this work is addressed to the development and dynamic study of constant mesh stiffness gears.

A study on a mesh stiffness model based on the Heaviside functions was performed and from there an analysis of the variation of the length of the gear contact lines was conducted since it has great influence on the gear mesh stiffness. Following that, an algorithm generating constant mesh stiffness gears (MSGears) was developed while taking into account the safety factors and efficiency of gears. The dynamic performance of the newly developed gear geometries was evaluated with a single degree of freedom dynamic model which considered teeth contact loss. The dynamic behavior of the MSGears was compared with other gears by analyzing the dynamic transmission error and the dynamic mesh force.

The analysis of the variation of the length of the contact lines demonstrated that for integer numbers of the contact or overlap ratios there was no variation of the length of the contact lines. Further analysis on the mesh stiffness showed that the application of integer numbers of either the contact ratio or the overlap ratio lead to an approximately constant mesh stiffness. Regarding the MSGears developed, all of them respected the gear design principles. Two kinds of MSGears were developed, one that was more biased towards efficiency and another towards higher safety factors. After evaluating the dynamic performance of the MSGears, the advantages of this new gear design were clear. The dynamic transmission error was much lower and exhibited less variations when comparing with non-MSGears. For the dynamic mesh force the conclusions were similar. The MSGears ended up revealing a huge potential in the reduction of noise, vibration and dynamic overload.

Keywords: Gear Design, Mesh Stiffness, Constant Mesh Stiffness, Gear Dynamics, Dynamic Transmission Error.

To everyone that truly supported me...

“Live as if you were to die tomorrow. Learn as if you were to live forever.”

Mahatma Gandhi

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Contents

Contents	vii
List of Figures	ix
List of Tables	xiii
Nomenclature	xv
1 Introduction	1
1.1 Background	1
1.2 Objectives	2
1.3 Layout	2
2 Literature Review	5
2.1 Gears	5
2.1.1 Considerations for Gear Tooth Design	5
2.1.2 Gear Geometry	6
2.1.3 Spur Gears	10
2.1.4 Helical Gears	10
2.2 Gear Dynamics	11
2.2.1 Models for Gear Dynamics	11
2.2.2 Contact Ratio and Dynamic Loads	18
3 Mesh Stiffness and Load Sharing Models	21
3.1 Description of the contact lines and the coordinate system	22
3.2 Quasi-Static Rigid Model	24
3.2.1 Length of the Contact Lines	25
3.2.2 Load Sharing	26
3.3 Quasi-Static Elastic Model	31
3.4 Analysis of the Contact Length Variation	36
4 Constant Mesh Stiffness Gears Development	41
4.1 CMSG Algorithm Development	41
4.1.1 Algorithm Performance	42
4.2 CMSG Algorithm Results	47
4.2.1 Results	48
5 Dynamic study of the Constant Mesh Stiffness Gears	53
5.1 Dynamic Model	53
5.1.1 Implemented Dynamic Model	55

5.1.2	Software Implementation	58
5.1.3	Validation	60
5.2	Results	63
5.2.1	Dynamic Transmission Error	64
5.2.2	Dynamic Mesh Force	72
5.2.3	Dynamic Contact Analysis	80
6	Conclusions and Future Work	87
6.1	Conclusions	87
6.2	Future Work	87
	References	89
A	Sum of the Unbounded Contact Length Ratio for Helical Gears	93
B	KISSsoft® Reports	95
B.1	CMS55-E	95
B.2	CMS55-R	106
B.3	CMS92-LE	117
B.4	CMS50-E	128
B.5	CMS60-R	139
B.6	C14	150
B.7	H951	161
C	Dynamic Analysis Figures	173
C.1	C14	173
C.2	H501	175
C.3	H951	177
C.4	CMS55-E	179
C.5	CMS55-R	181
C.6	CMS92-LE	183
C.7	CMS50-E	185
C.8	CMS60-R	187

List of Figures

2.1	Zone of tooth action [13].	6
2.2	Basic Gear Geometry [14].	7
2.3	Examples of the involute curve.	8
2.4	Definition of Pitch Circle and Pitch Point [14].	8
2.5	Definition of Circular Pitch [14].	9
2.6	Definition of Pressure Angle [14].	9
2.7	Dynamic model employed by Tuplin (adapted from [15]).	13
2.8	Example of torsional models with tooth compliance. I_i = mass moment of inertia of gear i ; m_{ie} = equivalent mass of inertia I_i ; m_e = equivalent mass of all inertias; k_e = equivalent tooth mesh stiffness; k_m = tooth mesh stiffness; c_m = tooth mesh damping; $e(t)$ = displacement input representing gear errors (adapted from [15]).	14
2.9	Typical models for gear dynamics. I_1, I_4 = mass moment of inertias of prime mover and load; I_2, I_3 = mass moment of inertias of gears; k_{1i} = torsional stiffness of shaft i ; k_m = tooth mesh stiffness; k_2, k_3 = stiffnesses representing lateral flexibility due to shafts and bearings (adapted from [15]).	16
2.10	Dynamic load factors for low-contact-ratio gears [16].	18
2.11	Dynamic load factors for transition from low-contact-ratio gears to high-contact-ratio gears [16].	19
2.12	Dynamic load factors for high-contact-ratio gears [16].	19
3.1	Contact lines for different gear geometries: (a) $\varepsilon_\beta = 0$; (b) $\varepsilon_\beta \leq \varepsilon_\alpha$; (c) $\varepsilon_\beta > \varepsilon_\alpha$ [10].	23
3.2	Definition of ξ	23
3.3	Approximation to the Heaviside function, equation (3.3) [10].	24
3.4	Example of the steps to reach the length of the contact lines for the C40, a spur gear.	27
3.5	Example of the steps to reach the length of the contact lines for the H501, a helical gear.	28
3.6	Comparison between the single contact length ratio and the total contact length for gears C40, H501 and H951.	29
3.7	Comparison between the load sharing functions according to the quasi-static rigid model for gears C40, H501 and H951.	30
3.8	$k_i^u(\xi)$, equation (3.18), ($\alpha_k = \frac{4}{5}$, $i = 0$) for the C40 gear (table 3.1) [10]. . .	32
3.9	Comparison between the single and total mesh stiffness for gears C40, H501 and H951.	34
3.10	Comparison between the load sharing functions according to the quasi-static elastic model for gears C40, H501 and H951.	35

3.11	Contour plot of the RMS of $L(\tau)/L_m$ for a range of profile and transverse contact ratio [20].	37
3.12	Comparison between the contour plot of the ratio length of the contact lines and the average length of the contact lines.	38
3.13	Representations of the RMS of $L^{s,h}(\xi)/\varepsilon_\alpha$ using the Heaviside model.	38
3.14	Representation of the RMS of $L^{s,h}(\xi)/\varepsilon_\alpha$ using the Heaviside model for a spur gear ($\varepsilon_\beta = 0$).	39
3.15	Contour plot of the RMS of $L^{s,h}(\xi)/\varepsilon_\alpha$ emphasizing the integer numbers.	39
4.1	Flowchart of the algorithm developed for the CMSGears.	42
4.2	Contour plot of the RMS of $L^{s,h}(\xi)/\varepsilon_\alpha$ with points from gears in table 4.5.	50
4.3	Representation of the RMS of $L^{s,h}(\xi)/\varepsilon_\alpha$ using the Heaviside model for a spur gear ($\varepsilon_\beta = 0$) pointing out the C40.	51
4.4	Total mesh stiffness for all the CMSGears developed.	52
5.1	Discrete dynamic model of a spur gear pair presented by Tamminana et al. [4].	54
5.2	Single degree of freedom modeling of a two gear system [29].	55
5.3	Dynamic model implemented.	56
5.4	Flowchart of the dynamic model implemented.	58
5.5	Comparison between the RMS of the oscillating DTE component for $T = 100$ Nm (+), $T = 200$ Nm ($\bigcirc$), $T = 300$ Nm ($\square$).	61
5.6	Comparison between the mean DTE for $T = 100$ Nm (+), $T = 200$ Nm ($\bigcirc$), $T = 300$ Nm ($\square$).	62
5.7	Comparison between the mesh stiffness for the gear pair (table 5.1) used in the models.	62
5.8	Difference of the error tolerances in the RMS of the oscillating DTE component.	64
5.9	Analysis of the DTE fluctuations in order to β_r (K_{11}).	65
5.10	Analysis of the DTE fluctuations in order to the rotating speed (K_{11}).	65
5.11	Analysis of the RMS of the oscillating DTE component focusing on low fluctuation gears (K_{11}).	66
5.12	Analysis of the maximum DTE in order to β_r (K_{11}).	67
5.13	Analysis of the maximum DTE in order to the rotating speed (K_{11}).	67
5.14	Analysis of the maximum DTE* in order to β_r (K_{11}).	68
5.15	Analysis of the maximum DTE* in order to the rotating speed (K_{11}).	68
5.16	RMS of the oscillating DTE component for low fluctuation gears and different torques.	69
5.17	Mean DTE for low fluctuation gears and different torques.	70
5.18	Maximum DTE for low fluctuation gears and different torques.	70
5.19	Maximum DTE* for low fluctuation gears and different torques.	70
5.20	Effect of the usage of the contact loss function, $g[x(t)]$	71
5.21	Dynamic mesh force analysis for all gears at K_{11}	72
5.22	Zoom of the mean dynamic mesh force for all gears at K_{11}	73
5.23	RMS of the oscillating DMF component for the low fluctuation gears (K_{11}).	73
5.24	Mean DMF for the low fluctuation gears (K_{11}).	74
5.25	Maximum DMF* for the low fluctuation gears (K_{11}).	74
5.26	Comparison between the static and dynamic normal surface single tooth load $\beta_r = 1$ (K_{11}).	76

5.27	Comparison between the static and dynamic normal surface single tooth load $\beta_r = 1$ (K_6).	77
5.28	Comparison between the static and dynamic normal surface single tooth load $\beta_r = 1$ (K_1).	78
5.29	RMS of the oscillating DMF component for low fluctuation gears and different torques.	79
5.30	Mean DMF for low fluctuation gears and different torques.	79
5.31	Effect of the contact analysis in the dynamic performance of CMS55-E at K_{11}	81
5.32	Effect of the contact analysis in the dynamic performance of CMS55-R at K_{11}	81
5.33	Effect of the contact analysis in the dynamic performance of CMS92-LE at K_{11}	82
5.34	Effect of the contact analysis in the dynamic performance of CMS50-E at K_{11}	82
5.35	Effect of the contact analysis in the dynamic performance of CMS60-R at K_{11}	83
5.36	Comparison between the design and the contact analysis mesh stiffness for the CMS55-E.	84
5.37	Comparison between the design and the contact analysis mesh stiffness for the CMS55-R.	84
5.38	Comparison between the design and the contact analysis mesh stiffness for the CMS92-LE.	84
5.39	Comparison between the design and the contact analysis mesh stiffness for the CMS50-E.	85
5.40	Comparison between the design and the contact analysis mesh stiffness for the CMS60-R.	85
C.1	RMS of the oscillating component of dynamic parameters for all three torques for the C14.	173
C.2	Mean dynamic parameters for all three torques for the C14.	173
C.3	RMS of dynamic parameters for all three torques for the C14.	174
C.4	Maximum dynamic parameters for all three torques for the C14.	174
C.5	Maximum normalized dynamic parameters for all three torques for the C14.	174
C.6	RMS of the oscillating component of dynamic parameters for all three torques for the H501.	175
C.7	Mean dynamic parameters for all three torques for the H501.	175
C.8	RMS of dynamic parameters for all three torques for the H501.	176
C.9	Maximum dynamic parameters for all three torques for the H501.	176
C.10	Maximum normalized dynamic parameters for all three torques for the H501.	176
C.11	RMS of the oscillating component of dynamic parameters for all three torques for the H951.	177
C.12	Mean dynamic parameters for all three torques for the H951.	177
C.13	RMS of dynamic parameters for all three torques for the H951.	178
C.14	Maximum dynamic parameters for all three torques for the H951.	178
C.15	Maximum normalized dynamic parameters for all three torques for the H951.	178
C.16	RMS of the oscillating component of dynamic parameters for all three torques for the CMS55-E.	179
C.17	Mean dynamic parameters for all three torques for the CMS55-E.	179
C.18	RMS of dynamic parameters for all three torques for the CMS55-E.	180

C.19 Maximum dynamic parameters for all three torques for the CMS55-E. . . .	180
C.20 Maximum normalized dynamic parameters for all three torques for the CMS55-E.	180
C.21 RMS of the oscillating component of dynamic parameters for all three torques for the CMS55-R.	181
C.22 Mean dynamic parameters for all three torques for the CMS55-R.	181
C.23 RMS of dynamic parameters for all three torques for the CMS55-R. . . .	182
C.24 Maximum dynamic parameters for all three torques for the CMS55-R. . . .	182
C.25 Maximum normalized dynamic parameters for all three torques for the CMS55-R.	182
C.26 RMS of the oscillating component of dynamic parameters for all three torques for the CMS92-LE.	183
C.27 Mean dynamic parameters for all three torques for the CMS92-LE.	183
C.28 RMS of dynamic parameters for all three torques for the CMS92-LE. . . .	184
C.29 Maximum dynamic parameters for all three torques for the CMS92-LE. . .	184
C.30 Maximum normalized dynamic parameters for all three torques for the CMS92-LE.	184
C.31 RMS of the oscillating component of dynamic parameters for all three torques for the CMS50-E.	185
C.32 Mean dynamic parameters for all three torques for the CMS50-E.	185
C.33 RMS of dynamic parameters for all three torques for the CMS50-E.	186
C.34 Maximum dynamic parameters for all three torques for the CMS50-E. . . .	186
C.35 Maximum normalized dynamic parameters for all three torques for the CMS50-E.	186
C.36 RMS of the oscillating component of dynamic parameters for all three torques for the CMS60-R.	187
C.37 Mean dynamic parameters for all three torques for the CMS60-R.	187
C.38 RMS of dynamic parameters for all three torques for the CMS60-R. . . .	188
C.39 Maximum dynamic parameters for all three torques for the CMS60-R. . . .	188
C.40 Maximum normalized dynamic parameters for all three torques for the CMS60-R.	188

List of Tables

3.1	Geometrical parameters of the C40, H501 and H951 gears	22
4.1	Parameters required for the definition of gears in the program	46
4.2	Parameters imposed for the development of the CMSGears	48
4.3	Calculated Safety Factors of the gear FZG C14 with the algorithm	48
4.4	Output parameters for the CMSGears developed	49
4.5	Contact and overlap ratios for the CMSGears	50
4.6	Gear loss factors for all gears	51
5.1	Design parameters of the spur gear pair used in [29]	61
5.2	Torques applied to the pinion in the dynamic model	63
5.3	Natural frequency and respective pinion rotating speed	64
5.4	Contact and overlap ratios of the contact analysis for the CMSGears	80

Nomenclature

Symbol	Units	Description
a	mm	Center distance
a'	mm	Working center distance
ab	mm	Effective length of the line of action
b	mm	Gear face width
CMSGear	-	Acronym for Constant Mesh Stiffness Gear
d	mm	Pitch circle diameter
d_b	mm	Base diameter
DTE	μm	Acronym for Dynamic Transmission Error
DMF	N	Acronym for Dynamic Mesh Force
$e(t)$	μm	Input displacement (external excitation)
$f_N(\xi)$	N/mm	Normal force per length
F_{bn}	N	Total tooth normal force (normal plane)
F_{bt}	N	Total tooth normal force (transverse plane)
$F_{Ni}(\xi)$	N	Surface normal force (tooth pair)
h_{fP}^*	-	Dedendum of basic rack profile per module
H_V	-	Gear loss factor
$H(\xi)$	-	Heaviside function
I_i	kgm^2	Polar mass moment of inertia of gear i
K_{max}^{ISO}	N/m	Maximum single stiffness (ISO)
K_A	-	Application factor
$l_i^{s,h}(\xi)$	-	Length of a single line of contact
$L^{s,h}(\xi)$	-	Sum of the lengths of the lines of contact
m	mm	Module
M_w	Nm	Torque applied in the gear w
n_1	rpm	Pinion rotating speed
ODE	-	Acronym for Ordinary differential equation
p	mm	Circular pitch
p_{bt}	mm	Transverse base pitch
p_{VZP}	W/m	Evolution of gears load losses per unit of contact length
P_{IN}	W	Transmitted power (gear mesh)
P_{VZP}^{single}	W	Evolution of the gears load losses for a single tooth pair
P_{VZP}^{total}	W	Evolution of the total gear load losses
P_{VZP}^G	W	Average gear load losses (integral method)
r_{bw}	m	Base radius of gear w
R_a	μm	Arithmetic average roughness
R_z	μm	Mean peak-to-valley roughness

R_X	mm	Equivalent radius of curvature at pitch point
RMS	-	Acronym for root mean square
$S_{F1,2}$	-	Tooth bending safety factor of pinion and wheel
S_H	-	Surface durability safety factor
T_1	Nm	Pinion torque
$Tl^{s,h}(\xi)$	-	Trim function
T_{mesh}	s	Mesh period
u	-	Velocity ratio or gear ratio
$U_{1,2}$	m/s	velocity at pitch point
$U_i^{s,h}(\xi)$	-	Unbounded contact line length ratio
v_g	m/s	Sliding velocity
x_i	-	Profile shift coefficient of gear i
X_L	-	Lubricant parameter
z_i	-	Number of teeth of gear i
α	rad	Pressure angle
α_t	rad	Transverse pressure angle
α'_t	rad	Working transverse pressure angle
β	rad	Helix angle
β_b	rad	Base helix angle
β_r	-	Frequency ratio
ε_α	-	Contact ratio
ε_β	-	Overlap ratio
ζ	-	Damping ratio
η_0	mPas	Dynamic viscosity at operating temperature
θ_i	rad	Angular displacement of gear i
μ_{mz}	-	Average coefficient of friction
μ_{mz}^S	-	Average coefficient of friction (Schlenk)
ξ	-	Non-dimensional coordinate along the path of contact
ρ_{fP}^*	-	Root fillet radius of the basic rack per module for cylindrical gears
ω_i	rad/s	Rotational speed

Introduction

1.1 Background

Gears are critical components in power transmission systems and can be found in many different fields, such as automotive, aerospace, agriculture and wind generation [1; 2]. When gear transmissions are operating, the dynamic behavior of their components is revealed. There are two main reasons that justify the importance of the dynamic behavior of gear systems. The first one is the gear durability and the second is the generated noise. The dynamic behavior is taken into account for the gear design as the forces acting on gear transmissions may be larger than under static conditions. In other words, the vibratory performance of a gear pair affects the stresses to which the gear set is submitted to and its corresponding fatigue life. The noise generated by a working gear pair is related to the vibration of these elements. The vibration starts with the excitation of the tooth mesh fluctuations during gear meshing. Inoue et al. [3] states that the vibration energy is transmitted to the gearbox housing through the shafts and bearings and at that point the gearbox housing radiates the structure-borne noise [3–5].

The noise generated by gear transmissions has always been a critical issue in the development of these mechanical components. According to Doğan et al. [6], in a automotive transmission, the rolling contact noise from gear pairs under loading is the most dominant type of transmission noise, revealing the importance of these components in noise generation. However, the gear noise and vibration is not just a matter of performance, it is also about the comfort and health. For example, in the automotive industry reducing the rattling and clattering is important for a comfortable car design [7]. Furthermore, some studies [8; 9] state that noise is an issue that affects a large number of people, specially in urban areas where the traffic noise is the main environmental pollutant. Moreover, they exhibit evidence that environmental noise can not only be taken as a quality-of-life issue as it also affects people’s health. Many diseases, such as sleep disturbance, mood disorders and, the most common, depression can be associated with environmental noise.

In a study on gear dynamics and power loss Marques [10] showed that gears with less mesh stiffness variations are less affected by the dynamic self-excitations. Marques [10] also showed that dynamic excitation can influence the average power loss and, eventually, the calculations of the lubricant parameter (X_L). This means that to perform power loss studies where the goal is to analyze the influence of the lubricant and not of the gear, a constant mesh stiffness gear with relatively high gear loss factor would be ideal due to the smoother working conditions and the ease in the measurement of the experimental torque loss.

Concluding, taking the aforesaid into account, if the mesh stiffness fluctuations are

reduced/eliminated not only the gear noise but also the dynamic performance and the gears' lifespan would be improved.

1.2 Objectives

Gear noise and vibration are associated to the transmission error and all of this connects back to the mesh stiffness. The transmission error fluctuation is, according to Faggioni et al. [11], the difference between the theoretical relative position of two unloaded gears (without manufacturing errors) and the relative position of a gear pair under operating conditions. When the transmission error is related to the gear dynamics it is called dynamic transmission error (DTE) and is affected by the gear manufacturing errors, the torque applied and the rotational speed [2].

The main purpose of this work is centered around the development of constant mesh stiffness gears (CMSGears) aiming for low/constant transmission error while maximizing energetic efficiency. To fulfill this main objective, the items of the following list need to be achieved:

- Study of the modeling of gear mesh stiffness;
- Understand how constant mesh stiffness can be imposed;
- Develop Constant Mesh Stiffness Gears (CMSGear) considering the safety factors and the gear efficiency;
- Comprehend the dynamic effects in the operation of a gear pair;
- Implement a dynamic model to study the dynamic performance of the new (CMS) gear geometry.

1.3 Layout

This work is divided into six chapters and a short description of each one of them is performed as it follows:

Chapter 1: Introduction - introductory notes describing the context in which this work is inserted, the objectives and the organization of this dissertation;

Chapter 2: Literature Review - review of several concepts about gear geometry and design as well as a background history of the dynamic models throughout the time. Also, the analysis of the dynamic loads on a spur gear pair is performed;

Chapter 3: Mesh Stiffness and Load Sharing Models - study of a mesh stiffness model that resorts to the Heaviside functions and load sharing functions obtained from the respective model. Moreover, a survey on the variation of the length of the contact lines in a gear pair is also developed;

Chapter 4: Constant Mesh Stiffness Gears Development - description of the program developed for the generation of the new gear geometry and presentation of the CMSGears;

Chapter 5: Dynamic Study of the Constant Mesh Stiffness Gears - brief review on two dynamic models. Dynamic model implementation and validation. Evaluation and comparison of the dynamic performance of the CMSGears;

Chapter 6: Conclusions and Future Work - the main conclusions of this work as well as its relevant aspects are exhibited. Finally, some suggestions to improve and to keep investigating the work developed are given.

Literature Review

2.1 Gears

“As engrenagens constituem o verdadeiro *ex-libris* da Engenharia Mecânica - ubíquas, encontram-se em toda a parte onde o homem chegue.” by Professor Paulo Tavares de Castro [12]. This section will focus on some concepts and definitions of gears that are crucial for the understanding of further subjects in this study. Thus, it is going to be centered in spur and helical gears.

2.1.1 Considerations for Gear Tooth Design

In order for gears to drive in a specified direction and to transmit power or motion smoothly and efficiently, the active profiles of the teeth on mating gears need to satisfy not only the continuity of action but also the conjugate action [13]. These conditions (continuity of action and conjugate action) are satisfied by accurately selecting the gear tooth proportions [13].

Continuity of Action

“Line of action” is where theoretically the gear teeth meshing takes action. The shape of the active profile gear teeth and the outside diameter of the gears are the main parameters that define the theoretical line of action. While the first defines its shape, the second controls the length of the lines of action [13].

For spur gears, during the meshing cycle the load is carried out by one or two pairs of teeth. The transition has to develop in a way that the second pair of teeth gets hold of its share of the load and prepares to take over the full load before the first pair of teeth leaves the line of action. This guarantees that there is a continuous flow of power as at least one pair of teeth is always in contact [13].

Continuity of action is established if the effective length of the line of action, ab (interception of the addendum diameter (outside diameter) with the line of action), is longer than the transverse base pitch (p_{bt}), see figure 2.1. Note that the transverse base pitch for spur gears is equal to the base pitch. The ratio of the aforementioned parameters is a numerical index of the continuity of action and it is named contact ratio, ε_α (equation (2.1)). The contact ratio can be interpreted as the average number of teeth in contact [13].

$$\varepsilon_\alpha = \frac{ab}{p_{bt}} \geq 1.2 \quad (2.1)$$

According to the American Gear Manufacturers Association (AGMA), the contact ratio for spur gears should not be less than 1.2, meaning that the transverse base pitch must be exceeded from 20% to 40% by the effective length of the line of action [13].

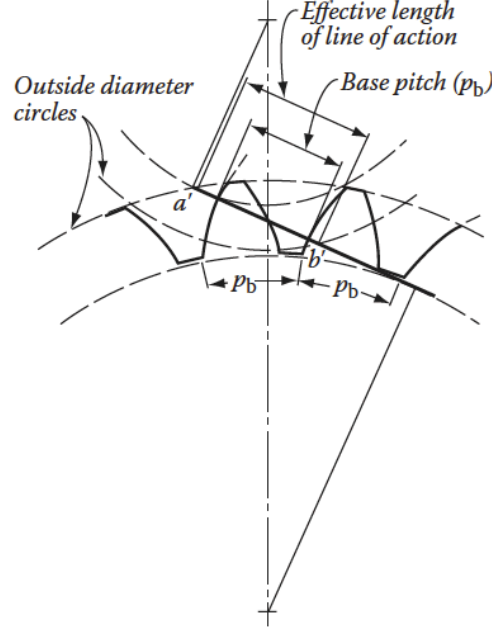


Figure 2.1: Zone of tooth action [13].

Conjugate Action

Conjugate curves are defined as curves that while interacting with each other impart power or motion smoothly with a constant driving ratio. The shape of these curves must satisfy the Willis' "basic law of gearing", which states, "normals to the profile of mating teeth must, at all points, pass through a fixed point located on the line of centers" [13].

For spur and helical gears, the most commonly used curves are from the involute family. The involute curve meets all the requirements of conjugate curves. The line of action formed from this type of curve is normal to both the driving and the driven involutes at all possible points of contact and passes through the pitch point [13]. Additionally, all contact occurs along the line of action [13].

From what was previously mentioned it can be concluded that the fixed point referred in the basic law of gearing is, in the case of involute curves, the pitch point [13].

2.1.2 Gear Geometry

The basis of gear geometry are displayed with the resort to spur gears for two main reasons. The first is that it is the simplest gear and therefore the most intelligible. The second one is because spur gears are the most commonly used gears [14].

Figure 2.2 illustrates the fundamental elements of a spur gear mesh. The crucial characteristics of a gear mesh are the center distance (a), the pitch circle diameters (pitch diameters (d)), module (m), number of teeth (z) and pressure angle (α) [14].

Along the following sections some of these parameters will be discussed in more detail.

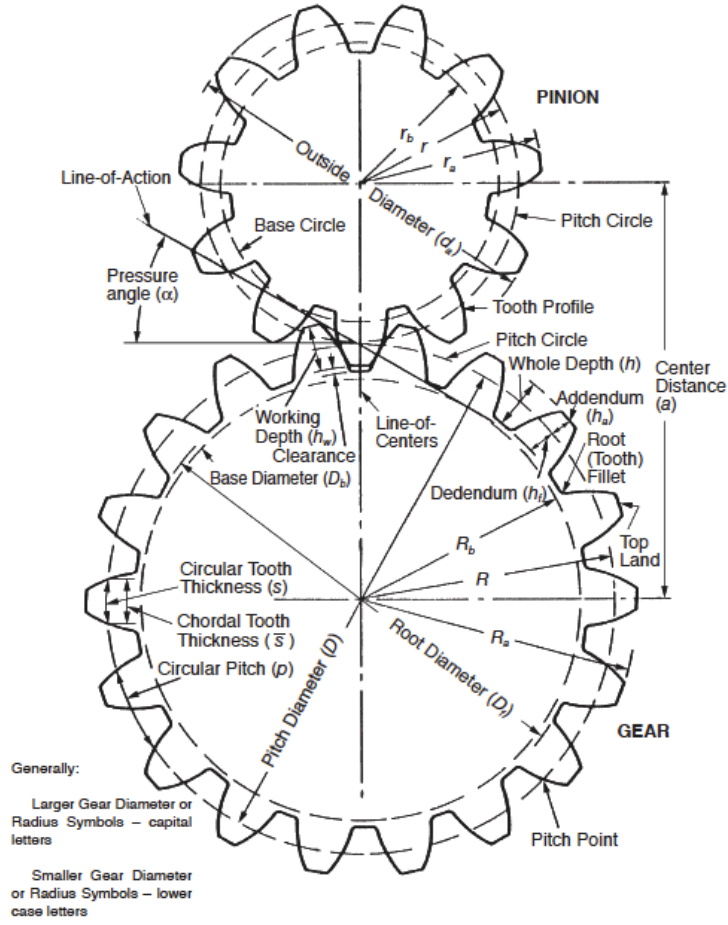


Figure 2.2: Basic Gear Geometry [14].

The Involute Curve

The involute curve is accomplished by tracing the free end of a taut string which is fixed on the other end to a circle. While the taut string is unwinding, the trace point describes the involute form. This phenomenon can be visualized in figure 2.3a.

The properties of the involute curve are (attend on figure 2.3b):

- The curve is generated by point V , complacent with the straight line TV . Since V is the trace point, the arc $\widehat{Q-5}$, measured on the circle, is equal to the straight line ρ_5 (tangent to the circle on 5). The same procedure can be reproduced for every respective pair arc-straight line [12];
- Any normal to the involute curve is tangent to the circle, line ρ_{11} is taken as an exemple [12];
- The radius of curvature of any point of the involute curve is equal to the straight line that passes through that point and tangent to the circle. For instance, the curvature radius of point 11 is ρ_{11} . This means that the circle is the geometric space of the centers of curvature of the involute. The previously defined circle is the base circle of a spur gear and it specifies the tooth profile, therefore it is a fixed characteristic of a gear. [12].

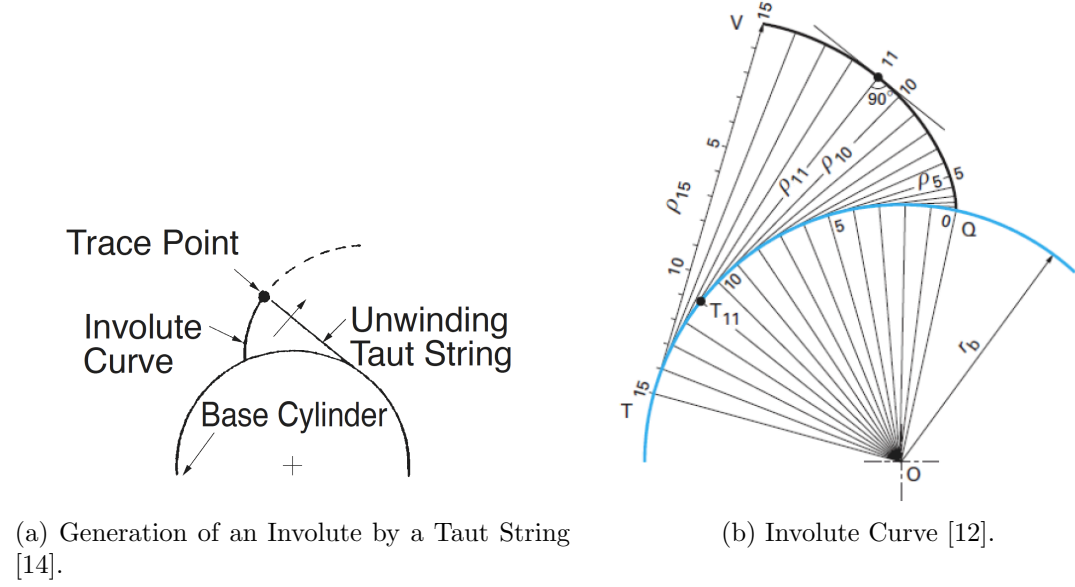


Figure 2.3: Examples of the involute curve.

Pitch Circles

Take notice of figure 2.4, the tangent to both base circles is the line of contact or line of action, as mentioned before. The point of intersection from the line of centers and the line of action is the pitch point, P. The pitch point defines the pitch circles for both gears and the velocity ratio is given by the ratio between the diameter of the pitch circles. The velocity ratio of gear 2 to gear 1 is, $u = d_1/d_2$ [14].

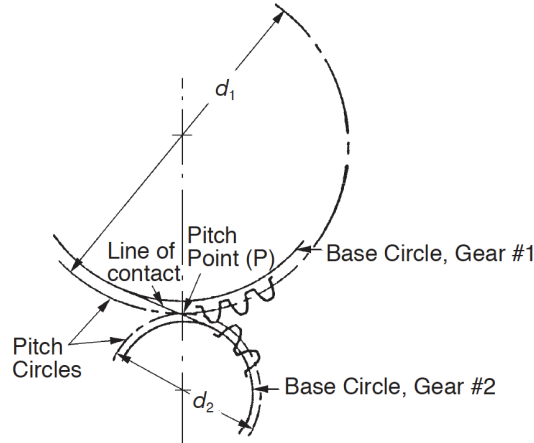


Figure 2.4: Definition of Pitch Circle and Pitch Point [14].

Pitch and Module

One of the most important features of the gear geometry is the size of the teeth. For that, two concepts are introduced, circular pitch and module.

Circular pitch is the length of the arc measured along the pitch circle between two consecutive analogue teeth profiles [14] and is defined in equation (2.2). Figure 2.5 allows a visual interpretation of the previously mentioned parameter.

$$p = \text{circular pitch} = \frac{\text{pitch circle perimeter}}{\text{number of teeth}} = \frac{\pi d}{z} \quad (2.2)$$

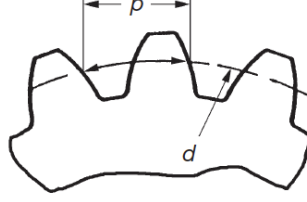


Figure 2.5: Definition of Circular Pitch [14].

The module is the length of the pitch circle corresponding to each teeth [14], equation (2.3).

$$m = \frac{d}{z} \quad (2.3)$$

From the formerly stated, the relationship between the circular pitch and the module can be deduced, equation (2.4).

$$pz = \pi d \Rightarrow \frac{d}{z} = \frac{p}{\pi} = m \quad (2.4)$$

Pressure Angle

The angle formed between the line of action and a straight line perpendicular to the line of centers is called pressure angle. From figure 2.6, which represents the pressure angle, it is possible to state that the pressure angle modifies with the center distance [14].

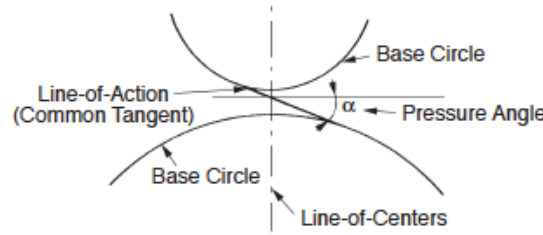


Figure 2.6: Definition of Pressure Angle [14].

Note that the base circle is an innate characteristic of a gear since it is established when the gear is manufactured. Although, that is not true for the pitch circle, whose diameter depends on the kinematics of the gear [12]. The aforementioned concepts lead to the distinction of two pressure angles. The standard pressure angle (related to base circle) is responsible for defining the gear geometry, therefore is immutable after the gear is produced. As for the operating pressure angle (related to the pitch circle), just like the name makes reference to, relies on the working center distance.

Profile Shifting

Profile shifting (or addendum modification) is materialized by moving the rack's reference line relatively to the pitch line. The distance between the formerly referred lines is the

extra feed of gear cutter (xm), where x is the profile shift coefficient [12; 14]. When the line is shifted away from the gear center, it is a positive addendum modification, otherwise it is named negative addendum modification [12].

Thereafter, the consequences of positive profile shifts on some geometric teeth characteristics are going to be exhibited.

Dedendum - reduction of the dedendum equal to the extra feed on the gear cutter (xm) [12];

Addendum - in order to avoid a decrease in the tooth whole depth, which would affect the contact ratio, the addendum is usually increased by xm [12];

Circular tooth thickness - although there is no modification on the pitch diameter, there is an increase in the circular tooth thickness [12].

In case of negative addendum modification, the effects on the previous parameters are the opposite, meaning that, there will be an increase in the dedendum, a decrease in the addendum and a reduction on the circular tooth thickness [12].

Profile shifting can be used for many different reasons, such as, to adjust center distance, to prevent undercutting and to increase the overall gear performance (in general it consists of balancing the specific sliding) [12]. With respect to profile shifting and to adjusting the center distance, using addendum modification does not necessarily mean that the center distance will be affected. If the profile shift is such that $(x_1 + x_2) = 0$, no alteration to the center distance is done. However, if $(x_1 + x_2) \neq 0$, the center distance is influenced according to each of the values of the coefficients.

2.1.3 Spur Gears

As mentioned above, spur gears are the most common and most used type of gears. This type of gears has radial teeth that project outwards from a cylinder and are parallel to the cylinder axis of rotation. The majority of the gears uses a tooth profile with an involute form. Although, there are different types of tooth shapes, for example, cycloidal teeth which are frequently used in precision mechanical clocks [13].

Another parameter that characterizes gear geometry is the pressure angle. For spur gears the most common pressure angles are 14.5° , 20° and 25° , however, the 14.5° pressure angle is mostly resorted to special designs and for some replacement gears. The advantage of lower pressure angles is that it leads to a smoother and quieter meshing owing to the fact of having larger profile contact ratios. Despite that, lower pressure angle gears have lower bending strength ratings, lower surface durability ratings as well as higher sliding velocities [13].

On the other hand, higher pressure angles give rise to a better loading-carrying capacity regarding both strength and durability. Besides that, it also reduces the sliding velocity and consequently increases the scoring and wear resistance [13].

Spur gears are the easiest gears to manufacture and therefore are, usually, the least expensive type of gears [13].

2.1.4 Helical Gears

Helical gears are cylindrical gears with helical teeth. As in helical gears the teeth mesh gradually, this is, the meshing teeth enter the path of contact progressively, they are smoother and quieter than spur gears [10; 13]. Although, in addition to the radial and

tangential forces present in spur gears, helical gears introduce an end thrust load along the rotating axis [13].

Helical gears can be seen as a series of staggered spur gears with infinitesimal width [10; 13]. In this type of gears, as a resemblance to spur gears, helical teeth are normally shaped with an involute form [13]. As previously stated, helical gears are quieter than spur gears, however, this phenomenon is only significant if the overlap ratio is equal or higher than the unit. In that condition, the gear faces are advanced at least one circular pitch from one another [13].

The practical range for helix angles is from a few degrees to around 45° and as it increases the noise level is generally decreased and the load capacity increased. It should be noted that for helix angles that largely exceed the interval 15° to 20° , the transverse tooth thickness is reduced giving rise to a decrease in the tooth bending capacity [13].

Finally, two conditions must be satisfied for external axis helical gears to mesh. On one hand, they must have the same helix angle. On the other hand, they have to be of different hand [13].

2.2 Gear Dynamics

This section is going to cover the research on the gear dynamics, more precisely, a review on different mathematical models used to describe the vibratory behavior of gears and a detailed overview of a work that investigates the influence of the contact ratio on the dynamic loads of spur gears.

In the review of the dynamic models, a series of dynamic models from different articles are analyzed and a brief description pointing out the most relevant aspects is made by Özgüven and Houser [15].

In the evaluation of the effect of the contact ratio on the gear dynamic loads performed by Liou et al. [16], a computer simulation was ran for a wide range of contact ratio gears to calculate the dynamic load factor. Additionally, the impact of the rotating speed is also investigated.

Even though the works mentioned were done many years ago, that does not mean they are outdated. The concepts presented in each article are fundamental for the comprehension of many effects and are essential for the development and assessment of the dynamic performance of gears.

2.2.1 Models for Gear Dynamics

There is an extensive amount of studies about gear dynamics and dynamic modeling of gear systems. Each model is developed considering the purpose of the study and therefore certain assumptions are made and some effects are or not included, depending on the parameters that are supposed to be analyzed and on the experimental test rig available [1; 15].

Dynamic modeling is resorted to accomplish different objectives, which can vary from the study of stresses (bending stresses and contact stresses), pitting and scoring, radiated noise, vibration, stability regions, transmission error and many others [15].

Özgüven and Houser [15] sorted dynamic models into five classes, where each class basic characteristics, objectives and parameters considered are as follows:

1. Simple Dynamic Factor Models: The early studies of the gear dynamics were mainly to achieve a dynamic factor in order to calculate the gear root stress. At that time,

the investigations' procedures were empirical and semi-empirical and simple dynamic models were applied [15].

2. Models with tooth compliance: There is a vast number of models which only include the tooth stiffness as the potential energy storing element in the system. In this group, the flexibility of all other elements, such as shafts and bearings, are disregarded. As the tooth compliance models do not take into consideration many elements and/or effects, in this kind of studies the systems are normally modelled as a single degree of freedom spring-mass system. Some models in this group are very simple and are developed just for the determination of the dynamic factor, creating an overlap between this group and the previous one [15].
3. Models for Gear Dynamics: Besides including the tooth compliance, the models for gear dynamics also include the flexibility of other elements. The main effects analyzed in these studies are the torsional flexibility of shafts and the lateral flexibility of the bearings and shafts along the line of action [15].
4. Models for Geared Rotor Dynamics: This group of models account for the shaft whirling since it allows the transverse vibration in two mutually perpendicular directions of the gear carrying shaft. The geared rotor dynamics models are usually modelled through the torsional vibration of the system [15].
5. Models for Torsional Vibrations: Unlike the previous models, this group neglects the flexibility of gear teeth and therefore can be treated as a pure torsional vibration model. The model is generally composed by torsional flexible shafts with rigid gears [15].

Simple Dynamic factor models

As formerly mentioned, this group is mainly focused on the determination of the dynamic factor. The study of the dynamic factor was the aim of the first dynamic gear models in history so that it could aid in the design of high speed gears. When gears are in mesh their tooth load can be divided into two major components, one of them is a static component originated from the transmitted torque and the other one is the dynamic component [15].

Walker, in 1868, was the first to come up with the definition of the dynamic factor, which, at that time, was named speed factor [15].

$$DF = SL/DL, \quad (2.5)$$

where DF is the dynamic factor, SL is the static load and DL is the dynamic load. Several other empirical approaches to define the dynamic factor were made and the DF was defined in order to the pitch line velocity. However, later, it was discovered that, besides the pitch line velocity, both tooth error and the effect of gear and pinion inertias were relevant and should be included in the determination of the dynamic gear forces. In 1931, Buckingham (quoted in [15]) published studies showing that the effective masses, effective errors and gear speed strongly affected the tooth load [15].

The use of vibratory models for the dynamic analysis of gears started a new period in the study of gear dynamics. In 1950, Tuplin (quoted in [15]) introduced the first spring-mass model. The model employed, as it can be seen in figure 2.7, consists of an equivalent constant tooth mesh stiffness (k_e), an equivalent mass (m_e) and a transmitted load (w). In addition, with the insertion and removal of a wedge at the base of the spring, gear errors can be modeled. Many different shapes were attributed to the wedge in order to evaluate

the effect of different types of errors. Due to the simplicity of this model, it can only be applied for an approximation of the dynamic factor for speeds far below the resonant speed.

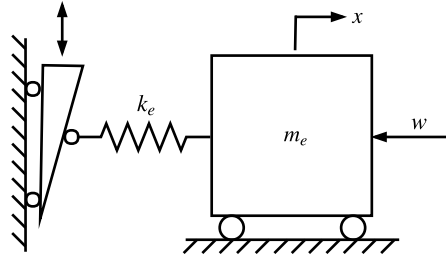


Figure 2.7: Dynamic model employed by Tuplin (adapted from [15]).

Strauch, in 1953, (quoted in [15]) considered a periodic excitation in his model. The changes in the mesh stiffness caused by the variation from one to two pair of teeth in contact were taken into account as a step change in the mesh stiffness. In his work he studied the forced vibrations from the continuous error between two unmodified involute gears. This work appears to be the first to consider a periodic excitation.

In 1970, Seireg and Houser (quoted in [15]), developed a formula for the dynamic tooth load. This semi-empirical formula was based on previous experimental tests done by the authors and to achieve it they used a geometrical analysis of the tooth meshing action. Their dynamic tooth load is valid for both spur and helical gears and considers the gear geometries, manufacturing errors and the operating conditions such as load and speed. The model was only meant for conditions below the system resonance.

Models with tooth compliance

The models covered by this group assume that the only compliance in the model regards the tooth mesh stiffness since all the remaining elements are considered to be perfectly rigid. There are two types of models in this group, it can either be a translational model (figures 2.8b and 2.8c) if the translational motion of the tooth is modeled, or a torsional model (figures 2.8a and 2.8d) in case the rotational motion of the gear is modeled. While the first kind of model allows to study the forced vibrations of the teeth by modeling the gear tooth as a cantilever beam, the second one enables the study of the torsional vibrations of meshing teeth. However, in both of this type of models, transmission error excitation is reproduced by a displacement excitation at the mesh [15].

The model developed in 1956 by Nakada and Utagawa (quoted in [15]) has a time-varying stiffness which is approximated as a rectangular wave. Their model accounts for the elasticity variations of the gear teeth in mesh. Besides that, the torsional vibration of the gear pair was converted to the equivalent translational vibratory system, as in figures 2.8b and 2.8c for example. Nakada and Utagawa established movement equations for different damping systems and it was assumed that the gear tooth profiles were precisely manufactured.

In the work published by Harris in 1958 (quoted in [15]), the author considered three sources of vibration, namely, manufacturing errors, variation in the tooth stiffness and the loss of tooth contact which caused a non-linearity in the tooth stiffness. Harris' one degree of freedom model encompasses all the formerly mentioned sources and treated the excitation as periodic. With his model he investigated the importance of the transmission error and concluded that the behavior of spur gears can be described by static transmission

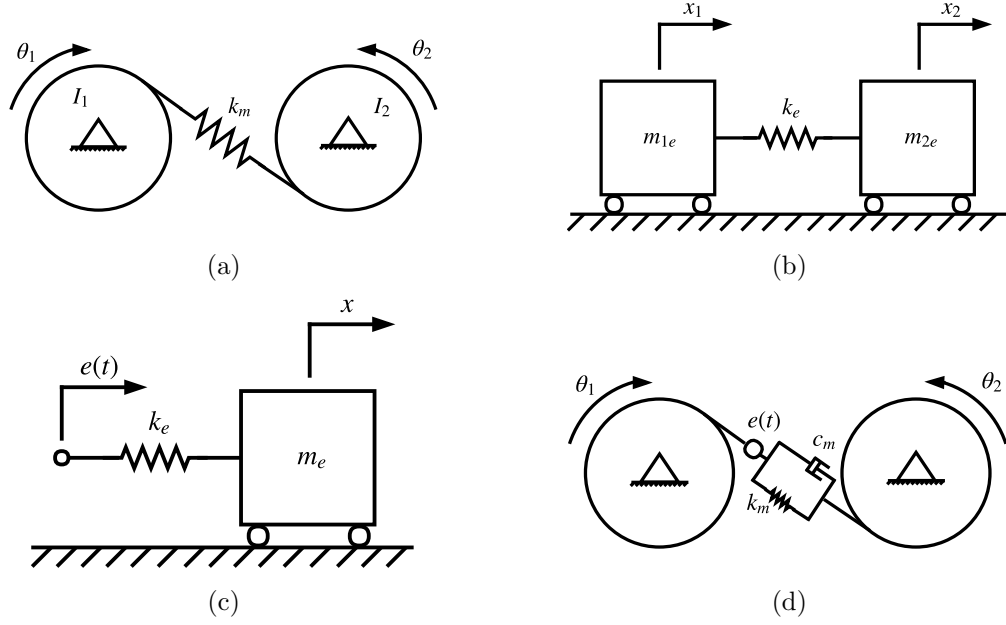


Figure 2.8: Example of torsional models with tooth compliance. I_i = mass moment of inertia of gear i ; m_{ie} = equivalent mass of inertia I_i ; m_e = equivalent mass of all inertias; k_e = equivalent tooth mesh stiffness; k_m = tooth mesh stiffness; c_m = tooth mesh damping; $e(t)$ = displacement input representing gear errors (adapted from [15]).

error curves.

Aida et al. (quoted in [15]) created a model in order to establish stability regions and steady state gear vibrations. This model included the effects of time-varying stiffness and periodic tooth errors. The results obtained were compared with the experimental measurements.

A research on the separation of tooth meshing with a single degree of freedom dynamic model was performed by Nakamura in 1967 (quoted in [15]). Nakamura considered not only tooth errors but also the variation of the number of teeth in contact. While the first effect was modeled as a sinusoidal wave, the second was modeled by implementing in the tooth mesh stiffness a square wave shape, which represents a sudden change of the number of teeth in contact. Nakamura solved the problem numerically and the results revealed that tooth separation occurs at a speed defined by the transmission error and the tangential load. Moreover, he concluded that the largest dynamic load happens right after the teeth lose contact.

In 1967, Bollinger and Harker (quoted in [15]) presented a study on the dynamic instabilities originated from the variation of the mesh stiffness. The single degree of freedom model created for that purpose was composed by an equivalent mass that represented the inertias of both the gear and the pinion and an harmonic mesh stiffness variation. This lead to a damped and forced Mathieu equation which was solved by the application of an analog computer. From this investigation Bollinger and Harker concluded that it is possible to reduce the dynamic load by reducing the tooth mesh stiffness variation or by increasing the system damping.

With the objective of studying the connection between the tooth deformation and the dynamic loads in heavy-loaded spur gears, Ichimaru and Hirano (quoted in [15]) made a gear dynamic model formed by an effective mass of the gear blanks and the gear mesh stiffness. In this model the main excitation is due to the manufacturing errors. The main

difference that makes this investigation stand out is the solution technique adopted which allows to represent the mesh stiffness as a function of the position and simultaneously use a linearized equation. The theoretical results were in quite good agreement when compared with the experimental results.

Cornell and Westervelt [17] took on other models and improved them so that it could be used for contact ratios up to 4. In their model the authors included the system damping, the non-linearity of the gear mesh stiffness, tooth errors, tooth profile modification and assumed that the gears acted like rigid inertias [15; 17]. With this study, it was concluded that the lifespan of a gear can be significantly affected by the operating condition and the dynamic characteristics of the gear system and that the profile modifications affect remarkably the tooth loading and the root stressing, according to the contact ratio, speed and applied load [17]. Finally, it was stated that gear errors influence the tooth loading and root stressing more negatively on high-contact-ratio gears than on low-contact-ratio gears [17].

Another noteworthy investigation was done by Umezawa et al. (quoted in [15]) where the authors used a single degree of freedom torsional vibration model. This torsional model used a periodic gear mesh stiffness, constant damping and accounted for the gear manufacturing errors. The errors in the gears were included by measurement in an automatic gear accuracy measuring instrument which had recently been developed. The problem was solved numerically with a Runge-Kutta-Gill method. The dynamic behavior of the spur gears tested was precisely predicted.

Models for Gear Dynamics

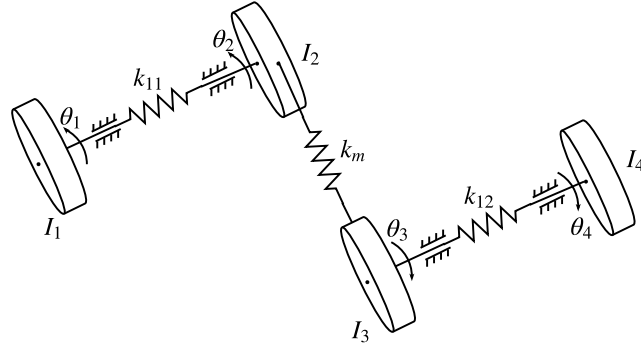
This class differs from the previous ones as the mass and stiffness of other elements, such as shafts and bearings, are included in the models. By considering this components, the model comes closer to describing the dynamic of gears in practical applications. However, the single degree of freedom models mentioned before are, most times, in accordance to the experimental tests, meaning that the assumptions made in the models are fulfilled by the experimental test rig. If a characterization of a more realistic system is required, one needs to include more elements and effects [15].

The models for gear dynamics can be divided into torsional (torsional stiffness of the gear and the remaining elements considered) (figure 2.9a) or torsional and translational (torsional and transverse flexibility of the gear-carrying shafts is included), as in figure 2.9b. Note that the translational vibration can only be considered in one direction, otherwise the shaft would be able to whirl, which, according to the previously established classification, would be a rotor dynamics model (discussed in the next section) [15].

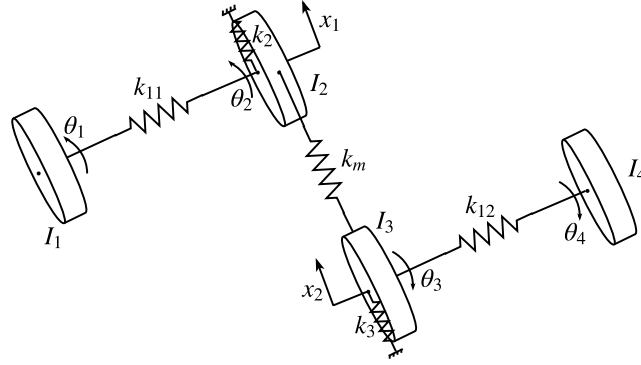
One of the first to include the shafts' vibration into a model was Johnson (quoted in [15]) in 1962. His simple model included a constant mesh stiffness, equal to the average of the varying mesh stiffness, so that a linear system could be obtained. After Johnson's work several authors presented models with the torsional vibration of gear shafts [15].

Kohler, Pratt and Thomson (quoted in [15]) constructed a constant tooth mesh stiffness, six degree of freedom model with four torsional degrees of freedom and two (one for each shaft) translational degrees of freedom in the direction of the tooth force. This model was developed after they concluded from their experiments that noise and dynamic load result from the steady state vibration of the gear system when forced by transmission error.

In 1972, Wang and Morse (quoted in [15]) created a model which considered the shaft and gear web stiffness along with a constant gear tooth stiffness. Two years later, Wang (quoted in [15]) upgraded the previous model so it would incorporate the variation of the



(a) Torsional model (adapted from [15])



(b) Torsional and translational model (adapted from [15])

Figure 2.9: Typical models for gear dynamics. I_1, I_4 = mass moment of inertias of prime mover and load; I_2, I_3 = mass moment of inertias of gears; k_{1i} = torsional stiffness of shaft i ; k_m = tooth mesh stiffness; k_2, k_3 = stiffnesses representing lateral flexibility due to shafts and bearings (adapted from [15]).

tooth stiffness, linear and non-linear damping elements, gear tooth backlash and multi-shock loadings.

Salzer and Smith, in 1975 (quoted in [15]) and Salzer, Smith and Welbourn, in 1977, (quoted in [15]) deliberated about the real time modeling of gearboxes. In both investigations a car gearbox was modeled through a six degree of freedom model with non-linear bearing stiffness, time-varying gear mesh stiffness, loss of tooth contact and spacing gear errors. An analog computer solution was employed and the periodic tooth profile errors were inserted as a rectified sine wave.

In 1975, Rettig (quoted in [15]) proposed simplified formulae for the dynamic factors of a single gear stage for three different regions, that being, subcritical, main resonance and supercritical regions. The system was modeled by a six degree of freedom model, four lateral degrees of freedom all in the same direction and two torsional degrees of freedom. Experimental tests were performed and the results were compared with the model calculations.

In 1980, Kubo and Kiyono (quoted in [15]) developed a helical gear model with torsional and translational degrees of freedom that included shaft stiffness to study the dynamic exciting force originated from different tooth form errors and the periodic fluctuation of the tooth stiffness. Between the five tooth form errors investigated, it was concluded that the less damaging was the convex shape.

Models for Geared Rotor Dynamics

The very first models in this group were not conceived to study the dynamic of gears but the gear-carrying shaft whirling [15].

Although Seireg, in 1966, (quoted in [15]) did not present a model, he studied the whirling of geared shafts experimentally and came to an empirical solution to predict the main resonance frequency of the system.

More experimental tests were performed by Mitchell and Mellen (quoted in [15]) where the authors analyzed the torsional-lateral coupling in a geared high speed rotor system. They reached the conclusion that for the right design of high-performance machinery, mathematical models need to contemplate the effects of torsional-lateral coupling.

In 1981, Hagiwara, Ida and Kikuchi (quoted in [15]) developed a simple model to study the forced vibration of geared shafts supported by journal bearings. The source of the excitation force was considered to be the unbalanced forces and run-out errors. The model takes into account the lateral flexibility of the shafts (by applying discrete stiffness values), constant mesh gear tooth stiffness and disregards the gear backlash and tooth separation. In this investigation it was detected that unbalanced forces and gear errors can stimulate the lateral and specially the torsional modes, leading to large displacements.

Models for Torsional Vibrations

This group of models consists of purely torsional vibration models as the gear tooth mesh is disregarded and, therefore, only the shaft stiffness and the gear and shaft inertias are usually represented. Due to their assumptions, this kind of models were mainly used to identify the natural frequencies of multi-gear-shafts systems [15].

Despite the previous statement, some research workers used purely torsional vibration models to calculate the dynamic loads and the influence of gear errors on the dynamic performance of these systems. For instance, Rieger (quoted in [15]) used this type of model for a drive train. Besides neglecting the gear tooth flexibility, the author also neglected the gear's inertia. The objective of his investigation was to study the influence of different types of errors [15].

Radzimovsky and Mirarefi (quoted in [15]) developed a model for a gear experimental rig to evaluate the impact of different factors on the coefficient of friction and in the efficiency of gear drives. Even though it was assumed that there were no geometric errors and there was an equal load sharing between the teeth in mesh, the results obtained were similar to the experimental tests.

The rigid gear tooth assumption was also used by Ikeda and Muto (quoted in [15]). The authors created a single degree of freedom model which considered the gear inertias, the torsional shaft stiffness and damping. Their aim was to investigate the gear vibrations originated by the transmission error and the teeth friction. There was a close correlation between the calculations and the experimental results.

Two models with rigid gear teeth were developed by Wang (quoted in [15]). While one of them was a two mass model with no spring, the other was a three mass model with an elastic element. However, both models included the effects of time-varying backlash, impact and displacement excitation. The dynamic loads created by the backlash impact were calculated through a linear iteration technique. As for the results obtained, they were in accordance with the experimental tests and it was proved that gears might be subjected to high tooth loads while under lightly loaded conditions.

2.2.2 Contact Ratio and Dynamic Loads

Liou et al. [16] studied the influence of the contact ratio (CR) on the dynamic load of a spur gear transmission using a computer simulation. The contact ratio was varied between 1.20 and 2.40 meaning that both low-contact-ratio and high-contact-ratio gears were analyzed. This contact ratio range was achieved by only changing the length of the tooth addendum, this is the most effective way to control the contact ratio without increasing the tooth stress. The gears had no profile modification.

A major source of vibration excitation in gears is the variation of the meshing stiffness which is influenced by the contact ratio. Therefore, as the contact ratio affects the meshing stiffness, it also has consequences on the gear dynamics [16].

The dynamic load factor for low-contact-ratio gears (LCRG), $CR = 1.668, 1.754$ and 1.868 , showed an identical configuration although there is a reduction in the dynamic load factor due to the increase of the contact ratio, attend on figure 2.10. Furthermore, there are some peaks in the dynamic load when the system is near resonant speed and its submultiples. At high speeds the dynamic load is lower since the time of single tooth contact is so small that the system can not react while it is being excited [16].

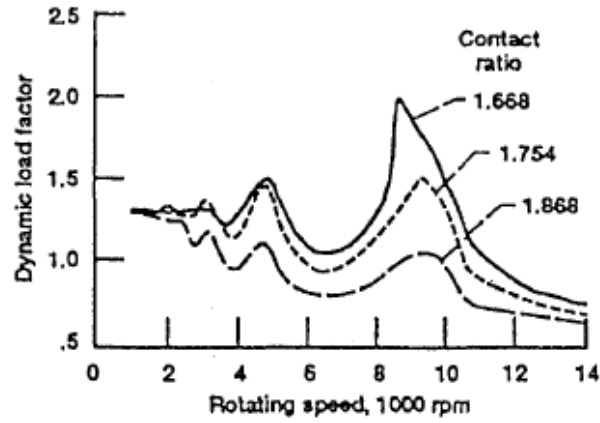


Figure 2.10: Dynamic load factors for low-contact-ratio gears [16].

In regard to “transition” gears ($CR = 1.952, 2.000$ and 2.145) the dynamic load factor can be analyzed from figure 2.11. The gear with the lowest contact ratio had a similar trend to the low-contact-ratio gears, but in the other cases there are some modifications in the systems response. At a CR equal to 2 the dynamic response is very subtle as it can be seen in figure 2.11 because there are only small variations on the mesh stiffness during tooth contact. For a contact ratio of 2.145 there is a variation in the number of teeth in contact (from double to triple tooth contact) that consequently leads to a variation in the meshing stiffness. Thus, it can be noticed some dynamic effects for lower speeds (under 5000 rpm) which make the dynamic load factor higher than the one at contact ratio 2 [16].

For high-contact-ratio gears ($CR = 2.226, 2.306$ and 2.412) the dynamic load factor curves are compared in figure 2.12. From its analysis it is clear that the curves have different configuration at the resonant speed of the gear (roughly 9300 rpm) and its submultiples (at approximately 4630 and 3100 rpm). In this particular case, a higher contact ratio does not necessarily mean a lower dynamic load factor. The lower the contact ratio, the higher the dynamic load factor at submultiples of the resonant speed. However, the gears with the highest contact ratio undergo the highest dynamic load at resonant speed. Liou et al. [16] believe that this effect occurs thanks to the transition from double to triple

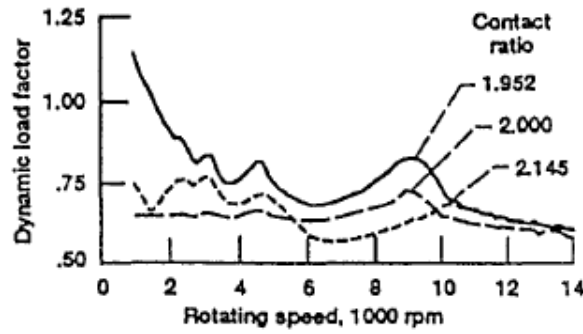


Figure 2.11: Dynamic load factors for transition from low-contact-ratio gears to high-contact-ratio gears [16].

tooth contact. For the gear with the lowest contact ratio ($CR = 2.226$) the band of triple tooth contact is shorter than the other two cases ($CR = 2.306$ and 2.412). The transition of the number of teeth in contact changes the meshing stiffness, which, in turn, acts like a short duration impulse. This excitation is more effective at lower speeds than at higher speeds [16].

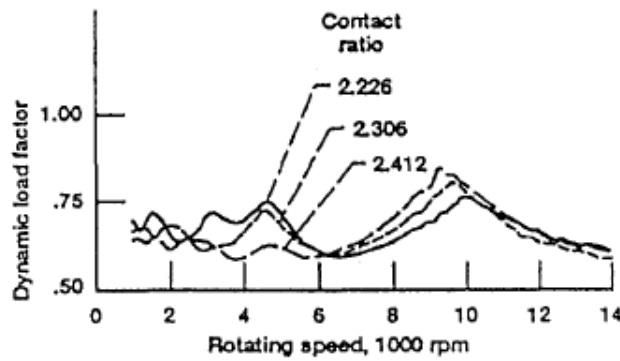


Figure 2.12: Dynamic load factors for high-contact-ratio gears [16].

From this investigation the authors concluded that the dynamic load is significantly lower for high-contact-ratio gears than for low-contact-ratio gears. Moreover, a contact ratio close to 2 minimizes dynamic load over a wide range of operating speeds. Additionally, for LCRG in general, increasing the contact ratio leads to a reduction in the dynamic load. As for HCRG the value that minimizes the contact ratio depends on the operating speed. Finally, for speeds over the natural resonant speed of the gear system, the effect of the contact ratio is less significant on the dynamic load [16].

Mesh Stiffness and Load Sharing Models

The idea of the CMSGears came up while Marques [10] was studying the mesh stiffness of different gears with the model he developed. So, in order to understand how this type of gears appeared, it is fundamental do perceive the ideas behind the load sharing models.

An accurate load sharing model is essential to design a gear since calculations on the gear capacity and efficiency have to be made [18]. Thus, the higher the model accuracy, the more trustworthy the calculations are.

Besides that, the simpler the model, the faster are the calculations and the acquisition of results. The simplicity of a model is highlighted specially while using iterative processes. In these cases, as the same calculations have to be made over and over again, with a simple model computation time can be drastically reduced [18].

Summing up, a good model has to be capable to obtain reliable results in the simplest way possible, this is, as fast as possible.

The load sharing models presented ahead consider rigid and elastic teeth but neglect both dynamic and Hertzian effects. That being said, Marques developed three load sharing models:

1. Quasi-Static Rigid Model (analytical);
2. Quasi-Static Elastic Model (analytical);
3. Quasi-Static Local Elastic Model (numerical-analytical).

The **quasi-static rigid model** assumes that the load per unit of length along the line of the contact over a tooth is constant for a given position in the path of contact. In addition, at a given position, the load per unit of length is the same for all meshing tooth pairs. With these assumptions, it was concluded that the load distribution along the path of contact is inversely proportional to the sum of the lengths of the lines in contact. The length of lines in contact was described trough the use of an approximation of the Heaviside step function [10].

The **quasi-static elastic model** assumes that for a certain position in the path of contact in a line of contact over a tooth, the load per unit of length is constant. Despite that, alike the quasi-static rigid model, at a given position, the load per unit of length is not the same for all meshing tooth pairs. This formulation applies the definition of stiffness and ,with an extra assumption, which is considering the gear is rigid until the base cylinder, combines the Heaviside function with the single tooth pair mesh stiffness to acquire the load sharing distribution [10].

Although the quasi-static local elastic model is not going to be employed in this work, it is still going to be briefly described. The **quasi-static local elastic model** was established through the minimization of the total potential energy of the gear system. This model is applied to the gear face width and the compliance coefficients are extracted with an open source Finite Element Method solver supported by a custom code [10].

The aforementioned models were tested for three gear geometries, those are, the C40, the H501 and the H951. Their geometrical parameters can be viewed in table 3.1. It is noteworthy here that the C40 gear is like an FZG type C gear, however, its face width is 40 mm. Furthermore, the C40 and H501 gears are spur and helical gears, respectively, and the H951 gear is a “low loss” (high efficiency) helical gear pair [10].

Table 3.1: Geometrical parameters of the C40, H501 and H951 gears

Gear type:	C40*		H501		H951	
	Driving	Driven	Driving	Driven	Driving	Driven
Number of teeth (z_i), [-]	16	24	20	30	38	57
Module (m), [mm]		4.5		3.5		1.75
Centre distance (a), [mm]		91.5		91.5		91.5
Pressure angle (α), [°]		20		20		20
Helix angle (β), [°]		-		15		15
Face width (b), [mm]		40		23		23
Profile shift (x_i), [/]	+0.1817	+0.1715	+0.1809	+0.0891	+1.6915	+2.0003
Addendum diameter (d_{ai}), [mm]	82.64	118.54	80.67	116.27	76.23	111.73
Transverse contact ratio (ε_α), [/]		1.44		1.46		0.93
Overlap contact ratio (ε_β), [/]		-		0.54		1.08
Average roughness (Ra), [μm]		≈ 0.7		≈ 0.35		≈ 0.35
Material	16MnCr5		16MnCr5		16MnCr5	

* - FZG type C gear with a face width of 40 mm;

During the implementation of these load distribution models, these gears were used with the intent of validating the results and making sure that the application of the models is performed correctly, as it will be presented further ahead.

3.1 Description of the contact lines and the coordinate system

While gear teeth are meshing with each other, they create a plane which is normal to the surface of each teeth and is tangent to both base cylinders. This plane is designated by plane of action [10].

Regarding spur gear contact lines, they are parallel to the lines that are tangent to the base cylinder and the plane of action, take on figure 3.1a as an example. Concerning the contact lines of helical gears, they are tilted with an angle equal to the base helix angle (β_b) in relation to line formed by the tangent of the plane of action with the base cylinder, figures 3.1b and 3.1c for instance [10].

Assuming that spur gears are a specific case of an helical gear in which the helix angle is equal to zero, it is possible to make a distinction between three different relations of the contact ratio (ε_α) and overlap ratio (ε_β), just as it follows [10]:

1. $\varepsilon_\beta = 0$;
2. $\varepsilon_\beta \leq \varepsilon_\alpha$;
3. $\varepsilon_\beta > \varepsilon_\alpha$.

Each of the previous cases determines a particular “shape” of the lines of contact, as it can be seen in figure 3.1. For $\varepsilon_\beta = 0$, as it a spur gear, the lines are described like in figure 3.1a. The other two relations are both regarding helical gears. On one hand, if $\varepsilon_\beta \leq \varepsilon_\alpha$, there are contact lines across all the gear face width, attend on figure 3.1b. On the other hand, if $\varepsilon_\beta \geq \varepsilon_\alpha$, the contact lines never pass over the entire gear face width, as shown in figure 3.1c [10].

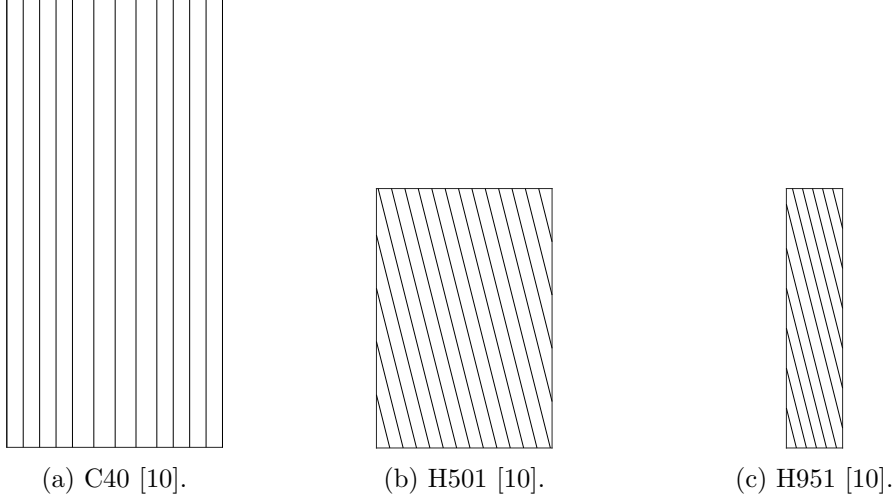


Figure 3.1: Contact lines for different gear geometries: (a) $\varepsilon_\beta = 0$; (b) $\varepsilon_\beta \leq \varepsilon_\alpha$; (c) $\varepsilon_\beta > \varepsilon_\alpha$ [10].

Throughout the gear meshing action, more than one teeth pair might be in contact. When this happens, the lines of the different teeth pairs in contact are placed apart from each other by a distance equal to the transverse base pitch (p_{bt}) [10].

Spur and helical gears have very distinctive contact line, this fact is originated by the way the meshing processes. When the teeth of a spur gear enter the path of contact, since the contact is made instantaneously, the length of the line of contact is equal to the gear face width (b). However, for helical gears, the contact between teeth is made progressively, meaning that, when the teeth enter the path of contact, the length of the lines in contact increases gradually until it reaches a maximum (the maximum can be either a plateau or a peak) and then decreases again [10].

At last, the coordinate used along the work is ξ , which is a non-dimensional coordinate along the path of contact. ξ is defined by the ratio of the distance along the path of contact and the transverse base pitch. Looking upon how this coordinate is defined, at the start of the path of contact $\xi = 0$ and at the end $\xi = \varepsilon_\beta + \varepsilon_\alpha$. Figure 3.2 helps visualizing the coordinate ξ [10].

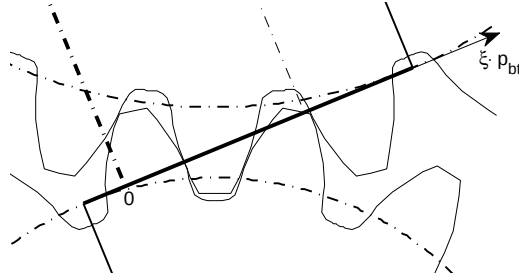


Figure 3.2: Definition of the ξ coordinates [10].

In some situations, additional coordinates may be needed, for example, the mesh period (T_{mesh}) and consequently the meshing time, which is useful describing in order to ξ . Respecting the mesh period, which is the time necessary to cover the transverse base pitch (p_{bt}), it is given by equation (3.1) [10].

$$T_{mesh} = \frac{2\pi}{\omega_i \cdot z_i} \quad (3.1)$$

where ω is the rotational speed and z is the number of teeth of gear i .

From all the coordinate definitions given so far, one can now relate the mesh time with the non-dimensional coordinate ξ through equation (3.2) [10].

$$t(\xi) = \xi \cdot T_{mesh} \quad (3.2)$$

3.2 Quasi-Static Rigid Model

It is important to enhance the assumptions of the quasi-static rigid model. It assumes that the load per unit of length along a line of contact over a tooth is constant for a given position in the path of contact. Furthermore, at a given position, the load per unit of length is the same for all meshing tooth pairs. One can conclude that the load distribution along the path of contact is inversely proportional to the sum of the lengths of the lines in contact [10].

In spur gears, when a teeth pair enters into contact, the length of the contact line is equal to the gear face width, which means, it behaves like a sudden increment in the length of the contact lines. The Heaviside function, a unit step function, can then be used to describe this phenomenon [10].

An approximation of the Heaviside function in order to ξ is given by equation (3.3), where a value of $k = 1000$ already presents good agreement with the theoretical Heaviside function [10]. Figure 3.3 is the representation of the Heaviside function approximation used in this work.

$$H(\xi) = \lim_{k \rightarrow +\infty} \left(\frac{1}{1 + e^{-2 \cdot k \cdot \xi}} \right) \quad (3.3)$$

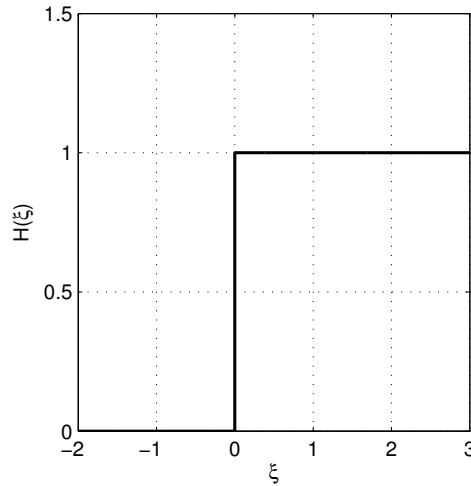


Figure 3.3: Approximation to the Heaviside function, equation (3.3) [10].

3.2.1 Length of the Contact Lines

When the tooth of a spur gear enters the active section of the path of contact, the contact line length per unit of face width is equal to one and outside the active section is zero. The shape of contact line length ratio is equal to a square with a length equal to ε_α and unitary height. This shape can be defined by subtracting two Heaviside functions, one translated by ε_α in relation to the other, as in equation (3.4) and as illustrated in figure 3.4a. This function also limits the domain of the length of the contact lines [10].

$$Tl^s(\xi) = H(\xi) - H(\xi - \varepsilon_\alpha) \quad (3.4)$$

It is relevant to point out that more than one pair of teeth may be engaged simultaneously and in that case, they are spaced by p_{bt} . In the current coordinate system (ξ), this spacing corresponds to an unit. The next step is to shift $Tl^s(\xi)$ in order to consider the teeth pairs that are forward and behind. By moving $Tl^s(\xi)$ in integer values i , the unbounded contact line length ratio, $Ul_i^s(\xi)$ (figure 3.4b), is obtained - equation (3.5). The definition of the integer values i is showed in equation (3.6) [10].

$$Ul_i^s(\xi) = \left[H(\xi - i) - H(\xi - \varepsilon_\alpha - i) \right] \quad (3.5)$$

$$i = -\text{floor}(\varepsilon_\alpha) : 1 : \text{floor}(\varepsilon_\alpha) \quad (\text{floor is rounding down}) \quad (3.6)$$

However, equations (3.4) and (3.5) only describe the domain and the unbounded length of the contact lines of spur gears, respectively. For helical gears, some changes have to be made to correctly represent these gears behavior [10].

One of them is the normalization, instead of being in order to the gear face width, the length of the contact lines is divided by $b/\cos(\beta_b)$. The other alterations are with respect to the aforementioned equations which have to be modified to characterize the length of the contact lines of helical gears [10].

The trim function has to account for the overlap ratio (ε_β), this means the domain is now $0 \leq \xi \leq \varepsilon_\alpha + \varepsilon_\beta$, which is the only difference from the trim function for spur gears, Tl^s - equation (3.4). By changing the trim function accordingly, $Tl^h(\xi)$ is obtained, equation (3.7) [10].

$$Tl^h(\xi) = H(\xi) - H(\xi - (\varepsilon_\alpha + \varepsilon_\beta)) \quad (3.7)$$

In relation to the unbounded length of the contact lines for helical gears, $Ul_i^h(\xi)$ (equation (3.8)), one needs to combine a linear function, this way the gradual increase, decrease and maximum can be represented, example in figure 3.5b [10].

$$\begin{aligned} Ul_i^h(\xi) = \frac{1}{\varepsilon_\beta} \cdot \left[H(\xi - i) \cdot (\xi - i) - H(\xi - \varepsilon_\beta - i) \cdot (\xi - \varepsilon_\beta - i) \right. \\ \left. - H(\xi - \varepsilon_\alpha - i) \cdot (\xi - \varepsilon_\alpha - i) \right. \\ \left. + H(\xi - (\varepsilon_\alpha + \varepsilon_\beta) - i) \cdot (\xi - (\varepsilon_\alpha + \varepsilon_\beta) - i) \right] \end{aligned} \quad (3.8)$$

The bounded contact line length ratio of meshing tooth pair i for both spur and helical gear pairs, $l_i^{s,h}(\xi)$ (equation (3.9)), is acquired by multiplying $Ul_i^{s,h}(\xi)$ by $Tl^{s,h}(\xi)$, this way $Ul_i^{s,h}(\xi)$ is limited to the respective domain [10]. In figures 3.4c and 3.5c it is possible to visualize an example for both type of gears.

$$l_i^{s,h}(\xi) = Ul_i^{s,h}(\xi) \cdot Tl^{s,h}(\xi) \quad (3.9)$$

Finally, the total length of the contact lines, $L^{s,h}(\xi)$, is given by the sum of the bounded contact line length ratio for all meshing tooth pairs and is exemplified in figures 3.4d and 3.5d for a spur and helical gear, respectively [10].

$$L^{s,h}(\xi) = \left[\sum_{i=-\text{floor}(\epsilon_\alpha+\epsilon_\beta)}^{\text{floor}(\epsilon_\alpha+\epsilon_\beta)} Ul_i^{s,h}(\xi) \right] \cdot Tl^{s,h}(\xi) \quad (3.10)$$

Figures 3.4 and 3.5 are an illustration of the different stages in order to acquire the total length of the contact lines for both spur and helical gears.

Concluding the formulation of the length of the lines in contact, the representation of the single contact line length ratio, $l_i^{s,h}(\xi)$, and the sum of the contact lines, $L^{s,h}$ is going to be conducted for all the gears presented in table 3.1 (see figure 3.6).

By analyzing figure 3.6 the effect of the behavior of the contact between spur and helical gears becomes clear. C40 presents step like variations in the total length of the lines in contact while in the helical gears, H501 and H951, the gradual contact between the teeth is evident.

3.2.2 Load Sharing

With the definition of $L^{s,h}(\xi)$, it is possible to establish the load per unit of length of the active section of the path of contact according to this model, equation (3.11). The normal load that a single pair of teeth withstands is given by equation (3.12), which is obtained by introducing $l_i^{s,h}(\xi)$ [10].

$$f_N(\xi) = \frac{M_w}{r_{bw}} \cdot \frac{1}{b \cdot L^{s,h}(\xi)} \quad (3.11)$$

$$F_{Ni}(\xi) = \frac{M_w}{r_{bw} \cdot \cos(\beta_b)} \cdot \frac{l_i^{s,h}(\xi)}{L^{s,h}(\xi)} \quad (3.12)$$

Analyzing equation (3.12), it can be concluded that the normal load acting on each pair of teeth is equal to a force $(M_w)/(r_{bw} \cdot \cos(\beta_b))$ multiplied by a sharing function which dictates how the load is distributed for each pair. This load sharing function is the quotient between the length of the contact lines for a single tooth by the length of the contact lines of all meshing tooth pairs in the path of contact, meaning that, the way the load is distributed is the same as the length of the contact lines [10].

The load sharing functions for the quasi-static rigid model (figure 3.7) show the distinctive way each gear distributes the loads along their mesh. As the load sharing functions are an extension of the formulation of the length of the contact lines, the effect of the meshing action is also evident. While the spur gear show an abrupt variation, the helical have progressive increases and decreases of the loads [10].

Even though the H501 and the H951 are both helical gears, which means they both have a gradual contact during the meshing, the H501 presents higher variations than the H951. Furthermore, the relation between the ϵ_α and ϵ_β is different for the aforementioned gears. For the H951 $\epsilon_\beta > \epsilon_\alpha$, this means that the maximum single tooth contact lines, $l(\xi)$, will never be above one since it is not possible to have a full line of contact [10].

The variation of the length of the contact lines will be investigated later on chapter 3.4.

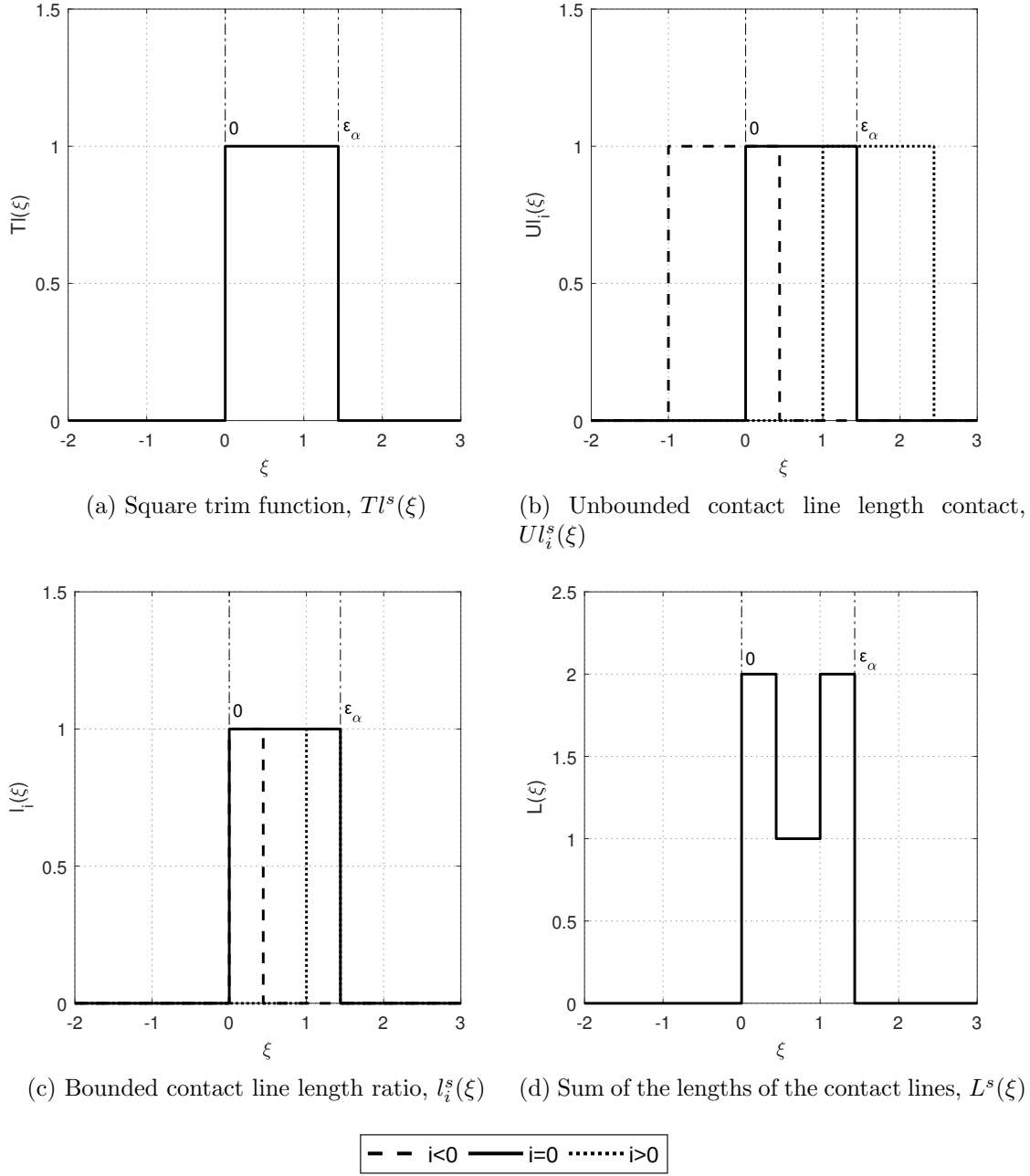
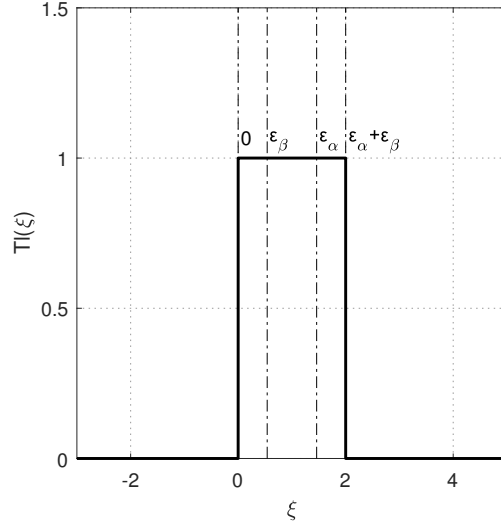
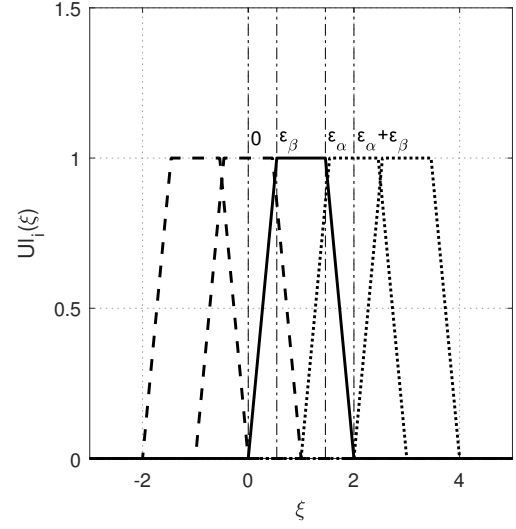


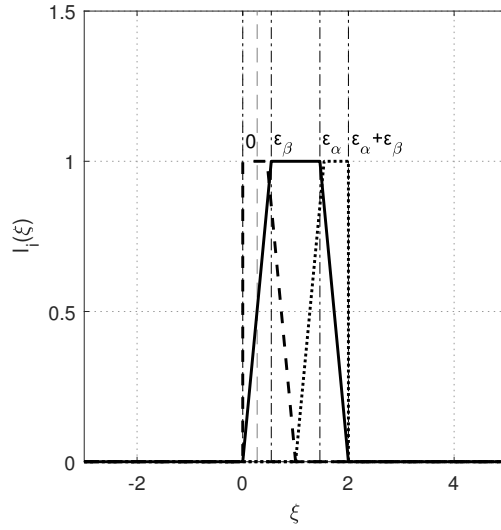
Figure 3.4: Example of the steps to reach the length of the contact lines for the C40, a spur gear.



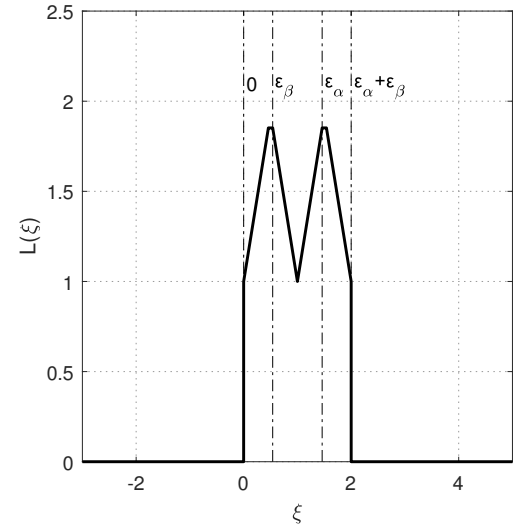
(a) Square trim function, $Tl^h(\xi)$



(b) Unbounded contact line length contact, $Uli^h(\xi)$



(c) Bounded contact line length ratio, $li^h(\xi)$



(d) Sum of the lengths of the contact lines, $L^h(\xi)$

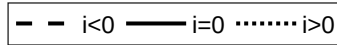


Figure 3.5: Example of the steps to reach the length of the contact lines for the H501, a helical gear.

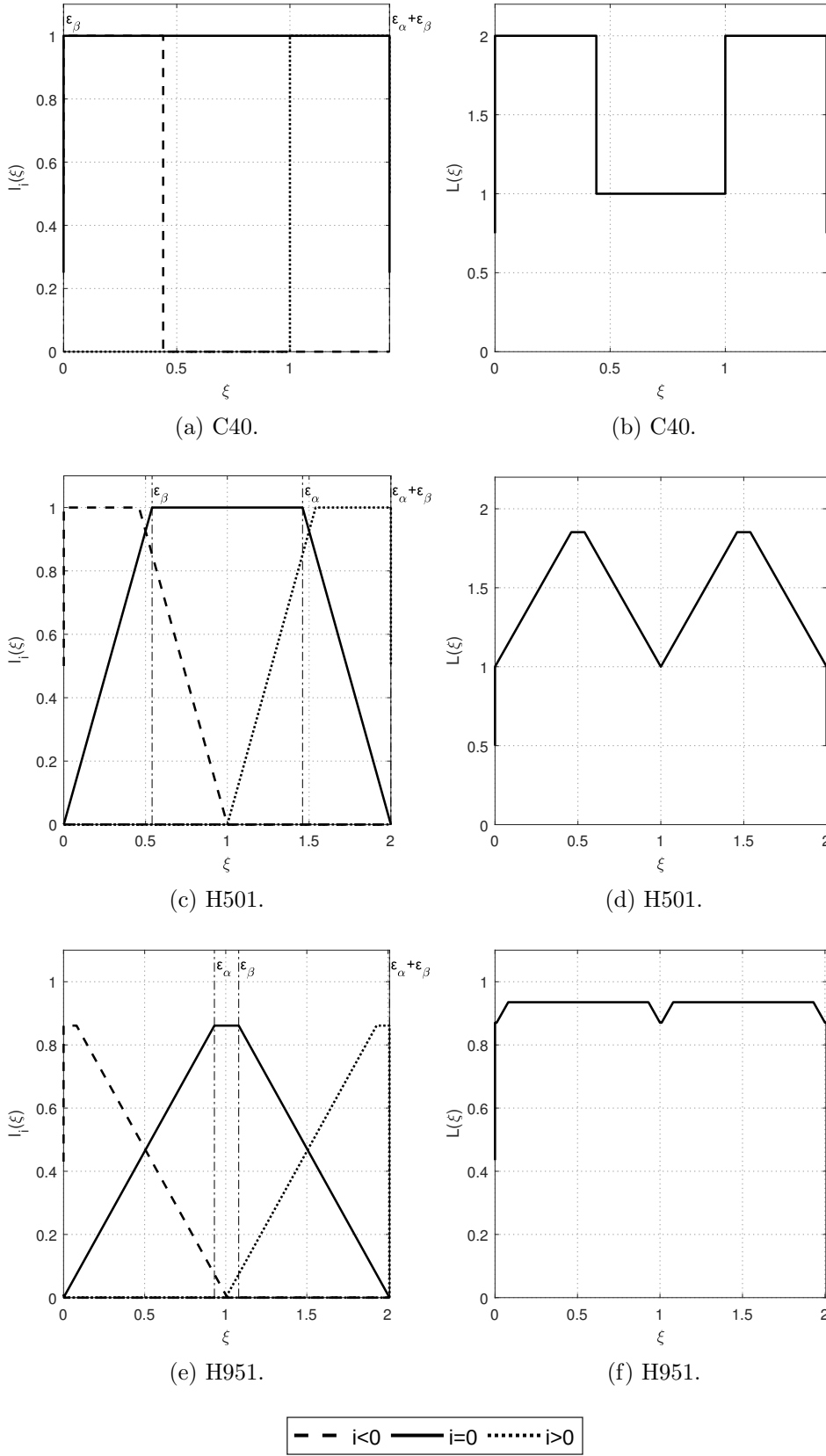
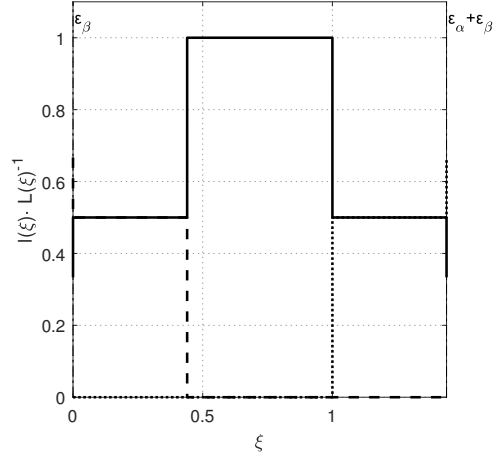
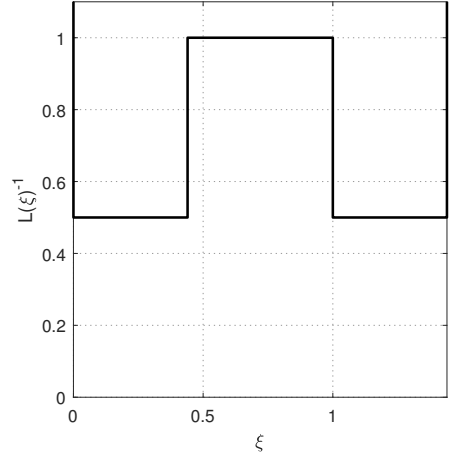


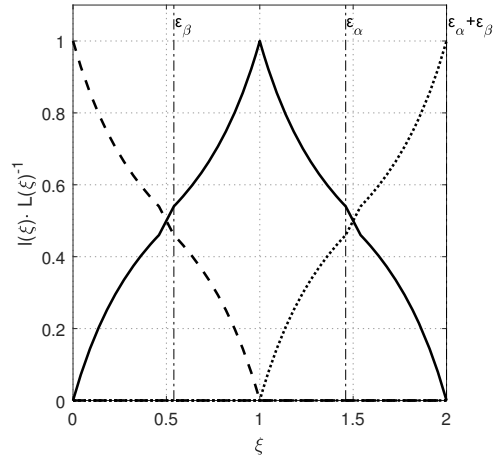
Figure 3.6: Comparison between the single contact length ratio and the total contact length for gears C40, H501 and H951.



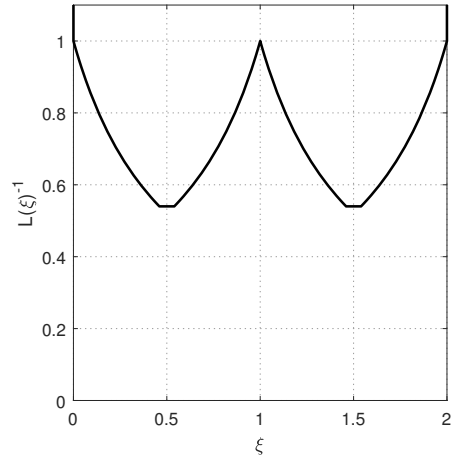
(a) C40.



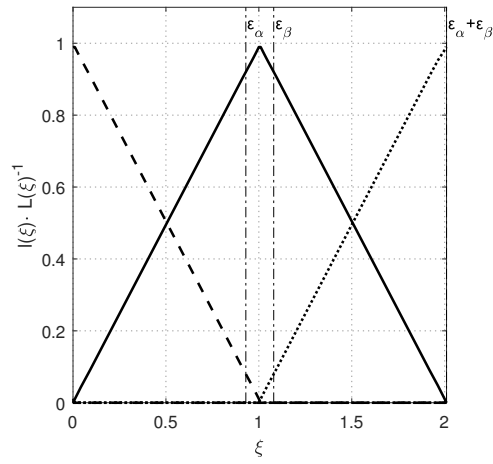
(b) C40.



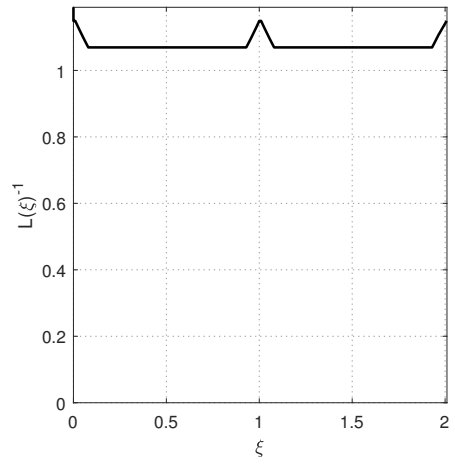
(c) H501.



(d) H501.



(e) H951.



(f) H951.

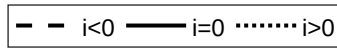


Figure 3.7: Comparison between the load sharing functions according to the quasi-static rigid model for gears C40, H501 and H951.

3.3 Quasi-Static Elastic Model

The main difference between this model and the previous one is that, the quasi-static elastic model considers elastic deformations. This means that, even though the load per unit of length for a certain position in the path of contact for a tooth pair is constant, this value changes between the different tooth pairs, alike the quasi-static rigid model [10].

It was also mentioned before that the gear is considered rigid up to the base cylinder and that Heaviside functions were going to be combined with the single tooth mesh stiffness to reach the load distribution. The single tooth mesh stiffness per unit of $b/\cos(\beta_b)$ for spur gears is approached by a parabolic/half sine-wave shape [10].

Considering the angular displacement extremely small ($d\theta$), the arc can be approximated to a linear displacement δ_b in the direction of the plane of action. Applying the definition of stiffness leads to equation (3.13) [10].

$$\delta_b = \frac{F_{bt}}{K^T(\xi)} \quad (3.13)$$

where δ_b is the linear displacement, F_{bt} is the total normal force in the transverse plane and $K^T(\xi)$ is the stiffness [10].

Each tooth pair i is submitted to the load $F_{Ni}^K(\xi)$, equation (3.14), equal to the multiplication of δ_b by the stiffness of the corresponding pair, $K_i(\xi)$ [10].

$$F_{Ni}^K(\xi) = \delta_b \cdot K_i(\xi) \quad (3.14)$$

The combination of equations (3.13) and (3.14) while having into account the fact that $F_{bt} = F_{bn}/\cos\beta_b$ results in the load distribution for a spur gear in the normal plane, equation (3.15) [10].

$$F_{Ni}^K(\xi) = \frac{K_i(\xi)}{K^T(\xi)} \cdot F_{bn} \quad (3.15)$$

It is worthwhile to take notice that the ratio $K_i(\xi)/K^T(\xi)$ represents the distribution of the load supported by the teeth pair i , therefore the output of equation (3.15) depends on what is multiplied by this load sharing function.

Defining the single tooth mesh stiffness per unit of gear width as a function of the maximum single mesh stiffness, yields equation (3.16).

$$K_i^u(\xi) = k_i^u(\xi) \cdot \frac{K_{max} \cdot \cos(\beta_b)}{b} \quad (3.16)$$

where $k_i^u(\xi)$ is the normalized single tooth mesh stiffness, K_{max} is the maximum single mesh stiffness and $b/\cos(\beta_b)$ is the gear face width [10].

K_{max} can be calculated by the formulation developed by ISO 6336-1 Standard [19]. In this formulation the maximum single tooth mesh (K_{max}^{ISO}) is calculated for spur and helical gears with deviations between -8% and 5% for $100 \leq F_{bt}/b \leq 1600$ N/mm [10].

Regarding the single tooth mesh stiffness, it is going to be approximated by a quadratic function. The quadratic function is normalized for the maximum single tooth mesh stiffness and therefore its maximum is one. As for the minimum single mesh stiffness, K_{min} , is equal to a percentage (α_k) of the maximum single mesh stiffness, according to equation (3.17) [10].

$$K_{min} = \alpha_k \cdot \frac{K_{max} \cdot \cos(\beta_b)}{b} \quad (3.17)$$

With all the previous considerations, the single tooth mesh stiffness normalized for the maximum single mesh stiffness can be written as in equation (3.18). Figure 3.8 represents equation (3.18) of the gear C40 for $\alpha_k = 4/5$ and $i = 0$ [10].

$$k_i^u(\xi) = \frac{4 \cdot (\alpha_k - 1)}{(\varepsilon_\alpha + \varepsilon_\beta)^2} \cdot (\xi - i)^2 - \frac{4 \cdot (\alpha_k - 1)}{\varepsilon_\alpha + \varepsilon_\beta} \cdot (\xi - i) + \alpha_k \quad (3.18)$$

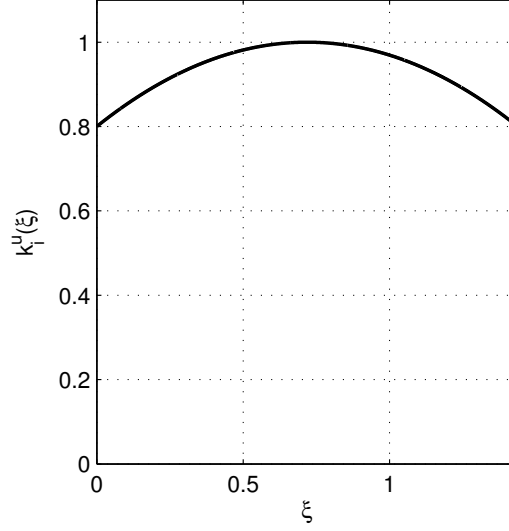


Figure 3.8: $k_i^u(\xi)$, equation (3.18), ($\alpha_k = 4/5$, $i = 0$) for the C40 gear (table 3.1) [10].

Assuming that an helical gear is a stock of spur disks with infinitesimal width and having in mind the following concepts, the definitions of $K_i^u(\xi)$ and $Ul_i^{s,h}$ and the fact that the load per contact line is constant, one can define the single pair stiffness of pair i according to (3.19) [10].

$$Kl_i^{s,h}(\xi) = Ul_i^{s,h}(\xi) \cdot \frac{b}{\cos(\beta_b)} \cdot K_i^u(\xi) \cdot Tl^{s,h}(\xi) \quad (3.19)$$

As several teeth may be engaged at the same time and the stiffness of each pair is parallel to the other, the total mesh stiffness is equal to the sum of the single pair stiffness of all meshing tooth pairs. Thus, equation (3.20) defines the mesh stiffness, $KL^{s,h}(\xi)$, for a given gear pair [10].

$$KL^{s,h}(\xi) = \left[\sum_{i=-\text{floor}(\varepsilon_\alpha + \varepsilon_\beta)}^{\text{floor}(\varepsilon_\alpha + \varepsilon_\beta)} Ul_i^{s,h}(\xi) \cdot \frac{b}{\cos(\beta_b)} \cdot K_i^u(\xi) \right] \cdot Tl^{s,h}(\xi) \quad (3.20)$$

At last, one can define the load distribution (F_{Ni}^{Ku} - equation (3.21)) and the load per unit length along the action line (f_{Ni}^{Ku} - equation (3.22)) for a pair i since all components for the load sharing function are determined.

$$F_{Ni}^{Ku}(\xi) = \frac{Kl_i^{s,h}(\xi)}{KL^{s,h}(\xi)} \cdot F_{bn} \quad (3.21)$$

$$f_{Ni}^{Ku}(\xi) = \frac{F_{Ni}^{Ku}(\xi) \cdot \cos(\beta_b)}{Ul_i^{s,h}(\xi) \cdot b} \cdot Tl^{s,h}(\xi) \quad (3.22)$$

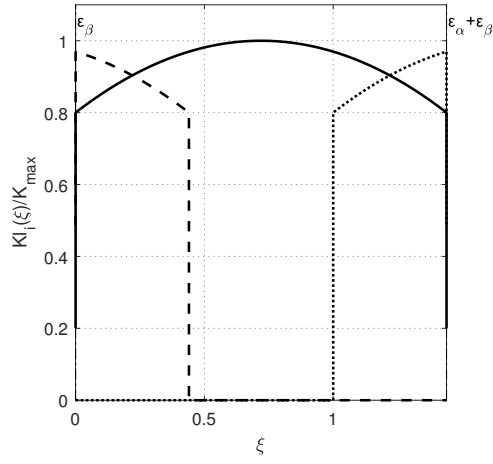
Figure 3.9 is the representation of the single and total mesh stiffness for the gear geometries in table 3.1 whereas figure 3.10 shows the single total load distribution and the load per unit of length for each tooth for the same gears. The following analysis is focused on the mesh stiffness since it is what mainly defines the load sharing functions and the dynamic behavior of the gear.

As expected, the C40 has instantaneous variation in the mesh stiffness that correspond to the transitions in the number of teeth in contact. During these transitions, the fluctuations in the mesh stiffness are approximately half the maximum stiffness value for this gear [10].

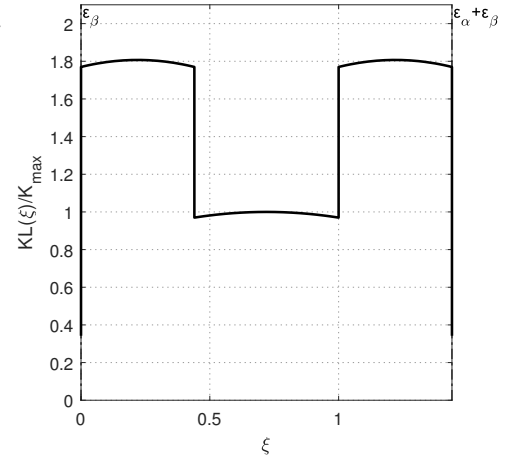
An helical gear does not necessarily have less stiffness variations, for instance, H501 has an amplitude of the same magnitude of the C40, but the way the variations behaves is different since for the helical gear the variation is gradual [10].

The H951 is a gear with an almost constant mesh stiffness as only slight fluctuations are presented during the meshing [10].

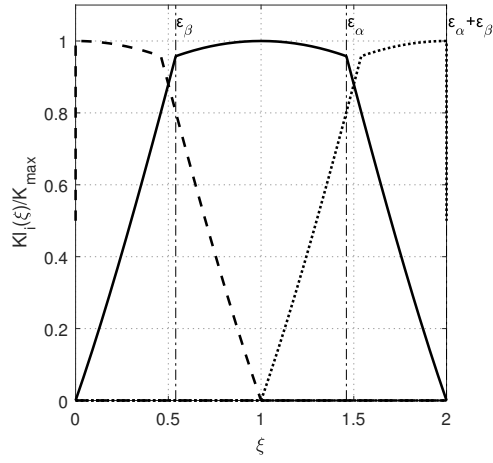
All figures reproduced using both models, the quasi-static rigid model and the quasi-static elastic models, are in agreement with the work of Marques [10] and therefore validated.



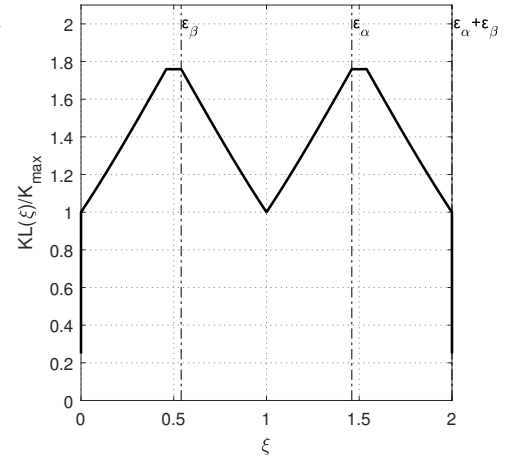
(a) C40.



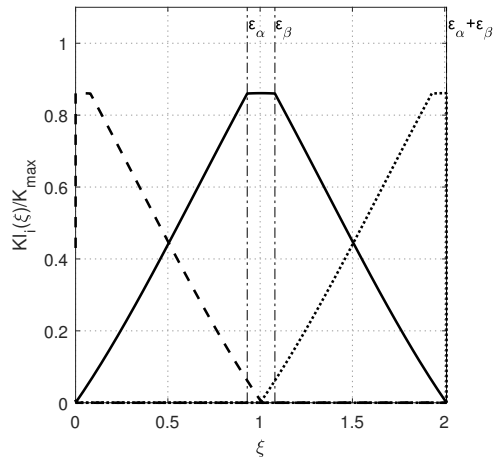
(b) C40.



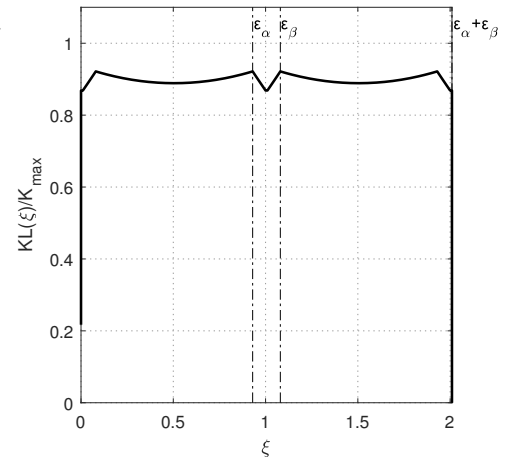
(c) H501.



(d) H501.



(e) H951.



(f) H951.

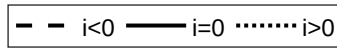


Figure 3.9: Comparison between the single and total mesh stiffness for gears C40, H501 and H951.

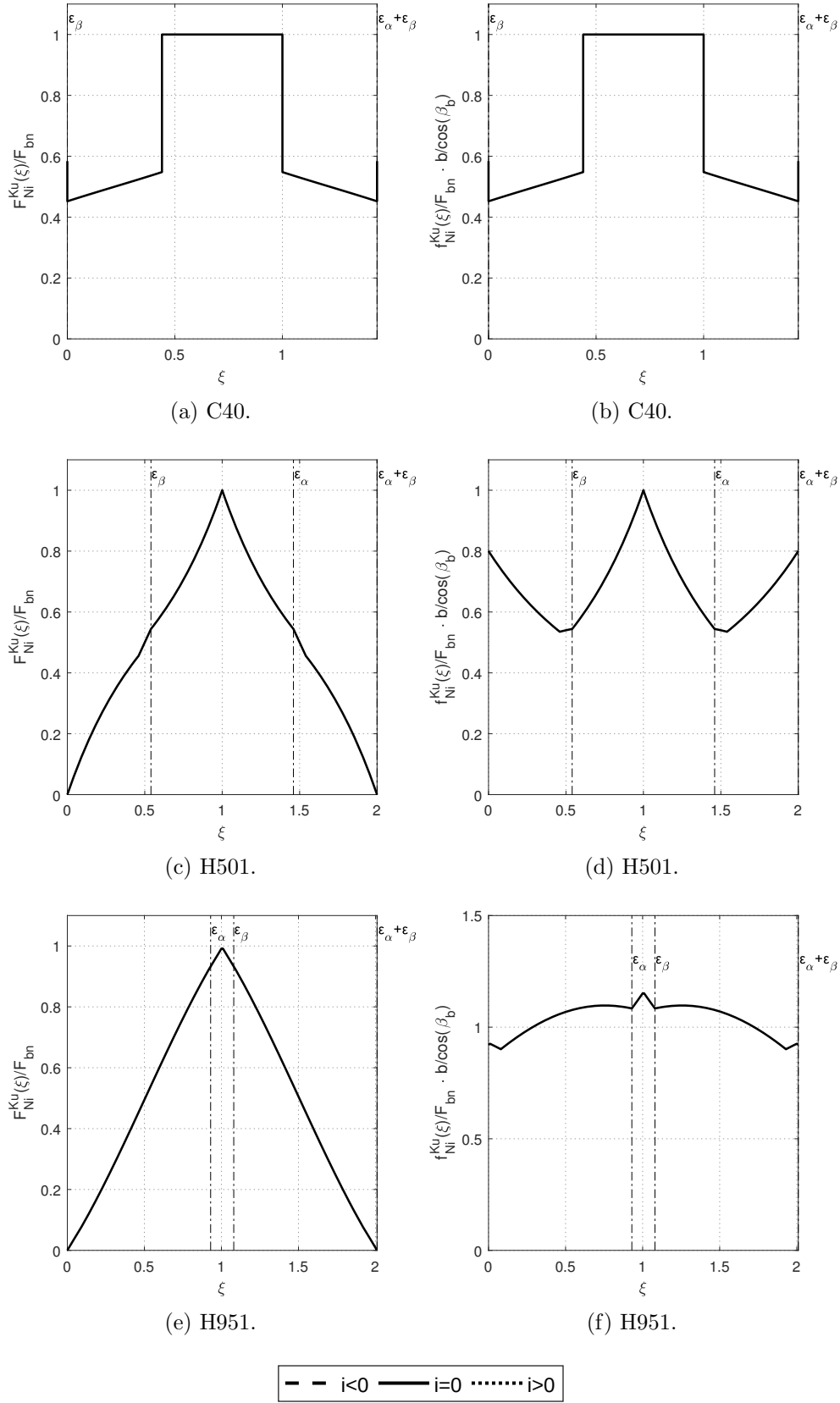


Figure 3.10: Comparison between the load sharing functions according to the quasi-static elastic model for gears C40, H501 and H951.

3.4 Analysis of the Contact Length Variation

In order to create the CMSGears, a further analysis of the behavior of the mesh stiffness is required. As the goal is a constant mesh stiffness or a mesh stiffness with minor variations, the total mesh stiffness should be studied. The total mesh stiffness, given by equation (3.20), is the sum of the single mesh stiffness of the different teeth in the path of contact, as mentioned before.

Looking further into the definition of the total mesh stiffness, equation (3.20), it is possible to identify two terms. The first one is the single mesh stiffness per unit of gear width, $K_i^u(\xi)$, which was formulated as a quadratic function and because of that can not be changed. Although it is important to mention that its maximum variation is 20%, meaning that its effect to the variation of the mesh stiffness is not that significant. This leaves the second term, which is the length of the lines in contact, given by $Ul_i^{s,h}(\xi) \cdot (b/\cos(\beta_b)) \cdot Tl^{s,h}(\xi)$. The length of the lines in contact is constituted by the unbounded contact length ratio ($Ul_i^{s,h}(\xi)$), the gear face width ($(b/\cos(\beta_b))$) and the trim function ($Tl^{s,h}(\xi)$). From all the previous elements, the only one that affects the variation of the stiffness is $Ul_i^{s,h}(\xi)$, thus it should be examined carefully.

The assessment of the mesh stiffness is made resorting to the sum of unbounded contact length ratio, which is, from the aforementioned statements, the main responsible for the appearance of variations in the mesh stiffness. This gave rise to the fact that for integer number of either the contact ratio or the overlap ratio, the length of the lines in contact is constant. Next, an example for a spur gear is presented since it is simpler and the idea behind the constant contact length is transmitted. However, as for helical gears the definition of $Ul_i^h(\xi)$ is different and there is the overlap ratio, the same procedure is applied and is included in the appendix A.

Take as an example the sum of $Ul_i^s(\xi)$ (equation (3.5)) for a contact ratio equal to 2 ($\varepsilon_\alpha = 2$), it follows that:

$$\begin{aligned}
\sum_{i=-2}^2 Ul_i^s(\xi) &= H(\xi + 2) - H(\xi - 2 + 2) + H(\xi + 1) - H(\xi - 2 + 1) + \\
&+ H(\xi) - H(\xi - 2) + H(\xi - 1) - H(\xi - 2 - 1) + H(\xi - 2) - H(\xi - 2 - 2) = \\
&= H(\xi + 2) - H(\xi) + H(\xi + 1) - H(\xi - 1) + H(\xi) - \\
&- H(\xi - 2) + H(\xi - 1) - H(\xi - 3) + H(\xi - 2) - H(\xi - 4) = \\
&= H(\xi + 2) + H(\xi + 1) - H(\xi - 3) - H(\xi - 4)
\end{aligned} \tag{3.23}$$

The mesh domain is $0 \leq \xi \leq 2$, so the last two terms of equation (3.23) are out of the boundary and therefore do not contribute for the sum. However, as the first two terms are within the mesh domain, these terms are the ones that are going to define how the sum of the unbounded length of the contact lines per unit of length behaves. Equation (3.23) can be rewritten into equation (3.24) by removing the unnecessary terms.

$$\sum_{i=-2}^2 Ul_i^s(\xi) = H(\xi + 2) + H(\xi + 1) \tag{3.24}$$

It is clear from equation (3.24) that during the mesh domain there are no variations because the Heaviside functions are step function and between the interval $0 \leq \xi \leq 2$ no other terms are added.

Concluding, $\sum_{i=-2}^2 Ul_i^s$ is constant for integer values of the contact ratio since none of the terms start during the mesh domain. Moreover, the definition of contact ratio is the “average number” of teeth in contact, which means that if the number of teeth in contact does not change, there will be no variation in the length of the lines in contact. During this section, a more detailed analysis of the variation of the contact length is going to be made.

Velex [20] studied the contact length variations for external spur and helical gears with involute profile and no manufacturing errors. The length of the lines in contact is represented by a Fourier series and from there he creates a contour plot of the root mean square (RMS) of the contact length ($L(\tau)$) divided by the average contact length (L_m), attend on figure 3.11.

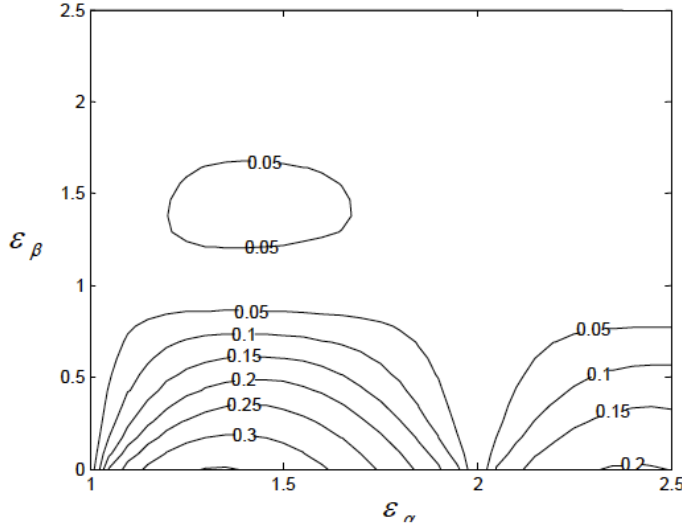


Figure 3.11: Contour plot of the RMS of $L(\tau)/L_m$ for a range of profile and transverse contact ratio [20].

The conclusions withdrawn were that when the values of the contact ratio or the overlap ratio were integer number, the contact length does not vary with time, which means it is constant. Besides that, for values of $\varepsilon_\alpha \leq 2$ and $\varepsilon_\beta \leq 1$ there are considerable variations in the contact length but for $\varepsilon_\beta \geq 1$ there are only slight variations [20].

The same procedure is applied to the formulated model of the length of the contact lines so that the results can be compared. The expression used to calculate the root mean square is presented in equation (3.25).

$$RMS_{f(t)} = \sqrt{\frac{1}{T_2 - T_1} \int_{T_1}^{T_2} [f(t)]^2 dt} \quad (3.25)$$

The current aim is to study the variation of the contact length, hence the function $f(t)$ is replace by $L^{s,h}(\xi)/\varepsilon_\alpha$ which is the total length of the contact lines, $L^{s,h}(\xi) \cdot (b/\cos(\beta_b))$, divided by the average contact length, $\varepsilon_\alpha \cdot (b/\cos(\beta_b))$. As for the limits of integration they are between zero and $\varepsilon_\alpha + \varepsilon_\beta$. That being said, equation (3.25) becomes equation (3.26).

$$RMS_{L^{s,h}(\xi)/\varepsilon_\alpha} = \sqrt{\frac{1}{\varepsilon_\alpha + \varepsilon_\beta} \int_0^{\varepsilon_\alpha + \varepsilon_\beta} [L^{s,h}(\xi)/\varepsilon_\alpha]^2 d\xi} \quad (3.26)$$

The application of equation (3.26) results in figure 3.13, where it is possible to visualize a 3D plot and a contour plot. However, before examining these figures it is important to compare the solution obtained with the one that Velez [20] presents, hence attend on figure 3.12.

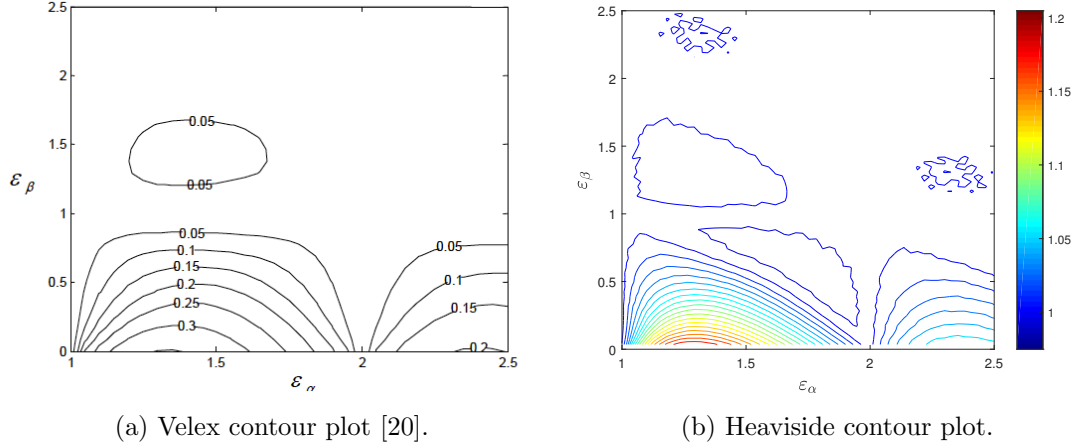


Figure 3.12: Comparison between the contour plot of the ratio length of the contact lines and the average length of the contact lines.

As it can be observed, both figures 3.12a and 3.12b are very similar, although there are some differences which can be justified for the following reasons. Firstly, the formulation used to model the length of the lines in contact is different. Whilst figure 3.12a uses a Fourier Series, figure 3.12b is based on the Heaviside function. Secondly and finally, the resolution of both contour plots is not the same, leading to some inconsistencies.

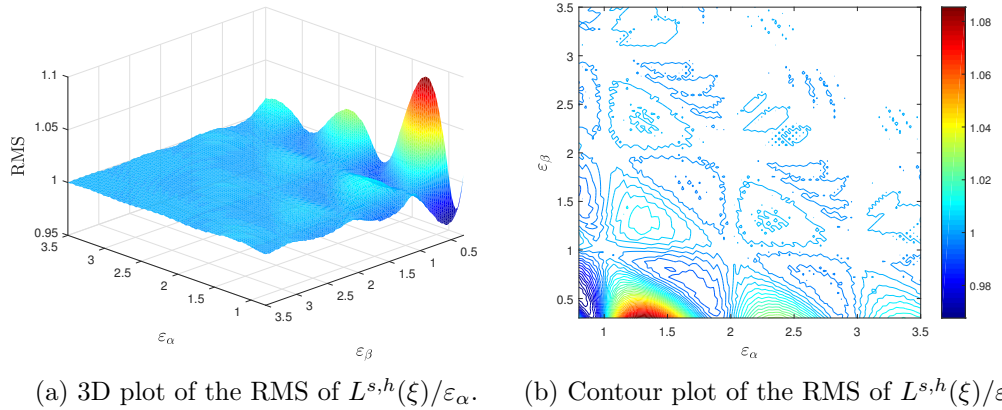


Figure 3.13: Representations of the RMS of $L^{s,h}(\xi)/\varepsilon_\alpha$ using the Heaviside model.

Moving to figure 3.13 and figure 3.14, it is clear it allows a better understanding of the variation of the length of the contact lines. Before moving to the analysis of these figures, it is relevant to point out that the bottom left corner ($\varepsilon_\alpha < 1 \wedge \varepsilon_\beta < 1$) of figure 3.13b and respective area in figure 3.13a represent gear geometries which are not valid and therefore must be ignored.

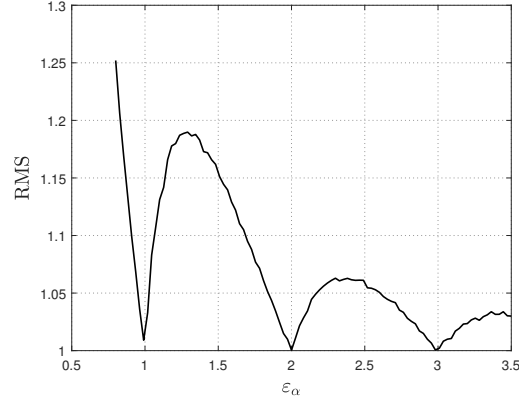


Figure 3.14: Representation of the RMS of $L^{s,h}(\xi)/\varepsilon_\alpha$ using the Heaviside model for a spur gear ($\varepsilon_\beta = 0$).

That being said, figures 3.13 and 3.14 reinforce the fact that for integer numbers of either the contact ratio or the overlap ratio the length of the contact lines is constant and figure 3.15 was generated for that purpose. Besides that, it is also possible to recognize that for higher values of the contact and overlap ratios the variations tend to diminish. It can also be concluded that the variations are concentrated for $\varepsilon_\alpha \leq 3$ and $\varepsilon_\beta \leq 2$.

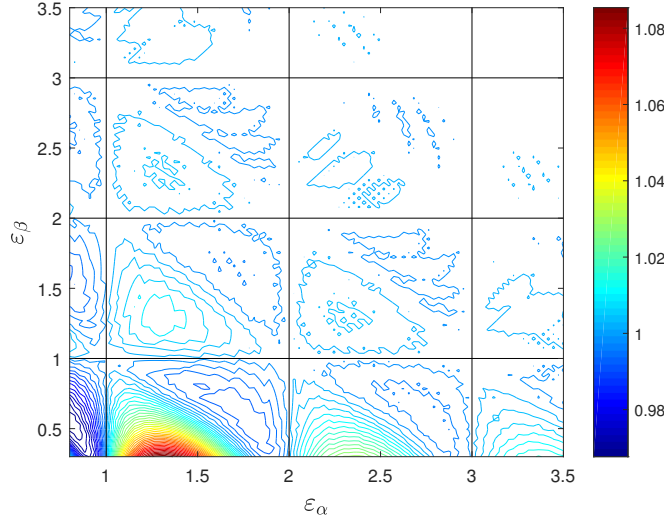


Figure 3.15: Contour plot of the RMS of $L^{s,h}(\xi)/\varepsilon_\alpha$ emphasizing the integer numbers.

After analyzing the variations of the length of the contact lines, the main conclusion reached is that for integer values of either ε_α or ε_β the length of the contact lines show no variation. Therefore, the constant mesh stiffness gear geometry is going to be obtained with the imposition of an integer value of the contact ratio (ε_α) or of the overlap ratio (ε_β).

Constant Mesh Stiffness Gears Development

This chapter is centered on the design of the constant mesh stiffness gears (MSGears). The entire process behind the development of these gears is explained, from the optimization algorithm up to the definition of the constant mesh stiffness (CMS) gear geometries.

For the development of the MSGears a MATLAB[®] program was created. The main aim of the program, as it was already implicit, was to generate a gear geometry with a constant mesh stiffness, which can be done by establishing an integer value for the contact ratio (ε_α) or for the overlap ratio (ε_β). In order to have a proper solution, the gear safety factors and efficiency were taken into account.

In relation to the implementation of the safety factors, the ISO 6336 Standards [21] were used with some simplifications. The gear meshing efficiency was included in the algorithm through the calculation of the gear loss factor, H_V^G . The approach used to calculate this efficiency factor was established by Marques [10]. Also, to make sure that all the gear design principles were fulfilled, the MAAG Gear Book [22] was used.

In the following sections a description of the program mentioning how it works and the difficulties experienced as well as a report on the application of the algorithm developed is done.

4.1 MSG Algorithm Development

At the center of the algorithm is a classical gradient iterative optimization process. This type of algorithms needs an initial point as an input to start the procedure. From that point, the search for a direction which reduces the objective function and does not violate the constraints begins. The path that the algorithm takes and the distance crossed depend on the constraints, the initial point given and the iterative process chosen, but generally small steps are taken each time. When another point is found, the procedure is repeated until no further minimization in the objective function is possible [23].

If the problem is defined in such way that the objective function presents several minimum points, the algorithm can mistake a relative minimum for an absolute minimum, meaning that a better outcome was still possible. In case the algorithm finds a relative minimum the procedure ends at that point because in no other direction is possible to reduce the objective function. So, in order for the algorithm to find a different path and reach a different solution, another starting point has to be specified and/or a new iterative process is used [23].

For the achievement of the CMS geometry, several starting points are used and therefore many different gear designs are obtained. From all the geometries generated, some of them are selected, depending on what is the main characteristic pretended for the gear.

4.1.1 Algorithm Performance

The overall operation of the algorithm core, composed by the Data Input and the Iterative Process, is pretty straightforward. The flowchart presented in figure 4.1 helps to understand the procedures of the algorithm and it can be used as a support along this section. The pretended gear geometry parameters and operating conditions are inserted together with the material and lubricant properties. Additionally, the starting point (starting gear geometry) is introduced. With all the needed data gathered, the iterative optimization process begins. Two main stages are included in the optimization algorithm. In the first one, all the necessary parameters and ratios for the calculation of the constraints and the objective function are computed. The second stage consists on finding a gear geometry that satisfies the constraints and, subsequently, determine the value of the objective function. This routine is repeated until a gear geometry that minimizes the objective function while respecting the constraints is obtained.

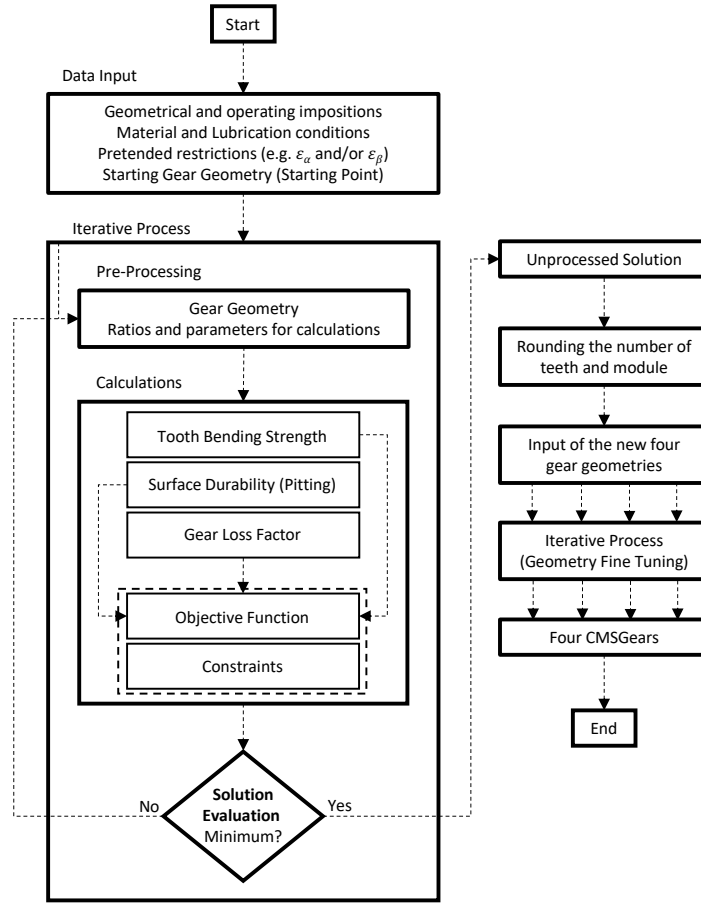


Figure 4.1: Flowchart of the algorithm developed for the CMSGears.

From the stated above, the algorithm can be divided into four parts, namely, data input, constraints, objective function and optimization algorithm.

Data Input

This part is responsible for introducing all the data required in the program. The software was implemented so that the relevant geometry and operating conditions of the gear are imposed. The user can insert, as a requirement for the gear pair, the pressure angle, the

working center distance, the rotating speed of the pinion, the velocity ratio (or gear ratio) and the torque applied to the pinion.

Further information relative to the working characteristics of the gear pair has to be given for the determination of the objective function. Some examples are the application factor, the lifetime and the safety factors which are used in the objective function.

Since the calculation of the load capacity of spur and helical gears (ISO 6336 [21]), in particular, the calculation of surface durability (pitting) (ISO 6336-2 [24]) and the calculation of the tooth bending strength (ISO 6336-3 [25]), is performed, both material and lubrication properties have to be provided.

Table 4.2 has the data needed to attain all the input parameters.

Constraints

The constraints define the constant mesh stiffness geometry and make sure the principles of gear design are satisfied. All the constraints applied in the algorithm and their corresponding function are displayed, that is:

Gear ratio: the gear ratio, equation (4.1), is an input for the algorithm and therefore it must be achieved in the final geometry.

$$u = \frac{z_2}{z_1} \quad (4.1)$$

Avoid interference on the mating gear pair: interference on a gear pair arises from the contact between non-involute parts of the gear teeth. The primary reason that causes interference is a faulty design and its consequence is severe shock during transmission [22]. The interference can happen at the root and at the tip of mating gears, therefore, as both kinds of interference have to be avoided, equation (4.2) along with equations (4.3) and (4.4) are used to avoid root interference and equation (4.5) for avoiding tip interference [22]. All the parameters presented are defined in more detail in MAAG Gear Book [22].

$$g_{inv} \leq g'_{inv} \quad (4.2)$$

$$g_{inv} = \frac{d_{b1}}{2} \cdot \tan \alpha_t - \frac{h_{inv}}{\sin \alpha_t} \quad (4.3)$$

$$\text{where } h_{inv} = m[h_{fP}^* - \rho_{fP}^*(1 - \sin \alpha) - x_1]$$

$$g'_{inv} = a \cdot \sin \alpha'_t - g_{a2} \quad (4.4)$$

$$\text{where } g_{a2} = \frac{d_{b2}}{2} \cdot \tan \alpha_{a2}$$

$$\text{and } \cos \alpha_{a2} = \frac{d_{b2}}{d_{a2}}$$

$$g_{a2} \leq a \cdot \sin \alpha'_t \quad (4.5)$$

Balance the maximum specific sliding: one of the criteria for the selection of the profile shift coefficients is the balance of the maximum specific sliding. This criteria reduces the wear at the critical points and contributes for a better performance of the gear pair as well as increases the gear durability [12; 26]. Equation (4.6) together with equations (4.7) and (4.8) are used to balance the maximum specific sliding [27]. Different criteria can be used for the selection of the profile shifts.

$$|gs_{1,max}| = |gs_{2,max}| \quad (4.6)$$

$$|gs_{1,max}| = \left| 1 - \frac{\sqrt{\left(\frac{z_2}{2} + 1 + x_2\right)^2 - \left(\frac{z_2 \cos(\alpha)}{2}\right)^2}}{(z_1 + z_2) \frac{\cos(\alpha)}{2} \tan(\alpha') - \sqrt{\left(\frac{z_2}{2} + 1 + x_2\right)^2 - \left(\frac{z_2 \cos(\alpha)}{2}\right)^2}} \cdot \frac{z_1}{z_2} \right| \quad (4.7)$$

$$|gs_{2,max}| = \left| \frac{\sqrt{\left(\frac{z_1}{2} + 1 + x_1\right)^2 - \left(\frac{z_1 \cos(\alpha)}{2}\right)^2}}{(z_1 + z_2) \frac{\cos(\alpha)}{2} \tan(\alpha') - \sqrt{\left(\frac{z_1}{2} + 1 + x_1\right)^2 - \left(\frac{z_1 \cos(\alpha)}{2}\right)^2}} \cdot \frac{z_2}{z_1} - 1 \right| \quad (4.8)$$

Limitation of the contact ratio ε_α : contact ratio boundaries, equation (4.9), were established for the algorithm not to go overboard with the values. Moreover, a minimum value is demanded for the continuity of action.

$$1.2 \leq \varepsilon_\alpha \leq 2 \quad (4.9)$$

Imposition of the overlap ratio ε_β : it was formerly mentioned that the constant mesh stiffness is obtained when either the contact ratio or the overlap ratio are an integer number. However, the overlap ratio is less influenced by the load deformation to which the gears are subjected to, thus, this parameter was forced to be an integer. Depending on what was pretended, the overlap ratio was forced to be 1 or 2, equation (4.10). The value of $\varepsilon_\beta = 1$ was chosen when a more efficient/light weight gear was pretended. On the other hand, when a gear with higher load carrying capacity was desired, $\varepsilon_\beta = 2$ was imposed. The motives that led to these values will be detailed further ahead.

$$\varepsilon_\beta = 1 \vee \varepsilon_\beta = 2 \quad (4.10)$$

Addendum modification coefficient: since the center distance is a requirement for the calculation of the gear geometry, equation (4.11) must be satisfied for a correct gear design.

$$(x_1 + x_2) = \frac{z_1 + z_2}{2} \cdot \frac{\text{inv}\alpha'_t - \text{inv}\alpha_t}{\tan \alpha} \quad (4.11)$$

Objective Function

The determination of the objective function, equation (4.12), involves the calculation of the gear tooth bending strength and surface durability, which are introduced in the equation through the respective safety factors, S_F for the tooth bending and S_H for the surface

durability. The term that accounts for the gear efficiency is the gear loss factor H_V^G . The constants C_i are the weights of each term in the objective function.

$$\text{Obj. Fun.} = C_1 \cdot |S_{Fmin1} - S_{F1}| + C_2 \cdot |S_{Fmin2} - S_{F2}| + C_3 \cdot |S_{Hmin} - S_H| + C_4 \cdot H_V^G \quad (4.12)$$

The safety factors were calculated according to the ISO 6336 Standard [21] and, in order to understand each parameter a brief study was performed. During the application of the ISO Standards some aspects were not clear so, some time was lost figuring out the exact way of implementation. Furthermore, simplifications in the ISO Standard were done, that is, the gears were considered to have no rim and no profile modifications and the ISO Standard was only applied for two material classes, through-hardened wrought steels (V - carbon steels and alloy steels) and case-hardened wrought steels (Eh). After the implementation of each part of the ISO Standard, the parameters obtained were compared with the KISSsoft® results.

Regarding the gear meshing loss power (P_{VZP}), the formulation used, which is defined below, was developed by Marques [10] and constitutes a general approach for the calculation of the gear power loss.

Firstly, the sliding power loss per unit of gear face width between the gear tooth in mesh, equation (4.13), has to be calculated. The formerly mentioned equation assumes a Coulomb friction model, this way, the friction force is equal to the product of the normal force, $F_N(t, y)$, by the coefficient of friction, $\mu(t, y)$ [10].

$$p_{VZP}(t, y) = F_N(t, y) \cdot \mu(t, y) \cdot v_g(t, y) \quad (4.13)$$

In the current work the normal force was determined resorting to the quasi-static elastic model. The implemented coefficient of friction is the average coefficient of friction formulation proposed by Schlenk (μ_{mz}^S), equation (4.14) [10].

$$\mu_{mz}^S = 0.048 \cdot \left(\frac{F_{bn}}{l \cdot R_X \cdot (U_1 + U_2)} \right)^{0.20} \cdot \eta_0^{-0.05} \cdot R_a^{0.25} \cdot X_L \quad (4.14)$$

where Ra is the arithmetic mean roughness, l is defined according to equation (4.15) and X_L is the lubricant correction factor ($X_L = 1$ for non-additised mineral oils in mixed film lubrication).

$$l = \varepsilon_\alpha \cdot \frac{b}{\cos(\beta_b)} \quad (4.15)$$

Secondly, the single tooth meshing power loss along the path of contact is obtained by integrating the sliding power loss, equation (4.13), over the tooth face width as described by equation (4.16) [10].

$$P_{VZP}^{single}(t) = \int_0^b [F_N(t, y) \cdot \mu(t, y) \cdot v_g(t, y)] dy \quad (4.16)$$

Thirdly, multiple pairs of teeth may be in contact, thus the total power loss is given by the overlap of the single tooth power losses, equation (4.17) [10].

$$P_{VZP}^{total}(t) = \sum_{i=-\text{floor}(\epsilon_\alpha + \epsilon_\beta)}^{\text{floor}(\epsilon_\alpha + \epsilon_\beta)} P_{VZP}^{single}(t - i \cdot T^{mesh}) \quad (4.17)$$

The next phase consists in the calculation of the average power loss. By integrating the total power loss over a mesh period (T_{mesh}), the total energy dissipated over a mesh period is calculated. Then, if the previous result is divided by the mesh period, the average power loss is achieved, equation (4.18) [10].

$$P_{VZP}^G = \frac{\int_0^{T_{mesh}} P_{VZP}^{total}(t) dt}{T_{mesh}} \quad (4.18)$$

Finally, the gear loss factor, equation (4.19), can be established considering the classical model for gear power loss. It is important to point out that the gear loss factor shown in equation (4.19) is formulated assuming a constant coefficient of friction along the plane of action [10]. The coefficient of friction currently implemented in the program is the Schlenk's average coefficient of friction, equation (4.14).

$$H_V^G = \frac{P_{VZP}^G}{P_{IN} \cdot \mu_{mz}} \quad (4.19)$$

Optimization Algorithm

The last part of the algorithm that is going to be discussed is the optimization process. As it was aforementioned, for the beginning of the optimization algorithm a starting point is needed, which, in this case, were predefined gear geometries. Besides that, the algorithm also requires that boundaries for all the parameters that define the starting point are inserted. Table 4.1 shows the parameters that are necessary for the definition of the initial point and consequently the output parameters that will define the new gears. The definition of the boundaries can be used as an advantage so that the search for the optimum point is restricted to an area that makes sense in terms of gear design and allows the employment in the FZG test rig available for future experimental validation. Before the algorithm starts running the iterative process has to be chosen. From the iterative processes available in the MATLAB[®] function *fmincon*, the “interior-point” and the “sqp-legacy” were the ones used.

Table 4.1: Parameters required for the definition of gears in the program

Symbol	Units	Description
z_1	-	number of teeth of the pinion
z_2	-	number of teeth of the wheel
β	rad	helix angle
b	mm	face width
m	mm	module
x_1	-	profile shift of the pinion
x_2	-	profile shift of the wheel
ρ_{fP}^*	-	root fillet radius of the basic rack for cylindrical gears per module

When the first part of the algorithm finishes running, parameters such as the number of teeth and gear module are obtained on a non-standardized form, e.g., the number of teeth and module can be any real positive number within the imposed restrictions. A gear can only have an integer number of teeth and the values of the module and root fillet radius of the basic rack are normalized. To solve this problem, the number of teeth for the pinion and the wheel were both rounded up and down and the module was set to the closest normalized values. By applying this procedure, four new starting gear designs can

be formed with the combination of the number of teeth (up or down) with the module (up or down). In addition, as four new starting points are given, the chances of finding the absolute minimum is higher.

The second part of the algorithm repeats the calculations for the determination of the CMSGears (Geometry Fine Tuning) with the given constraints and data input but with four starting points, leading to four constant mesh stiffness geometries. However, this procedure presents a couple of flaws. One of them is that by simply rounding the number of teeth and the module up and down, it is not guaranteed that the gear design is valid for an input. If the geometry inserted leads to an imaginary contact ratio, this means the gear is not valid and the program can not proceed with its calculations, in other words, the program can not recover from an input gear that has an imaginary contact ratio. In order to solve this setback, before the algorithm starts running the new gear geometries, the contact ratio is evaluated and in case it is an imaginary value, that starting geometry is eliminated. The other issue that arises is the fact that the value of the transmission ratio might not be exactly equal to the pretended. If the transmission ratio is within 1.5% of relative error, the gear is accepted, otherwise it is discarded. Moreover, whenever the combinations from the rounding procedure resulted in transmission ratios distanced from the pretended, new combinations were implemented for the same initial conditions.

4.2 CMSG Algorithm Results

Before beginning to extract constant mesh stiffness gears, the program was validated. The safety factors were approved resorting to KISSsoft® and the maximum relative error found was around 6%, this can also be confirmed comparing the developed CMSGears and the reports from KISSsoft® in appendix B. In relation to the gear loss factor, the values acquired for the H501 and H951 gears were compared to the ones obtained by Marques [10] and the maximum relative error calculated was also about 6%. The errors obtained are within the expected considering that the calculations methods and the assumptions made are not exactly the same.

For the development of the CMSGears, some inputs have to be given to perform the calculations. Table 4.2 shows all the information needed to reach the introduced parameters. The material properties are available in the ISO 6336-5 [28]. When an input did not have a specific requirement, the default values in KISSsoft® were used.

Two kinds of constant mesh stiffness gears were pretended: efficient (E) CMSGears and robust (R) or high safety factor (HSF) CMSGears. The efficient type of CMSGears was obtained by setting $\varepsilon_\beta = 1$, this was implemented because with this condition the helix angle of the gears will be smaller. Smaller helix angles reduce the axial load transmitted to the rolling bearings and thus their power loss can be reduced. By applying this condition to develop the efficient CMSGears, the gearbox efficiency can also be increased. Regarding the generation of the high safety factor gears, the overlap ratio was forced to $\varepsilon_\beta = 2$, this way the gears tend to have higher load carrying capacity. Even though the axial thrust on the bearings is increased for higher values of the helix angle, the only focus of robust gears is the capacity to stand higher loads. A good example of application of these gear pairs is in the slave box of the FZG rig.

It was previously stated that the algorithm needed a starting gear geometry to begin the iterations. Two different starting points were used, one is the H501 gear and the other is the H951 gear, both gear geometries are defined in table 3.1. Thereby, for each of the starting gear geometries an efficient and a high safety factor gear is sought after.

On the subject of the safety factors, it was required that all gears have at least the

Table 4.2: Parameters imposed for the development of the CMSGears

Symbol	Description	Units	Value
α	Pressure Angle	rad	$20 \cdot (\pi/180)$
a'	Working Center Distance	mm	91.5
u	Velocity Ratio	-	1.5
n_1	Rotating Speed of Pinion	rpm	1200
T_1	Pinion Torque	Nm	477.78
Material	16MnCr5 (Class Eh)	-	-
ν	Poisson's Ratio	-	0.3
E	Modulus of Elasticity	Nmm ⁻²	206000
ρ	Density	kgmm ⁻³	$7.83 \cdot 10^{-6}$
σ_{Hlim}	Allowable Stress Number	Nmm ⁻²	1500
σ_{Flim}	Bending Stress Number	Nmm ⁻²	430
ρ'	Slip-layer Thickness	mm	0.003
Ra	Average Roughness	μm	$\simeq 0.35$
Rz_{root}	Mean Peak-to-Valley Roughness Root	μm	$\simeq 20$
Rz_{flank}	Mean Peak-to-Valley Roughness Flank	μm	$\simeq 2.5$
K_A	Application Factor	-	1.25
Accuracy Grade	-	-	6
Lubricant	ISO-VG 220 (Mineral-oil base)	-	-
ν_{40}	Kinem. Viscosity oil at 40°C	mm ² s	220
ν_{100}	Kinem. Viscosity oil at 100°C	mm ² s	17.5
TS	Oil Temperature	°C	70
Required Service Life	-	horas (h)	20000

same safety coefficients of a FZG C14 gear with the same material and under the same operating conditions (see table 4.3).

One final consideration relative to the constants C_i of the objective function, equation (4.12), and the iterative process applied in the algorithm. Both of the aforementioned factors can be changed regardless the type of gear pretended and the initial point of the algorithm. The values attributed to the constants C_i were $0 \leq C_i \leq 1$. If an efficient CMSGear was pretended, the weight of the C_4 was bigger than the others so that the algorithm could focus on the gear loss factor. On the other hand, if a high safety factor gear was pretended, the weight was shifted to C_1 , C_2 and C_3 . The alteration of the iterative process is useful because it helps making sure that the absolute minimum for the objective function is found, providing new gear geometries. During the application of the algorithm these factors were constantly changed in order to obtain as many different gear geometries as possible.

Table 4.3: Calculated Safety Factors of the gear FZG C14 with the algorithm

Safety Factor	Value
S_{F1}	0.8786
S_{F2}	0.9105
S_H	0.8388

4.2.1 Results

After running the algorithm in search for the CMSGears, it was understood how each parameter influenced the final geometries. Additionally, from the results of the intermediate gear geometry (unprocessed solution) some conclusions could be made and some modifications were implemented.

The root fillet radius in the unprocessed gear geometry was always getting to extremely low values which were not near the minimum normalized and therefore not acceptable. For that reason, another constraint was added that forced the root fillet radius per module to approximate to 0.25. Also, whenever a module of around 1.5 mm was obtained, the minimum safety factors were not reached in the final solution.

Contemplating the contact ratio constraint, equation (4.9), this was extended, in some cases, to $1 \leq \varepsilon_\alpha \leq 2$. Take as an example the H951 gear (table 3.1) which contact ratio is lower than one, actually $\varepsilon_\alpha = 0.93$. Still, it is relevant to point out that the H951 gear is a very particular gear geometry.

The gear face width, b , was restricted to a maximum of $b = 40$ mm, this way it could be used in the FZG gear test rig.

After these adjustments in the algorithm, several sets of gears were calculated and saved so that later a selection of the best geometries could be made.

For the evaluation of each kind of gears, the geometric parameters along with the safety factors and the constraints are interpreted. This means that, when an HSF gear was pretended, the parameters considered were not only the safety factors but also the gear loss factor. Thus, a balance between the load capacity and the power loss of the gear is performed. However, for the efficient gears, the aim was the most efficient gear that fulfills the minimum established safety factors. During the analysis of the output parameters it was noticed that the constraints were not always satisfied.

After a complete analysis of all sets of gear geometries, the chosen CMSGears are shown in table 4.4.

Table 4.4: Output parameters for the CMSGears developed

Parameters	CMS55-E	CMS55-R	CMS92-LE	CMS50-E	CMS60-R
Starting Gear Geometry	H951	H951	H951	H501	H501
$z_1/-$	22	22	37	20	24
$z_2/-$	33	33	55	30	36
m/mm	3.00	3.00	1.75	3.25	2.75
$\beta/^\circ$	15.3177	28.1147	21.6909	18.2219	28.6452
b/mm	35.68	40	29.75	32.65	36.04
$x_1/-$	1.0924	-0.1998	1.3702	1.0110	-0.2944
$x_2/-$	1.2971	-0.4326	1.8475	1.1843	-0.5398
$\rho_{fP}^*/-$	0.25	0.25	0.25	0.25	0.25
$S_{F1}/-$	1.3326	2.2261	0.7708	1.3789	1.9524
$S_{F2}/-$	1.3617	2.2485	0.8819	1.3840	1.9887
$S_H/-$	0.96	1.26	0.8867	0.9326	1.2105
$H_V^G/-$	0.0926	0.1658	0.0587	0.1020	0.1579
Constraints	CMS55-E	CMS55-R	CMS92-LE	CMS50-E	CMS60-R
Lower limit of ε_α	✓	✓	✓	✓	✓
Upper limit of ε_α	✓	✓	✓	✓	✓
Root Interference	✓	✓	✓	✓	✓
Tip Interference	✓	✓	✓	✓	✓
Imposition of ε_β	✓	✓	✓	✓	✓
Balance Max. Specific Slid.	✓	✓	✓	✓	✓
Addendum Modification	✓	✓	✓	✓	✓

The gears' nomenclature is composed by CMS, which stands for Constant Mesh Stiffness, followed by the sum of the number of teeth of the gear pair and a letter to represent whether the CMSGear is an efficient (E) or a robust (R) type of gear. Aside from the proposed gears, there is the CMS92-LE (LE: limited efficient) which was selected because

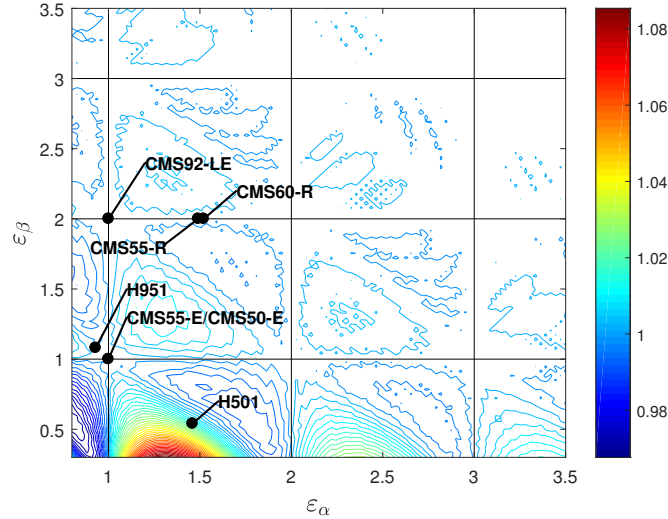
the calculated gear loss factor was lower than the H951. However, the CMS92-LE does not respect the required safety factors but they are higher than the safety factors of the H951 gear in the same conditions and therefore this gear was accepted, appendix B. This gear is also different in a way that the overlap ratio is equal to two ($\varepsilon_\beta = 2$), which even though was not planned was accepted since the helix angle is reasonably small.

From table 4.4 it can be concluded that all the gears satisfy the constraints established. It is also noteworthy here that all the safety factors of the CMSGears are higher than the safety factors of the C14 gear (table 4.3) except for the CMS92-LE gear as mentioned before. The CMS92-LE is the only gear that the velocity ratio is not exactly equal to the imposed value but as it is within less than 1% of relative error it was accepted. Finally, a distinction between the efficient and the high safety factor gears is clear. The helix angle for the efficient gears is smaller than for the robust gears. The safety factors are undoubtedly higher for the robust gears and the gear loss factor is smaller for the efficient gears, just as expected.

Figure 4.2 is a modification of figure 3.15, contour plot of the RMS of $L^{s,h}(\xi)/\varepsilon_\alpha$ emphasizing the integer numbers, in which the location of the CMSGears developed is highlighted along with the H501 and the H951, attend on table 4.5. The C40 location is marked in figure 4.3 since it is a spur gear, $\varepsilon_\beta = 0$. All the CMSGears, due to their definition, are located over a line of an integer number of the overlap ratio.

Table 4.5: Contact and overlap ratios for the CMSGears

Gear	C40	H501	H951	CMS55-E	CMS55-R	CMS92-LE	CMS50-E	CMS60-R
ε_α	1.440	1.460	0.930	1.000	1.491	1.001	1.000	1.521
ε_β	0.000	0.540	1.080	1.000	2.000	2.000	1.000	2.000

Figure 4.2: Contour plot of the RMS of $L^{s,h}(\xi)/\varepsilon_\alpha$ with points from gears in table 4.5.

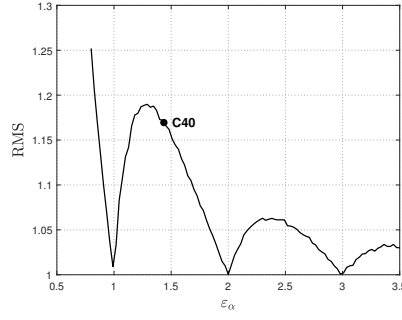


Figure 4.3: Representation of the RMS of $L^{s,h}(\xi)/\varepsilon_\alpha$ using the Heaviside model for a spur gear ($\varepsilon_\beta = 0$) pointing out the C40.

With the new geometries defined their mesh stiffness is going to be plotted. In figure 4.4 the mesh stiffness for all the CMSGears developed is plotted and can be compared with the mesh stiffness of the gears in table 3.1 (see figure 3.9).

As it can be seen, the stiffness is approximately constant for all the CMSGears (figure 4.4) but specially for the robust CMSGears where $\varepsilon_\beta = 2$ was imposed. When comparing the stiffness of these gears with the stiffness of the C40, H501 and H951, the difference in the variations are remarkable.

Regarding the efficiency of the CMSGears (table 4.6) all of them have a gear loss factor lower than the C40 ($H_V^G \simeq 0.1969$) and the H501 ($H_V^G \simeq 0.1853$). The CMS92-LE is the only gear with an H_V^G lower than the H951 ($H_V^G \simeq 0.0705$), the remaining CMSGears have a gear loss factor between the H501 and the H951.

Table 4.6: Gear loss factors for all gears

Gear	C40	H501	H951	CMS55-E	CMS55-R	CMS92-LE	CMS50-E	CMS60-R
H_V^G	0.1969	0.1853	0.0705	0.0926	0.1658	0.0587	0.1020	0.1579

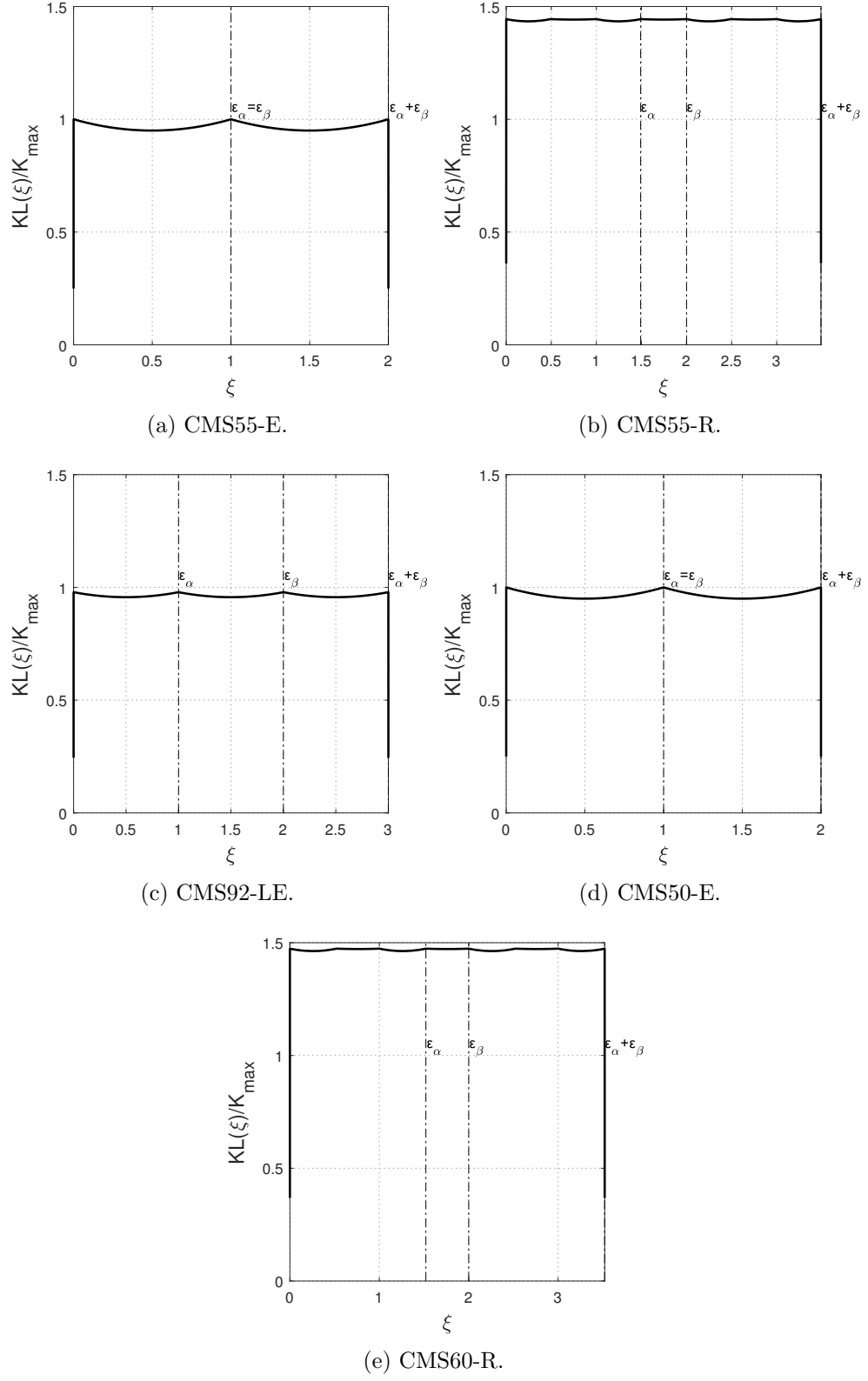


Figure 4.4: Total mesh stiffness for all the CMSGears developed.

Dynamic study of the Constant Mesh Stiffness Gears

In the previous chapter, the constant mesh stiffness gears were developed and the procedures to obtain such geometries were fully explained. Now, it is important to analyze their dynamic performance in order to conclude about the benefits and drawbacks of these gears.

For this purpose, a single degree of freedom dynamic model, modeling the torsional vibrations of a gear pair, was implemented. With this model the goal is to calculate the dynamic transmission error (DTE) and the dynamic mesh force (DMF) of the CMSGears and compare it with other kind of gear geometries and between the CMSGears. It is expected that the CMSGears present lower DTE and DMF oscillations. Besides that, as during the operation of a gear pair there are variations in the values of the contact and overlap ratios, a brief analysis on the effects of such variations in the dynamic behavior of the CMSGears was done.

In this chapter the dynamic model is fully described along with the deduction of the equation of motion for the torsional vibrations of a gear pair. The equation of motion was solved numerically with a single step, explicit Runge-Kutta formula (Dormand-Prince), ordinary differential equation solver available in MATLAB[®]. Finally, the results are examined and discussed.

5.1 Dynamic Model

Before proceeding to the development of the dynamic model, two different articles were examined to understand which assumptions should be made to achieve a model that can replicate the torsional vibration of a gear pair.

First, the model presented by Tamminana et al. [4], which is a non-linear, time varying dynamic model (figure 5.1) was examined. As figure 5.1 shows, this model comprises two rigid wheels of base radii, r_1 and r_2 , and polar mass moments of inertia, I_1 and I_2 (calculated using a disk approximation). It considers a periodically time-varying mesh stiffness, $k(t)$, caused by the variation in the number of teeth in contact. Additionally, there is a viscous damper with constant c which is calculated assuming a damping ratio of one percent (determined experimentally). The gear supports, shafts and bearings, are considered rigid. This model accounts for the gear backlash (magnitude $2b$) by including the function g , which is a non-linear restoring function, equation (5.1).

$$g[x(t)] = \begin{cases} x(t) - b, & x(t) > b, \\ 0, & |x(t)| \leq b, \\ x(t) + b, & x(t) < -b. \end{cases} \quad (5.1)$$

The manufacturing errors and the intentional modification on the tooth profile are inserted by forcing an external excitation, $e(t)$. However, this model disregards the friction forces at the tooth contact. Torque fluctuations are not taken into consideration owing to the experimental set-up used [4].

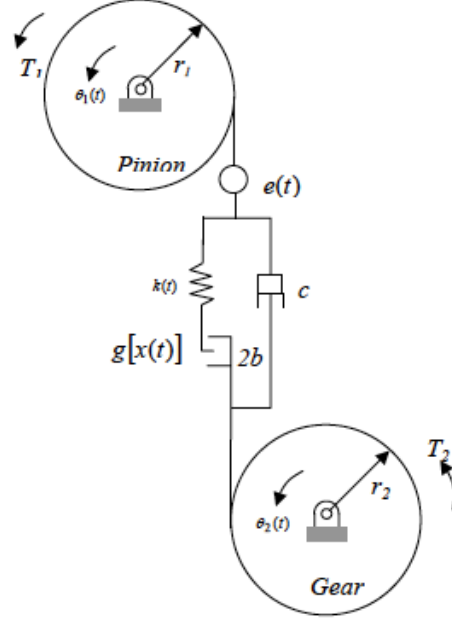


Figure 5.1: Discrete dynamic model of a spur gear pair presented by Tamminana et al. [4].

The current model can be reduced to a single degree of freedom, in this case, Tamminana et al. [4] used the difference between the DTE and the unloaded static transmission error (STE) as the degree of freedom, $x(t) = r_1\theta_1(t) + r_2\theta_2(t) - e(t)$. The mesh stiffness ($k(t)$) and the external excitation ($e(t)$) are obtained resorting to the deformable-body (finite element) model developed by the same authors and are included in the model in the form of a Fourier series.

To solve the non-linear equation of motion, Tamminana et al. [4] used a fourth order variable step Runge-Kutta (Dormand-Prince pair) numerical integration routine available in MATLAB®. The predicted DTEs using the model were in accordance with the experimental data, validating this way the model [4].

Secondly, the models used by Parker et al. [29], they are simpler or at least do not take into consideration as many effects as the previous model. The models presented by Parker et al. [29] are distinguished by the representation of the mesh stiffness, this will be clarified later.

Figure 5.2 shows the single degree of freedom model of a gear pair examined by Parker et al. [29]. The resemblances between this model and the model in figure 5.1 are obvious but small distinctions can also be detected.

I_1 and I_2 are the polar mass moments of inertia and r_1 and r_2 are the base radius of the gears. The mesh stiffness is included in the equation of motion through the elastic mesh force, $F(t)$, defined in equation (5.2) [29].

$$F(t) = \begin{cases} k(t) \cdot x(t), & x(t) \geq 0, \\ 0, & x(t) < 0. \end{cases} \quad (5.2)$$

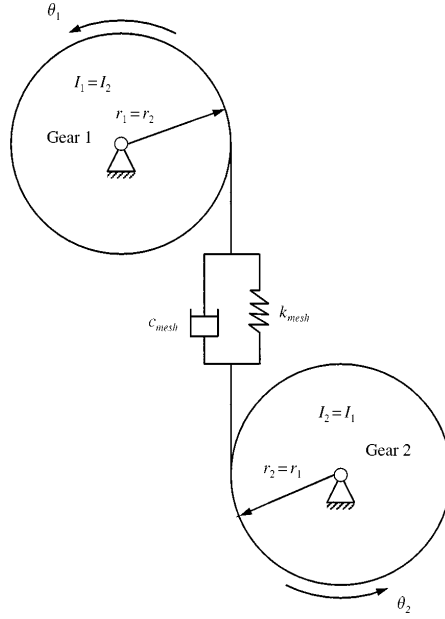


Figure 5.2: Single degree of freedom modeling of a two gear system [29].

From the analysis of the elastic mesh force, equation (5.2), it can be concluded that this model, alike the model of Tamminana et al. [4], does not account for the gear backlash. Although, it assumes loss of contact between the meshing teeth, there will never be a displacement that creates contact with the teeth in the back [29].

Concerning the gear mesh stiffness, two kinds of time varying stiffness were considered. One of them is not torque dependent and uses the rectangular wave approximation. The other, was calculated using the definition of stiffness, $k = (T/r)/STE$, where T is the torque, r is the base radius and STE is the static transmission error which is determined with a static finite element analysis. The damping is calculated considering an average mesh stiffness and a damping ratio of eight percent (determined with numerical impact tests and adjusted for the discrete model). The degree of freedom considered is equal to the dynamic transmission error, this is, $DTE = x(t) = r_2\theta_2(t) + r_1\theta_1(t)$. The condition where a pair of teeth are under unloaded contact corresponds to $x(t) = 0$ [29].

When comparing the results from the torque-independent stiffness model with the experimental tests, the predicted non-linear dynamic behavior correlated well with the rehearsal. However, the same is not true for the torque-dependent stiffness model. Even though this model allegedly has a more realistic mesh stiffness approximation, the results obtained were clearly different and were not in accordance with the experimental investigations. This lead to the conclusion that the approximation function used for the mesh stiffness is of remarkable importance [29].

5.1.1 Implemented Dynamic Model

The dynamic model implemented for the study of the dynamic performance is shown in figure 5.3. It is similar to both of the previously presented models and uses assumptions and effects from both of them. The gear supports are assumed to be rigid and the external excitation is included to account for the manufacturing errors since no profile modification was used in the CMSGears. However, in this work the gears were considered perfect and therefore the external excitation ($e(t)$) was assumed to be zero.

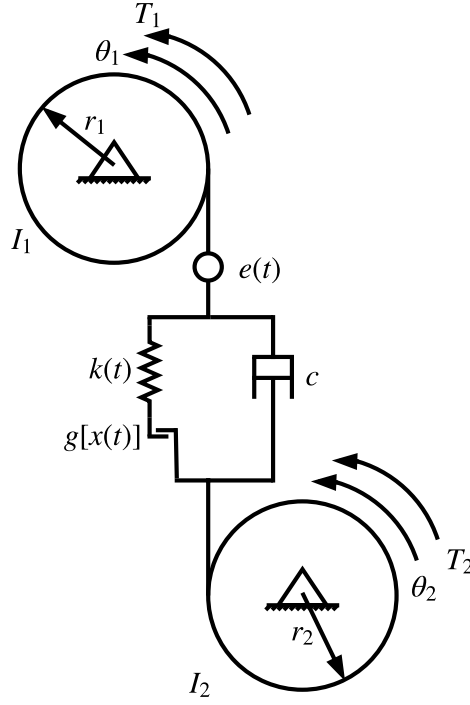


Figure 5.3: Dynamic model implemented.

Like the aforementioned models, the gears are approximated by two disks of base radius r_1 and r_2 and polar mass moments of inertia I_1 and I_2 . The periodically time varying mesh stiffness, $k(t)$, is calculated with the quasi-static elastic model and the viscous damper constant c is determined assuming a damping ratio of eight percent, which is in accordance with the model presented by Parker et al. [29]. The function $g[x(t)]$ is the contact loss function. T_1 and T_2 are the torque in the pinion and in the wheel, respectively.

The equations of motion can be obtained, in generalized coordinates, resorting to the Lagrange equations, equation (5.3).

$$\frac{d}{dt} \left(\frac{\partial T}{\partial \dot{q}_j} \right) - \frac{\partial T}{\partial q_j} + \frac{\partial V}{\partial q_j} = Q_j \quad j = 1, \dots, n. \quad (5.3)$$

where q_j are the generalized coordinates, Q_j are the generalized forces of each coordinate and T and V are, respectively, the kinetic and potencial energy of the system.

The generalized coordinates of the system are the angular displacements θ_1 and θ_2 . The kinetic energy (T) and the potencial energy (V) of the model are written as a function of the linear displacement in equations (5.4) and (5.5).

$$T = \frac{1}{4}m_1 \left(\dot{\theta}_1 r_1 \right)^2 + \frac{1}{4}m_2 \left(\dot{\theta}_2 r_2 \right)^2 \quad (5.4)$$

$$V = \frac{1}{2}k(t) \left(\theta_1 r_1 + \theta_2 r_2 - e \right)^2 \quad (5.5)$$

The damping of the model is going to be included in the equation of motion trough the Rayleigh's dissipation function, F , equation (5.6). This way the generalized forces are equal to the gradient of the Rayleigh's dissipation function and the external forces, equations (5.7) and (5.8).

$$F = \frac{1}{2}c \left(\dot{\theta}_1 r_1 + \dot{\theta}_2 r_2 - \dot{e} \right)^2 \quad (5.6)$$

$$Q_1 = T_1 - \frac{\partial F}{\partial \dot{\theta}_1} \quad (5.7)$$

$$Q_2 = T_2 - \frac{\partial F}{\partial \dot{\theta}_2} \quad (5.8)$$

By applying the Lagrange equations to the generalized coordinates, the equations of motion of the system are obtained, equation (5.9).

$$\begin{cases} \frac{1}{2}m_1\ddot{\theta}_1 r_1^2 + k(t)r_1 \cdot (\theta_1 r_1 + \theta_2 r_2 - e) = T_1 - c r_1 \cdot (\dot{\theta}_1 r_1 + \dot{\theta}_2 r_2 - \dot{e}) \\ \frac{1}{2}m_2\ddot{\theta}_2 r_2^2 + k(t)r_2 \cdot (\theta_1 r_1 + \theta_2 r_2 - e) = T_2 - c r_2 \cdot (\dot{\theta}_1 r_1 + \dot{\theta}_2 r_2 - \dot{e}) \end{cases} \quad (5.9)$$

As it is now, the system is represented by two degrees of freedom. To transform it into a single degree of freedom the equations in (5.9) have to be added. If the first equation is multiplied by $m_2/((m_1 + m_2)r_1)$ and the second by $m_1/((m_1 + m_2)r_2)$ then it is possible to add them. After working the expression, equation (5.10) is reached.

$$\begin{aligned} \frac{1}{2} \frac{m_1 m_2}{m_1 + m_2} (\ddot{\theta}_1 r_1 + \ddot{\theta}_2 r_2) + c (\dot{\theta}_1 r_1 + \dot{\theta}_2 r_2 - \dot{e}) + k(t) (\theta_1 r_1 + \theta_2 r_2 - e) = \\ = \frac{1}{2} \frac{m_1 m_2}{m_1 + m_2} \left(\frac{T_1 r_1}{I_1} + \frac{T_2 r_2}{I_2} \right) \end{aligned} \quad (5.10)$$

Defining the degree of freedom $x = \theta_1 r_1 + \theta_2 r_2 - e$, equation (5.10) becomes equation (5.11).

$$m_e \ddot{x} + c \dot{x} + k(t)x = m_e \left(\frac{T_1 r_1}{I_1} + \frac{T_2 r_2}{I_2} \right) - m_e \ddot{e}(t) \quad (5.11)$$

where $m_e = \frac{1}{2} \frac{m_1 m_2}{m_1 + m_2}$ is the equivalent mass.

The model being developed accounts for the contact loss and therefore the single degree equation of motion, equation (5.11), must be modified so that this effect is included through $g[x(t)]$.

$$m_e \ddot{x} + c \dot{x} + k(t)g[x(t)] = m_e \left(\frac{T_1 r_1}{I_1} + \frac{T_2 r_2}{I_2} \right) - m_e \ddot{e}(t) \quad (5.12)$$

$$g[x(t)] = \begin{cases} x(t), & x(t) \geq 0, \\ 0, & x(t) < 0. \end{cases} \quad (5.13)$$

Equation (5.12) together with equation (5.13) define the equation of motion for the pretended model. The contact loss is introduced in the system by equation (5.13), which as it was mentioned before, does not account for the gear backlash. This means that, independently of the displacement x , there will never be contact with the preceding teeth.

From the description of the equation of motion, it can be deduced that the only non-linearity is due to the contact loss. In order for the mesh stiffness to introduce non-linear behavior in the system, it had to depend on the position. If the mesh stiffness depended on the position, during the calculation of the equation of motion there would be one more

term originated from the partial derivative of the stiffness in order to space. This will be verified later by determining the differences in the dynamic response of the system with and without the contact loss function $g[x(t)]$.

5.1.2 Software Implementation

To solve the equation of motion and obtain the dynamic transmission error and the dynamic mesh force, the model was implemented in MATLAB[®]. An ordinary differential equation (ODE) solver available in MATLAB[®] that resorts to an explicit Runge-Kutta formula (Dormand-Prince) was used. The dynamic response had to be analyzed when the system was in steady state and on account of the non-linearities a speed up and speed down has to be performed, this is, the system is accelerated and decelerated. Figure 5.4 is the flowchart representing the dynamic model equation solver procedure.

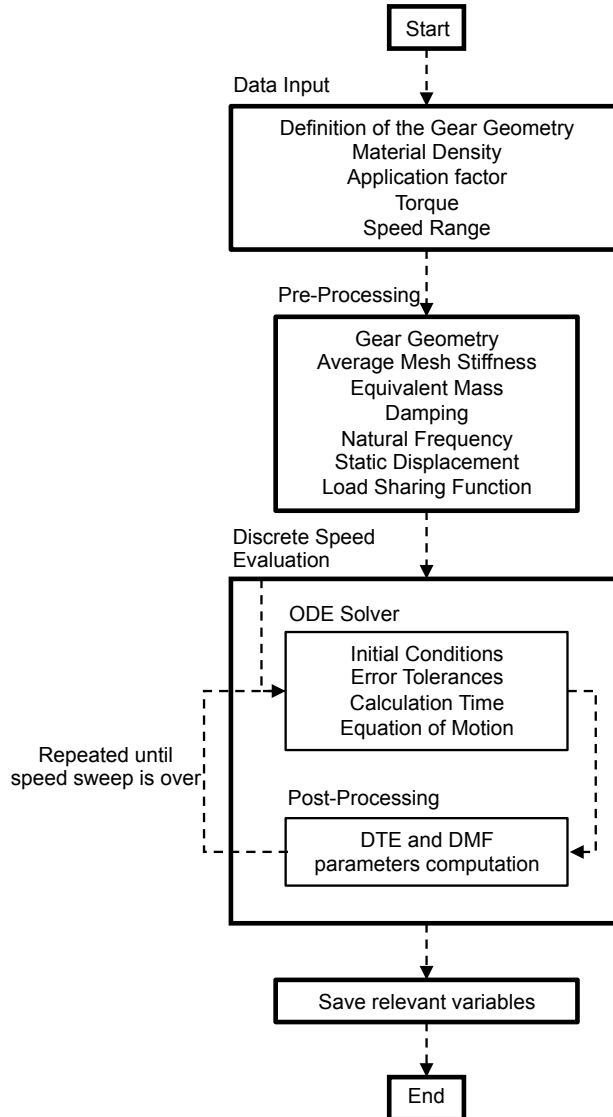


Figure 5.4: Flowchart of the dynamic model implemented.

The program is divided into four main parts, namely, data input, pre-processing, ODE solver and post-processing. In the **data input**, certain parameters have to be defined in

the program, that is, the gear geometry, the material density (ρ) and the application factor (K_A). Besides that, the torque to which the pinion is subjected to is also included in the input data. Finally, the speed range that the program evaluates the dynamic performance of the gears is defined.

Moving on to the **pre-processing**, the rest of the gear geometry is calculated for the determination of the average stiffness (k_{avg} - equation (5.14)), the equivalent mass (m_e), the damping (c - equation (5.15)) and the natural frequency (ω_n - equation (5.16)). The static displacement, δ_{static} equation (5.17), the total tooth normal force, F_{bn} equation (5.18), and the load sharing function from the quasi-static elastic model, F_N^K/F_{bn} equation (3.21), are all computed since they are needed for the post-processing.

$$k_{avg} = \frac{\int_0^1 k(\xi) d\xi}{1} \quad (5.14)$$

$$c = 2\zeta\sqrt{m_e}\sqrt{k_{avg}} \quad (5.15)$$

$$\omega_n = \sqrt{\frac{k_{avg}}{m_e}} \quad (5.16)$$

$$\delta_{static} = \frac{F_{bn}}{k(t)} \quad (5.17)$$

$$F_{bn} = \frac{T_w}{r_{bw}} \quad (5.18)$$

The next step is the **ODE solver** which involves the definition and solution of the equation of motion. First, the initial conditions and the error tolerances have to be defined. Secondly, the number of periods which the dynamic performance is calculated is stipulated. Note that, the system has to be analyzed when it is in steady state, therefore, the number of periods settled needs to allow the dynamic balance of the system. Thirdly, the equation of motion is defined. As all the required terms except for the mesh stiffness are already established in the pre-processing, for the complete determination of the equation of motion, the instantaneous mesh stiffness has to be calculated. To conclude this stage, the dynamic response (dynamic transmission error) of the system is determined by the solver.

With the dynamic transmission error (DTE) it is now possible to perform further calculations. In the **post-processing** the pretended variables to study the dynamic behavior of a gear pair are determined. The dynamic mesh force (DMF) is calculated by taking into account that the gear mesh damping force is so small compared to the dynamic gear mesh force that it can be neglected [4]. Thus, the DMF is approximated as the product of the DTE and the gear mesh stiffness, equation (5.19) [4].

$$DMF = DTE \cdot k(t) \quad (5.19)$$

For the following computations, only the last period of the DTE and the DMF is analyzed since that it is everything needed due to the periodicity of the dynamic gear action. Similar parameters are evaluated for the DTE and the DMF. Regarding the DTE, the maximum value over a period is determined, the mean DTE is obtained through the mean value theorem, equation (5.20), and the maximum normalized DTE (DTE^* , equation (5.21)), is also computed.

$$\overline{DTE} = \frac{\int_0^{T_{mesh}} DTE dt}{T_{mesh}} \quad (5.20)$$

$$DTE^* = DTE / \delta_{static} \quad (5.21)$$

Moreover, the root mean square (RMS), defined in equation (3.25), is calculated not only for the DTE but also for its oscillating component (DTE_{osc}), equation (5.22).

$$DTE_{osc} = DTE - \delta_{static} \quad (5.22)$$

As it was previously stated, the same factors are calculated for the dynamic mesh force. The main difference is in the normalization. Whereas in the DTE the normalization is in order to the static displacement, in the DMF the normalization is performed regarding the total tooth normal force, F_{bn} . Equation (5.23) is the normalized dynamic mesh force where a maximum value (dynamic overload) is also assessed.

$$DMF^* = DMF / F_{bn} \quad (5.23)$$

In addition, the calculation of the root mean square for the DMF and the respective oscillating component (DMF_{osc}), equation (5.24), are performed as well.

$$DMF_{osc} = DMF - F_{bn} \quad (5.24)$$

However, an additional parameter is computed for the DMF, which is the dynamic load distribution on the teeth. This is obtained by multiplying the load sharing function of the quasi-static elastic model by the dynamic mesh force, as in equation (5.25).

$$F_N^D(t) = \frac{Kl(t)}{KL(t)} \cdot DMF \quad (5.25)$$

To conclude the software implementation it is relevant to detail how the speed sweep was encompassed in the program. The solver calculates the dynamic response at a specific rotating speed considering the initial conditions established. For the first rotating speed evaluated, the initial conditions (dynamic transmission error and associated speed) are assumed to be zero. For the next rotating speed, the initial conditions of displacement and speed are the final conditions of the previous response. This is repeated until all the rotating speeds are evaluated.

5.1.3 Validation

To test if the model was implemented correctly, the figures representing the RMS of the oscillating DTE component and the mean DTE developed by Parker et al. [29] were reproduced (figures 5.5 and 5.6). The gear pair geometric parameters used in the model are showed in table 5.1.

The curves were obtained by setting $m_e = 1.4$ kg so that $f_n = (1/2\pi)\sqrt{k_{avg}/m_e} = 2900$ Hz, which is in agreement with the finite element model that Parker et al. [29] presented. Concerning the mesh stiffness in the figures 5.5a and 5.6a, a rectangular wave form approach was considered. The mesh stiffness for the model developed in this work was determined according to the quasi-static elastic model [10]. Even though the average mesh stiffness from both models is similar, the shape of the mesh stiffness is slightly different,

Table 5.1: Design parameters of the spur gear pair used in [29]

Number of teeth, $z_1/-$	50
Number of teeth, $z_2/-$	50
Pressure angle, $\alpha/^\circ$	20
Helix Angle, $\beta/^\circ$	0
Gear face width, b/mm	20
Module, m/mm	3
Profile shift, $x_1/-$	0
Profile shift, $x_2/-$	0

attend on figure 5.7. Relatively to the damping c , it is calculated in the same way, thus minor differences might be noticed owing to the differences in the average stiffness values.

In this section the main focus is the validation of the implemented model, so the following analysis is centered in the comparison of the figures obtained and not on the description of the dynamic behavior of gears. The results are compared in figure 5.5, for the RMS of the oscillating DTE component, and in figure 5.6, for the mean DTE.

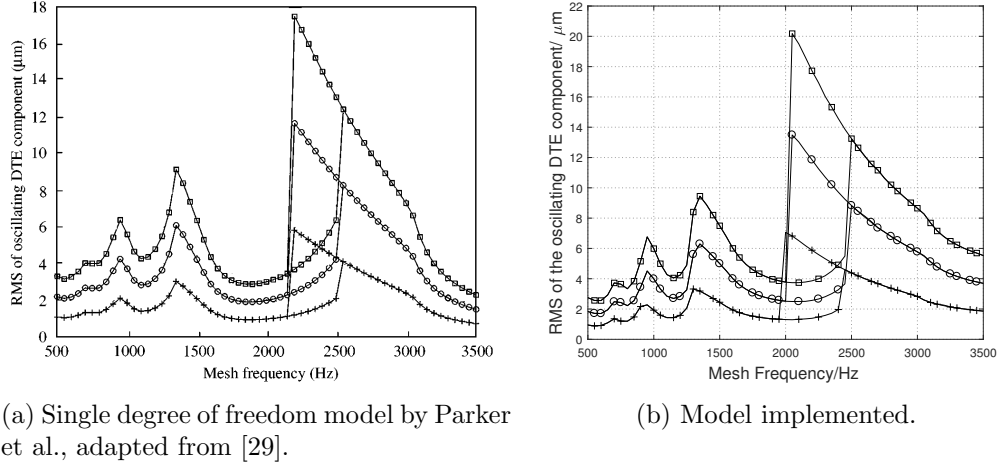


Figure 5.5: Comparison between the RMS of the oscillating DTE component for $T = 100 \text{ Nm}$ (+), $T = 200 \text{ Nm}$ (○), $T = 300 \text{ Nm}$ (□).

Observing figure 5.5 it can be affirmed that the overall shape of the curves are in agreement. A deeper analysis reveals that the maximum RMS for all the torques in the implemented model (figure 5.5b) is higher than in the curves presented by Parker et al. [29] (figure 5.5a). Although, the difference between the values of the maximum RMS for the torques applied is basically the same. The mesh frequency at which the jumps occur are slightly shifted to the left when compared with the single degree of freedom model presented by Parker et al. [29]. Regarding the resonance submultiples, at approximately 1450 Hz and 967 Hz, both the RMS of the oscillating DTE component and the mesh frequency are in agreement.

Comparing the mean DTE it is clear that the general behavior is very close for all the torques. The plotted curves for $T = 200 \text{ Nm}$ and $T = 300 \text{ Nm}$ (figure 5.6b) show the same differences relatively to the model that they are being compared (figure 5.6a). The progression of these curves is close to the ones in figure 5.6a, for instance, the changes in the resonance submultiples are in the same range of values and occur at the same mesh frequency. The main distinction is in the jump up non-linearity (Mesh Frequency $\simeq 2000$

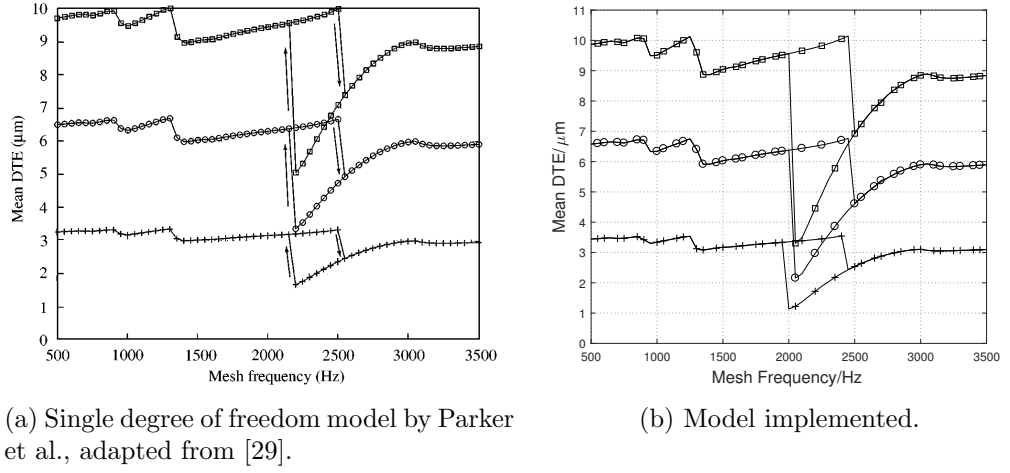


Figure 5.6: Comparison between the mean DTE for $T = 100 \text{ Nm}$ (+), $T = 200 \text{ Nm}$ (○), $T = 300 \text{ Nm}$ (□).

Hz) which takes place at a lower mesh frequency than it was supposed to. A similar decrease also happened in the value of the minimum mean DTE. The mean dynamic transmission error for $T = 100 \text{ Nm}$, besides showing the same divergences of the other torques, the positioning of the jump down is moderately inferior just as the values of the minimum mean DTE obtained.

It was already mentioned that these models present a high sensitivity to the mesh stiffness and the major discrepancy between the two implementations is, precisely, in the mesh stiffness modeling, as shown in figure 5.7. The shape of the mesh stiffness in figure 5.7b is not the same as the one in figure 5.7a but the mesh stiffness range is very similar for both models.

Concluding, in spite of the variations found, they are not significant and are most likely due to the different mesh stiffness approximations. Therefore the model is considered validated.

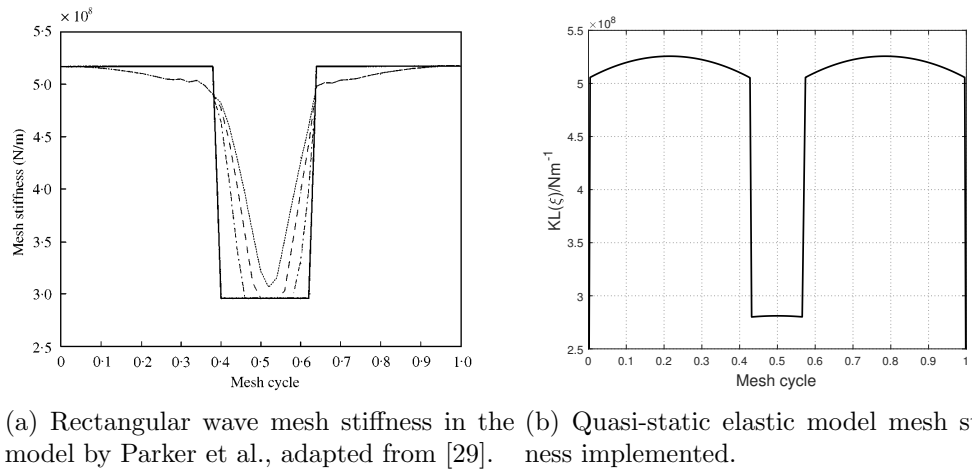


Figure 5.7: Comparison between the mesh stiffness for the gear pair (table 5.1) used in the models.

5.2 Results

Now that the implemented model was validated, the dynamic analysis of the CMSGears can take place. In order to perform the simulations, some additional data has to be defined, e.g., the speed range, the input torques, the response time and the numerical error tolerances.

The speed range was chosen considering that the resonance mesh frequency must be included in the interval, so an assessment regarding the natural frequencies of the gear pairs for analysis was conducted to make sure the resonance was not missed. The mesh frequency (f_{mesh}) sweep was fixed from 300 Hz to 9000 Hz with steps of 50 Hz for all the gear pairs.

For the selection of the loads, the torques that can be applied to the pinion in the FZG test rig available with an arm of 0.35 m were taken into account. Additionally, with the purpose of examining the dynamic behavior from lightly loaded to heavy loaded systems, three different torques were used, as it can be seen in table 5.2.

Table 5.2: Torques applied to the pinion in the dynamic model

K_1	K_6	$\simeq K_{11}$
3.3 Nm	98.82 Nm	318.52 Nm

The time which the solver is going to calculate the system response has to be enough for the system to achieve steady state. The time inserted is n (n being a natural number) mesh periods, the reason is that this way if at the last point determined the time equivalent to a mesh period (T_{mesh}) is removed, the last period of the dynamic response is readily obtained. The number of periods n was specified bearing in mind that the mesh period changes with the rotating speed as well as with the gear geometry. Hence, the number of periods was chosen in excess to compensate for these variations. After an analysis to the steady state time required by the gear pairs, the total response time selected was $100 T_{mesh}$.

The ODE solver was initially used with the default error tolerances, however for some gear pairs the error tolerances were not good enough, generating very significant numerical noise. Whenever this phenomenon happened, new simulations were performed with tighter error tolerances. The relative error tolerance was set to 10^{-8} and the absolute tolerance to 10^{-10} . An example of the difference in changing the error tolerances is presented for two gear pairs in figure 5.8. The gears with the designation “TOL” have a relative error tolerance of 10^{-6} and an absolute error tolerance of 10^{-8} while the others have 10^{-8} and 10^{-10} , respectively. The difference between these error tolerances is evident in figure 5.8.

In appendix C, all the parameters calculated for the dynamic performance of each gear pair studied are plotted and the curves for the different input torques can be compared.

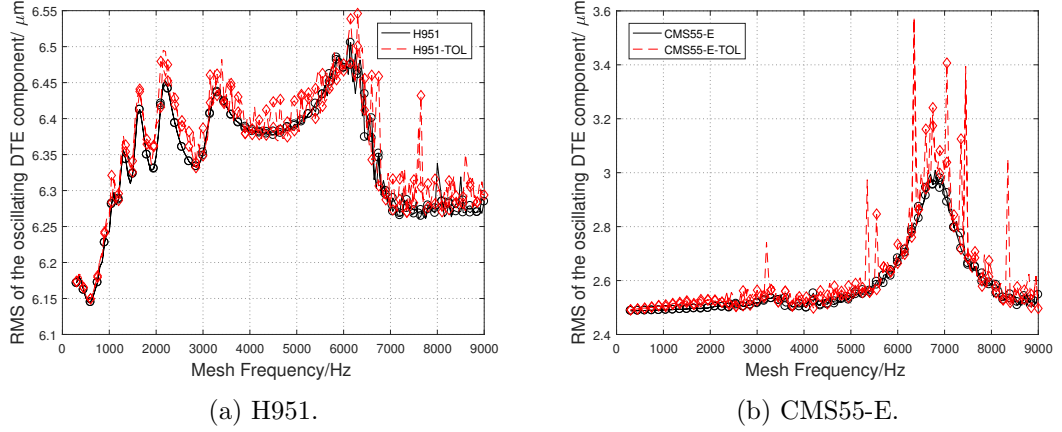


Figure 5.8: Difference of the error tolerances in the RMS of the oscillating DTE component.

5.2.1 Dynamic Transmission Error

In this section, not only the results related with the dynamic transmission error are examined but also the source of the non-linearity in the model is analyzed. A first analysis is performed for the CMSGears (table 4.4) and three other gear geometries, the C14, the H501 and the H951 (table 3.1).

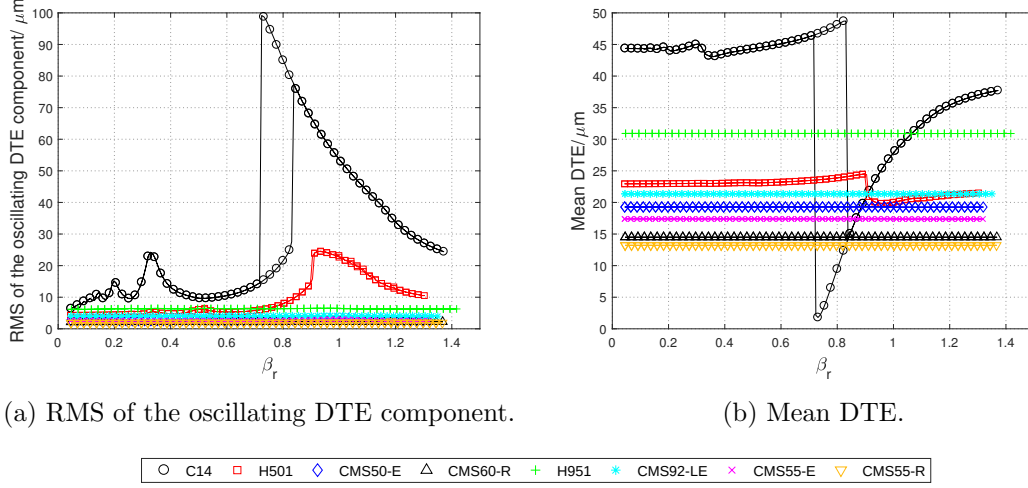
The following figures are represented in two different x axis, one of them is $\beta_r = f_{mesh}/f_n$ (frequency ratio) and the other is the rotating speed of the pinion in rpm. While the β_r axis allows a better comparison between the gears, the rpm axis enables a comparison considering the operating speed range. Table 5.3 shows the natural frequencies and the respective pinion rotating speed for all the gears being studied. This table is useful in the analysis of the β_r axis figures since it allows to locate the pinion operating speed from it.

Table 5.3: Natural frequency and respective pinion rotating speed

	C14	H501	CMS50-E	CMS60-R	H951	CMS92-LE	CMS55-E	CMS55-R
f_n/Hz	6568	6910	6822	6581	6351	6666	6828	6586
n_n/rpm	24606	20730	20467	16453	10027	10809	18621	17962

When figure 5.9 is examined, it becomes clear that two gears have a completely distinct behavior from the others. Those gears are the C14 (spur gear) and the H501 (conventional helical gear). In figure 5.9a the jump phenomena, the softening non-linearity and the high DTE variations of these gears stand out and do not allow a proper analysis of the remaining gears, for that reason figure 5.11a was plotted. However, in figure 5.9b, even though the previously mentioned gears still have a distinctive behavior, it is possible to make a comparison between all the gears.

Both the RMS of the oscillating DTE component and the mean DTE are similar for the C14 and the H501 but the non-linearities are clearer in the C14. Since the phenomenons behind the non-linearities for these gears are the same, a more detailed explanation is done for the C14. At the primary resonance ($\beta_r \simeq 1$), in figure 5.9a, the peak bends to the left, which represents a softening non-linearity. This kind of non-linearity is caused by the loss of stiffness that occurs due to the loss of contact at the teeth mesh. At the resonant submultiples there is also excitation in the system, however in this points the system does not present non-linearities. Moreover, there is the jump up ($\beta_r \simeq 0.8$) and the jump down

Figure 5.9: Analysis of the DTE fluctuations in order to β_r (K_{11}).

($\beta_r \simeq 0.7$) where it is relevant to point out that the system reaches the maximum RMS value for the speed down. In the experiments performed by Blankenship and Kahraman [30] the difference between the upper branch and lower branch was noticeable in the sound pressure level. This correlates well with the fact that DTE variations are the source of the noise generated by a gear pair. Comparing the jumps that occur in figures 5.9a and 5.9b, when the RMS jumps up (or down), the mean DTE jumps down (or up) but they still happen at the same frequency ratio.

Concerning the mean DTE, where all gears can be examined, the C14 shows an amplitude variation of around $45\mu\text{m}$ and the H501 of approximately $5\mu\text{m}$. For all the other gears the mean DTE is constant. This highlights the different behavior of the gears and allows to create a group of low fluctuation gears which is composed by the CMSGears and the H951. In spite of not being a CMSGear, the H951 has got $\varepsilon_\beta = 1.08$ and $\varepsilon_\alpha = 0.93$ which are both almost integer numbers. For that reason its dynamic performance is close to the CMSGears.

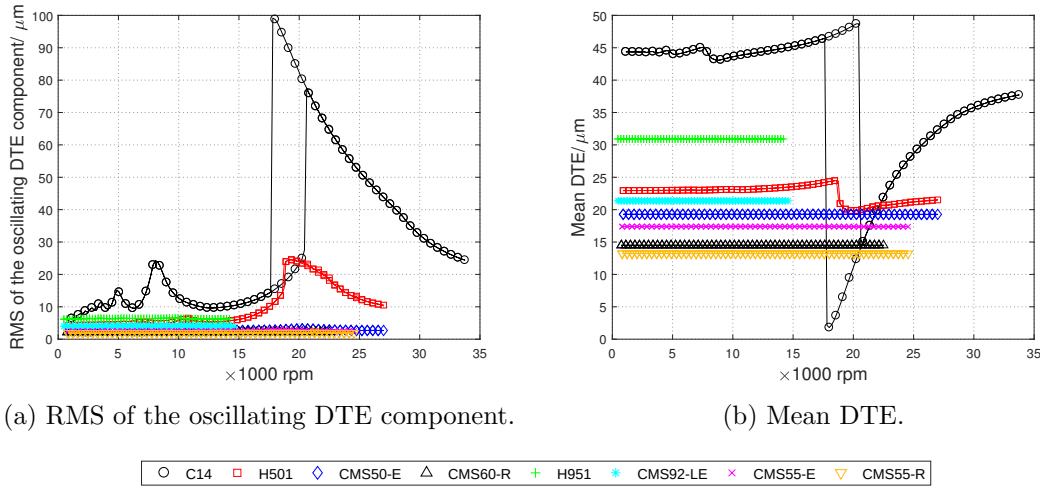
Figure 5.10: Analysis of the DTE fluctuations in order to the rotating speed (K_{11}).

Figure 5.10 and figure 5.9 represent the same curves but with different x axis. With

the rotating speed on the x axis the operating speed range for the gears is evident and a different perception of the previous figures is obtained. The differences in the rotating ranges are due to the fact that the speed evaluation interval was imposed through the mesh frequency. The relation between the mesh frequency and the pinion rotating speed is in equation (5.26). As the number of teeth enter in equation (5.26) the intervals are significantly changed.

$$n_1 = \frac{60 \cdot f_{mesh}}{z_1} \quad (5.26)$$

Once again, as the low fluctuation gears operate in a whole different range of values, the study of the RMS of the oscillating DTE component for these gears will be performed resorting to figure 5.11.

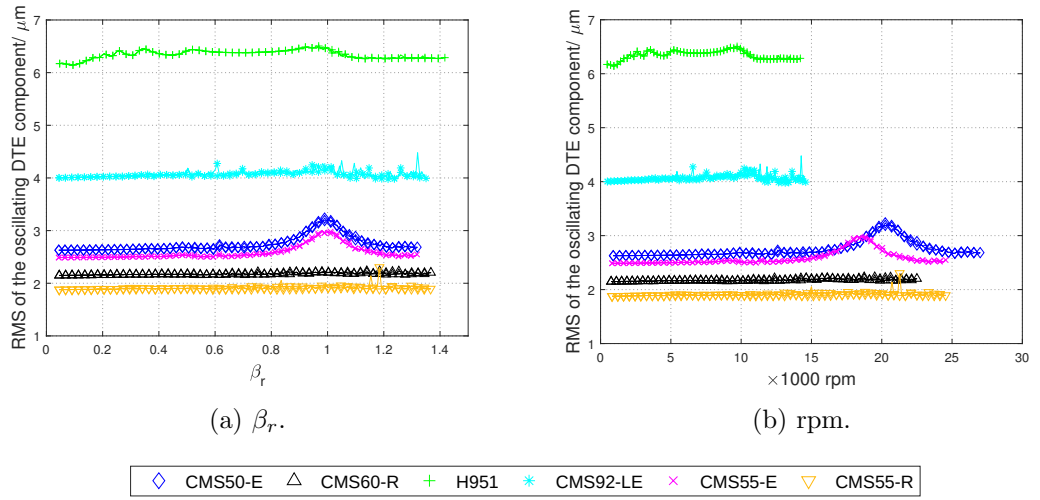


Figure 5.11: Analysis of the RMS of the oscillating DTE component focusing on low fluctuation gears (K_{11}).

Starting with figure 5.11a, the curves are not only at much lower values than the C14 and the H501 but also have less fluctuations and there are no records of non-linearities.

Three gear groups that exhibit identical responses can be formed. The first group which is composed by the high safety factor gears (CMS55-R and CMS60-R) reveals the lowest and a constant RMS of the oscillating DTE component. With a slighter higher value for the RMS, the second group (CMS55-E and CMS50-E) already display a peak for $\beta_r \simeq 1$. In the third group (H951 and CMS92-LE) is where the curves are more different from each other. While the CMS92-LE has a lower RMS value and more fluctuations around the resonant speed, the H951 shows greater variations distributed along its domain. In addition, if figure 5.9b is used to perform an evaluation, the gears that have smaller values for the RMS also present the lower mean DTE values.

The fact that these gears were grouped together is no coincidence. In fact, the first group is constituted by the robust CMSGears, which reminding have $\varepsilon_\beta = 2$. Regarding the second group, $\varepsilon_\beta = 1$, it corresponds to the efficient CMSGears. Finally, the third group is composed by the most efficient gears and the only ones that do not have the safety factors of the C14.

An analysis with the rotating pinion speed does not change much except for the efficient CMSGears. In figure 5.11a the maximum is lower for the CMS55-E than for the CMS50-E. In figure 5.11b the maximum keeps being lower for the CMS55-E but it happens at a lower

rotating speed. For a certain interval of the rotating speed the CMS50-E exhibits lower values than the CMS55-E due to the translation from the x axis switch.

The final figures for the assessment of the dynamic transmission error are relative to the maximum DTE (figures 5.12 and 5.13) and the maximum DTE*, which is represented in figures 5.14 and 5.15.

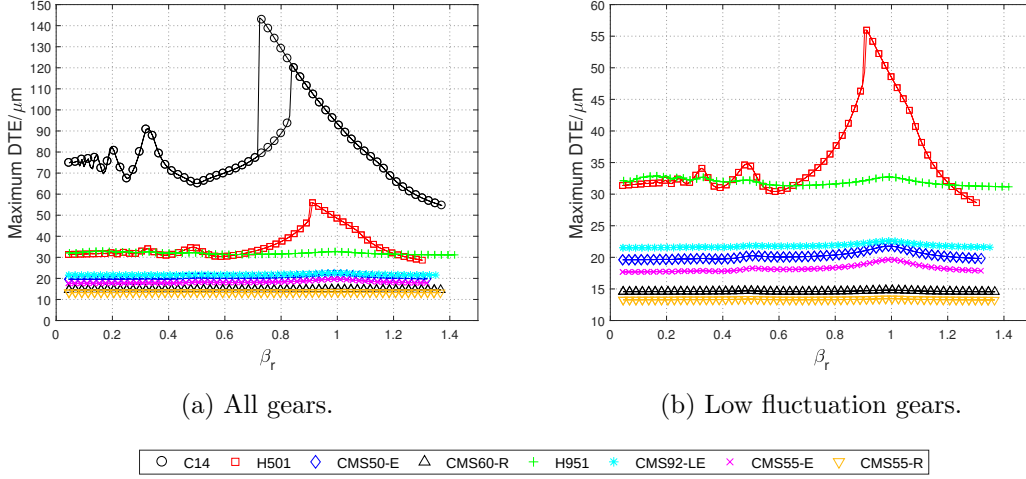


Figure 5.12: Analysis of the maximum DTE in order to β_r (K_{11}).

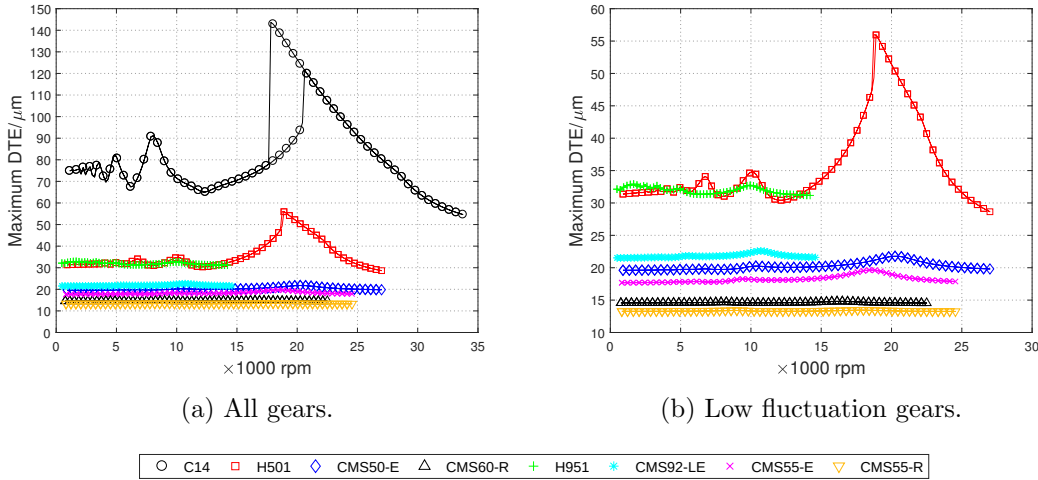
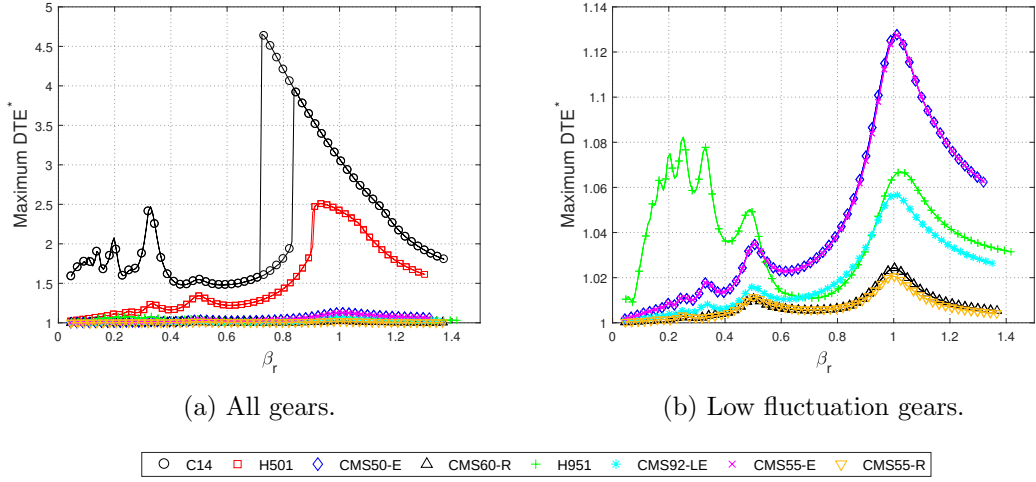
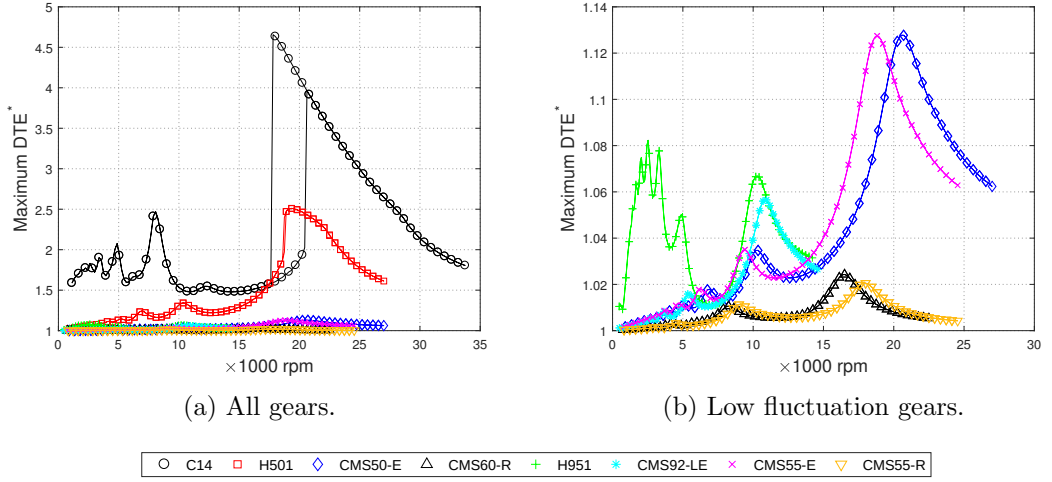


Figure 5.13: Analysis of the maximum DTE in order to the rotating speed (K_{11}).

Regarding the maximum DTE (figures 5.12 and 5.13) the CMSGears present the lowest values. Comparing the maximum DTE of the low fluctuation gears with the mean DTE for the same gears (figure 5.9b), the fact that the values shown are in the same range reveals that there are almost no fluctuations in the dynamic transmission error. On the other hand, the FZG C14 and the H501, but specially the C14, exhibit a considerable difference between the maximum and the mean DTE.

For the analysis of the maximum DTE*, take the H501 as an example. The maximum value registered is about 2.5, meaning that the dynamic displacement at this point is 2.5 times higher than the static displacement. Moving on to the low fluctuation gears (figure 5.14b), the CMS55-E and the CMS50-E exhibit the highest value among this type


 Figure 5.14: Analysis of the maximum DTE* in order to β_r (K_{11}).

 Figure 5.15: Analysis of the maximum DTE* in order to the rotating speed (K_{11}).

of gears with a value of around 1.13. In other words, the maximum normalized dynamic displacement is 13%, which, comparing with the robust (R-)CMSGears, is around 6 times higher. However, comparing the efficient (E-)CMSGears with the H951 and the CMS92-LE, even though the maximum dynamic displacement found is, in general, higher, when looking at the RMS of the oscillating DTE component and the mean DTE the E-CMSGears are both under the other group.

Figure 5.14b enhances the similarities between the aforementioned three gear groups. Although, it is important to point out the shape of the H951 curve since it differs from the other gears. As it was mentioned before, the H951 is not a CMSGear and for that reason its stiffness has more fluctuations. The response at low rotating speed is mainly dominated by the stiffness and the higher stiffness variations this gear presents, lead to a worst dynamic behavior, emphasized at low rotating speeds.

Figure 5.15 reveals the importance of selecting the gear with attention to the pretended operating speed.

All the previous figures were only for one input torque, $K_{11} \simeq 318.52$ Nm. However, for the other torques ($K_1 = 3.3$ Nm and $K_6 = 98.82$ Nm) the relative positioning of the

curves and the respective shape is kept the same, having only a vertical shift. This can be verified in figures 5.16, 5.17, 5.18 and 5.19. It is noteworthy here that due to its definition the maximum DTE^* is load independent.

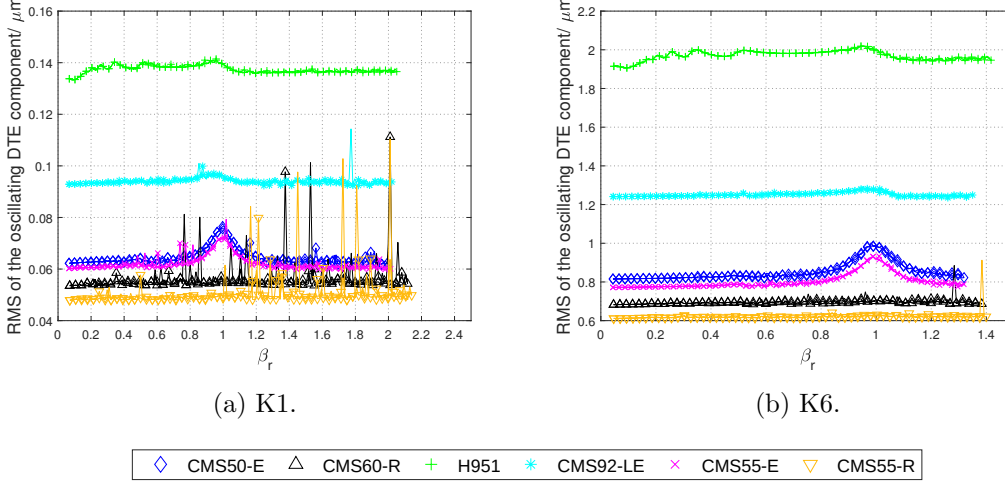


Figure 5.16: RMS of the oscillating DTE component for low fluctuation gears and different torques.

A few conclusions can be withdrawn from the dynamic transmission error study. The outcome of this first analysis already reveals the better performance of the CMSGears relatively to the gears studied. Besides having a lower dynamic transmission error, the CMSGears also have less DTE fluctuations. According to literature [31; 32], the transmission error is directly related to the gear noise and vibration. That being said, CMSGears will potentially generate lower sound pressure and work much smoother due to the lower and more stable dynamic transmission error.

The C14 and the H501 dynamic performance is worse than all other gears so these two gears will be removed from further analysis, with rare exceptions. Besides that, from now on all the curves will be plotted in order to the frequency ratio β_r in view of the fact that it allows a direct comparison between the different gears. Also, whenever the operating range is needed, table 5.3 helps in the determination of the respective range, apart from that, the figures previously shown already transmit the idea of the location of the rotating operating speed for each gear.

Effect of the contact loss function

Although the dynamic transmission error analysis is complete, a small test, to verify whether the source of the non-linearities is the contact loss function $g[x(t)]$ or not, is performed. For that purpose, the model was modified, this is, the contact loss function was removed. Subsequently, the new RMS of the oscillating DTE component curves are obtained and compared with the former respective curves for all gears, as it follows.

From the analysis of figure 5.20 it can be concluded that the gears that did not present non-linearities, when the contact loss function was removed, the non-linearities also did not appear since both curves are coincident. The opposite is not true, as the contact loss function is removed from the gears with non-linearities, C14 and H501, the non-linearities disappeared. This proves that the non-linearities are due to the contact loss between the meshing teeth.

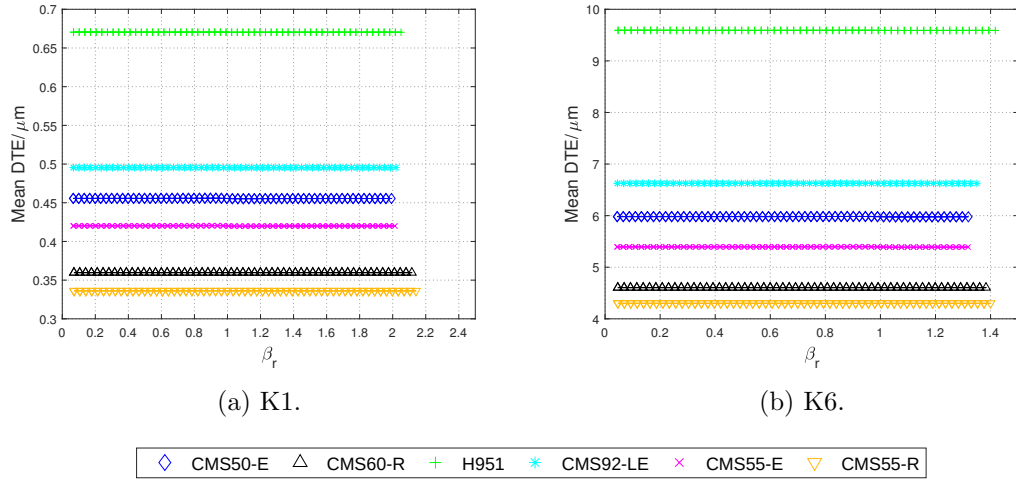


Figure 5.17: Mean DTE for low fluctuation gears and different torques.

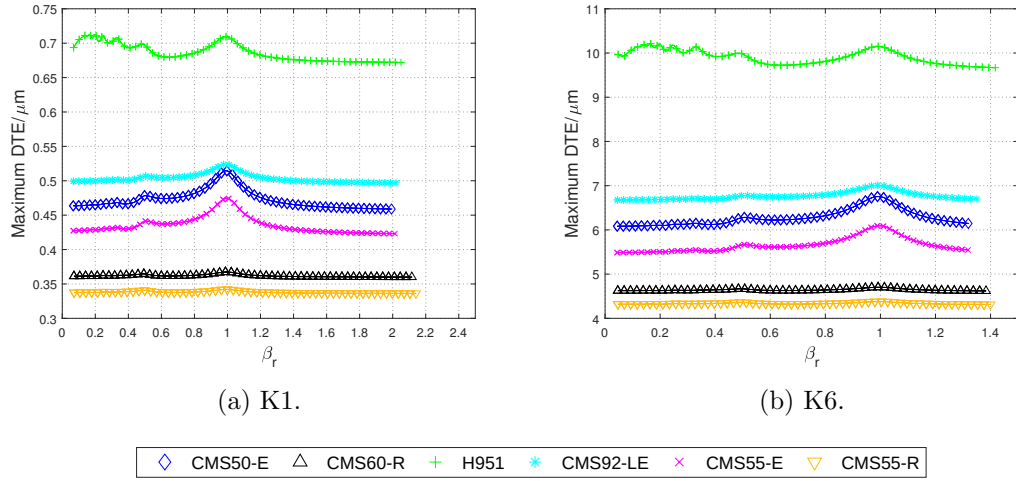


Figure 5.18: Maximum DTE for low fluctuation gears and different torques.

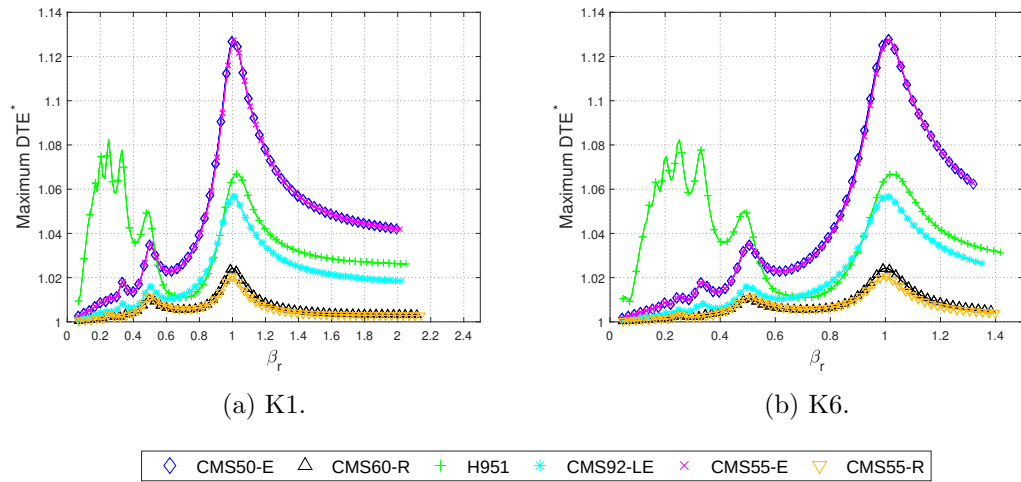
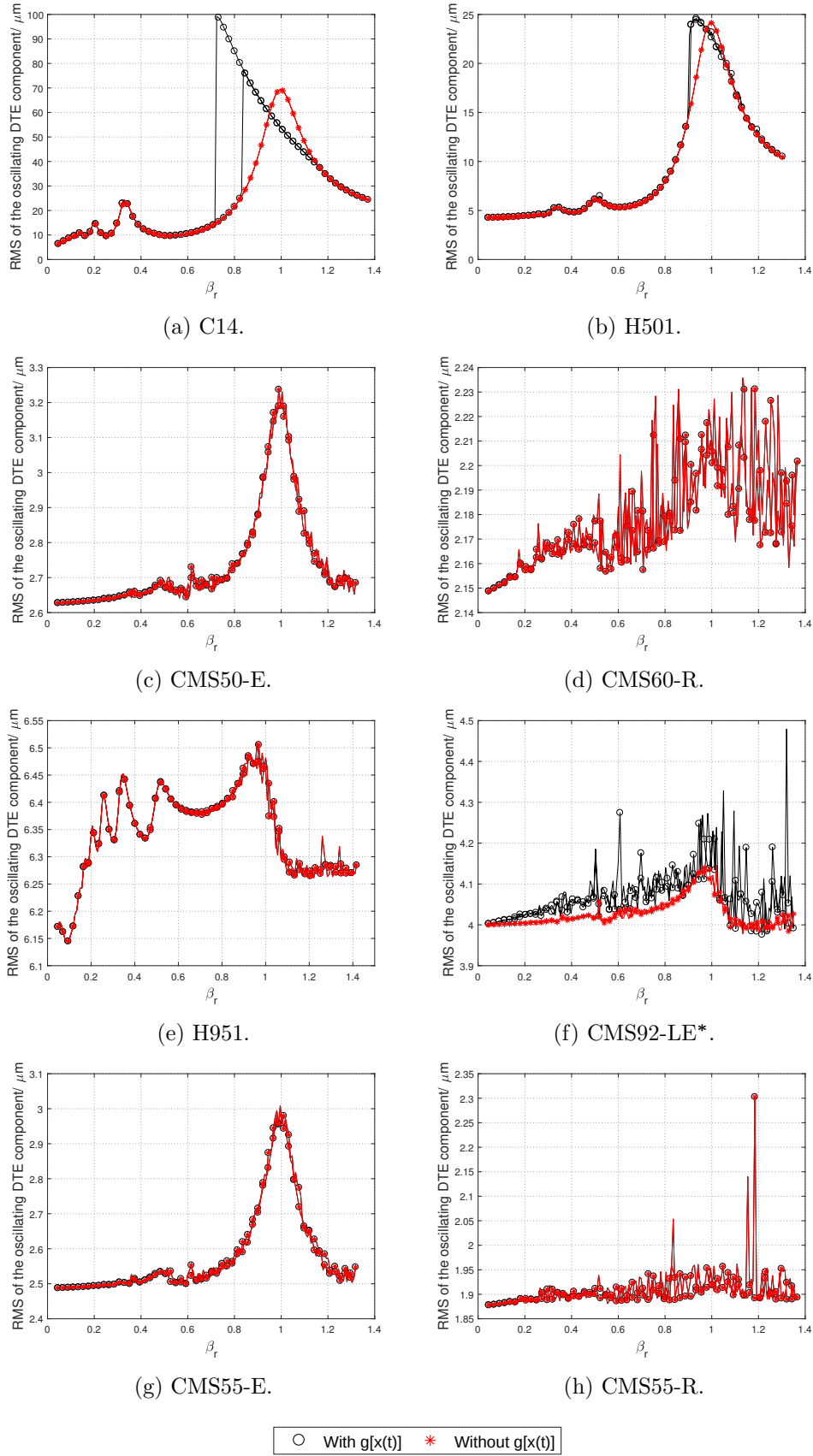


Figure 5.19: Maximum DTE^* for low fluctuation gears and different torques.



* - Different error tolerances;

Figure 5.20: Effect of the usage of the contact loss function, $g[x(t)]$.

5.2.2 Dynamic Mesh Force

In the study of the dynamic mesh force three parameters are evaluated, they are, as a resemblance to the DTE, the RMS of the oscillating DMF component, the mean DMF and the maximum DMF^* . Additionally, a comparison between the static and the dynamic surface normal force supported by a single pair of teeth is accomplished.

In figures 5.21a, 5.21b and 5.21c, a set of the aforementioned parameters is shown for all the gears and a single torque (K_{11}) so that the difference between the low fluctuation gears and the other two is fully clarified.

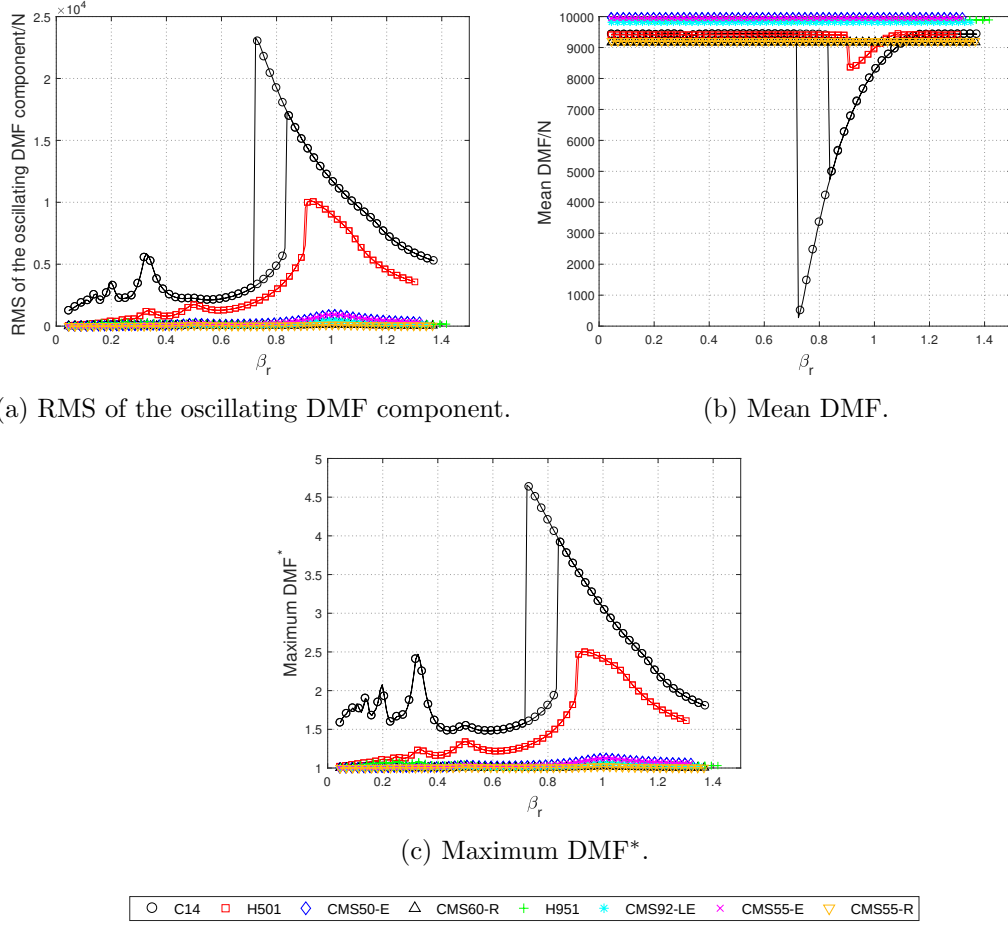


Figure 5.21: Dynamic mesh force analysis for all gears at K_{11} .

Figures 5.21a and 5.21c both keep the same trend as the corresponding figures in the DTE but regarding the mean DMF (figure 5.21b) the same can not be said since the relative positions of the curves is different, attend on figure 5.22. Notice that the high safety factors gears have the lowest mean DMF right under the C14 and the H501 and with the highest mean DMF, the four most efficient gears.

In the last section it was also observed that by varying the torque, the curves only shifted up and no relative change between the curves was detected. Thus, this analysis is conducted only for torque K_{11} . The figures for the remaining torques (figures 5.29 and 5.30) were still plotted and can be seen at the end of this section.

Looking at the RMS of the oscillating DMF component (figure 5.23) the configuration of the curves is basically the same for all analyzed gears when comparing with the analogous DTE curves. The most relevant difference is in the values reached by the E-CMSGears

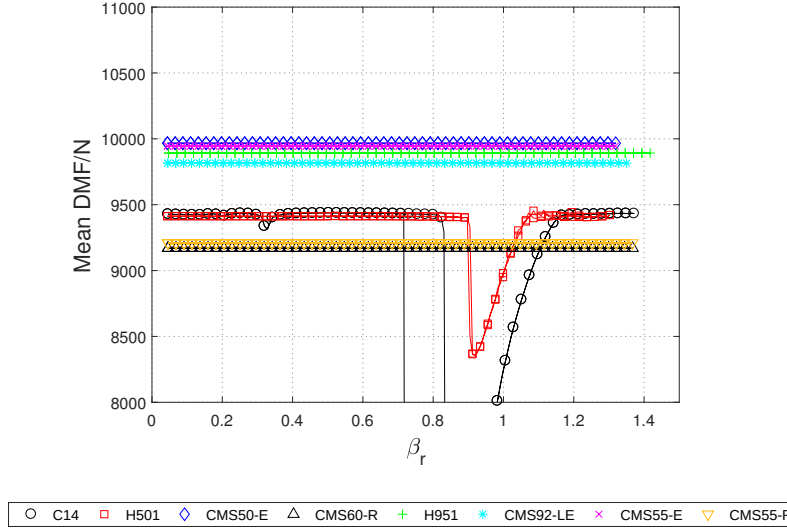


Figure 5.22: Zoom of the mean dynamic mesh force for all gears at K_{11} .

which were previously below the H951 and the CMS92-LE and are now above them. This change is related to the number of teeth and/or the module value of the gears (see tables 3.1 and 4.4). Despite that, figure 5.23 points out the gap between the formerly mentioned group of gears when they are both at the resonant rotating speed. Checking table 5.3 leads to the conclusion that the resonant rotating speeds of the gears are pretty distinct, in fact, the resonant rotating speed for the E-CMSGears is roughly the double of the H951 and the CMS92-LE. Therefore, comparing the values in $\beta_r = 0.5$ for the E-CMSGears and $\beta_r = 1$ for the other, which is about the same pinion rotating speed, the E-CMSGears present a lower RMS value.

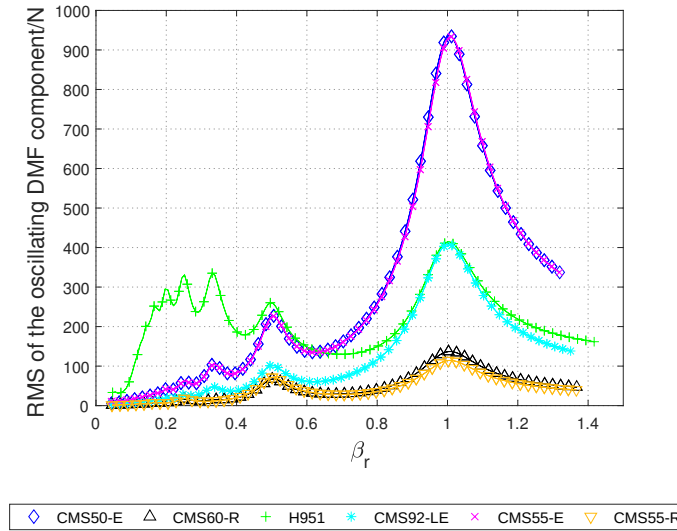


Figure 5.23: RMS of the oscillating DMF component for the low fluctuation gears (K_{11}).

The changes in the mean DMF (figure 5.24) are similar to the ones in the RMS of the oscillating DMF component, which means that these modifications are caused by the same phenomenon. Comparing the mean DTE with the mean DMF for the same torque,

the order of the curves is different just as it happened with the RMS of the oscillating component. However, it is important to note the difference between the lowest and the highest mean DTE and DMF for the low fluctuation gears. While the highest mean DTE curve is more than two times the value of the lowest DTE curve ($\simeq 2.286$, see figure 5.9b), for the mean DMF the highest value is $\simeq 1.087$ times lowest, see figure 5.24.

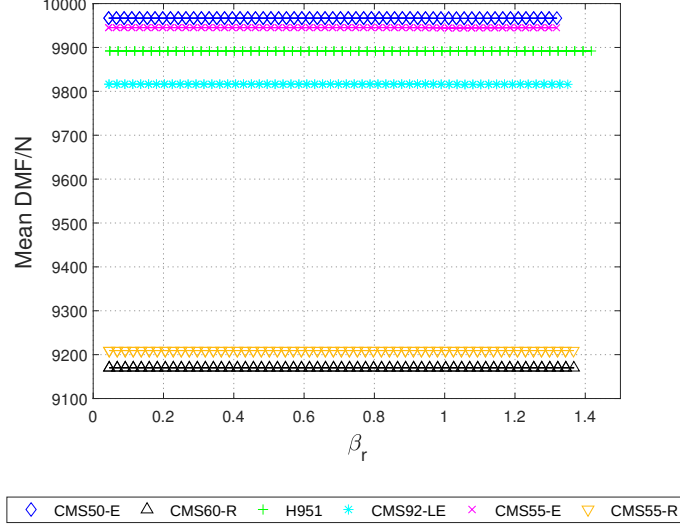


Figure 5.24: Mean DMF for the low fluctuation gears (K_{11}).

Analyzing the maximum DMF* (figure 5.25) it is verified that is exactly the same as the maximum DTE* due to the definition given to both of them, equation (5.23) and equation (5.21) respectively. It follows that with one figure the interpretation of two different parameters is performed. For this reason at the end of this section the figures concerning the maximum DMF* for the remaining torques are not exhibited.

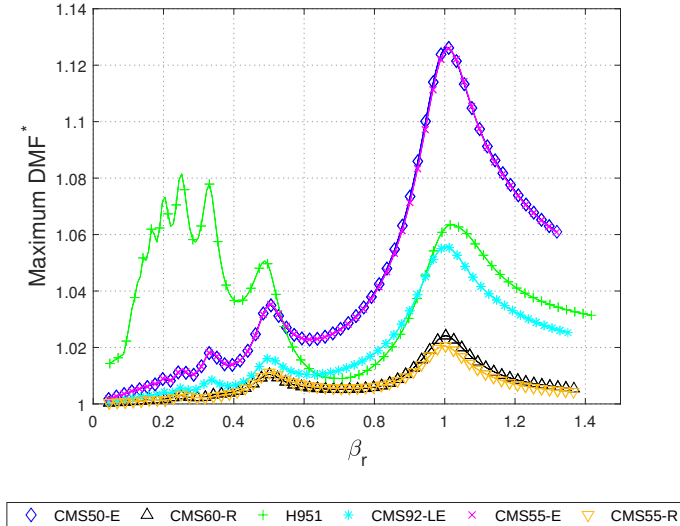


Figure 5.25: Maximum DMF* for the low fluctuation gears (K_{11}).

It is worthwhile to leave a comment about the position switch of the E-CMSGears with the H951 and the CMS92-LE. In the maximum DTE* (figure 5.14) and DMF* (figure 5.25)

the E-CMSGears have a higher value at $\beta_r = 1$ but comparing the mean DTE they are under and in the mean DMF they are above. With this information it can be concluded that in terms of absolute maximum DTE value, the E-CMSGears present lower dynamic transmission error since the mean DTE is lower. The opposite occurs for the maximum DMF, as the maximum dynamic mesh force will be superior for the CMS55-E and the CMS50-E.

Figure 5.26 compares the dynamic surface normal force for a single teeth pair with the its static load for $\beta_r = 1$. The torque chosen is indifferent since the shape is kept the same for all torque, as it can be seen in figures 5.27 and 5.28.

With figure 5.26 the reason why the C14 and the H501 have such values of the RMS of the oscillating DMF component is clear, as the dynamic curve is nowhere near the static. Note that these gears are the only ones that have negative forces, which are originated from the loss of contact between the gear teeth. However, the forces should not be negative (should be equal to zero), this only happens because the mesh stiffness model used to calculate these parameters does not take into consideration the contact loss, alike the mesh stiffness in the dynamic model. Besides that, due to the softening non-linearity the maximum RMS for these two gears is not at $\beta_r = 1$ meaning that far contrast would be found amid these curves.

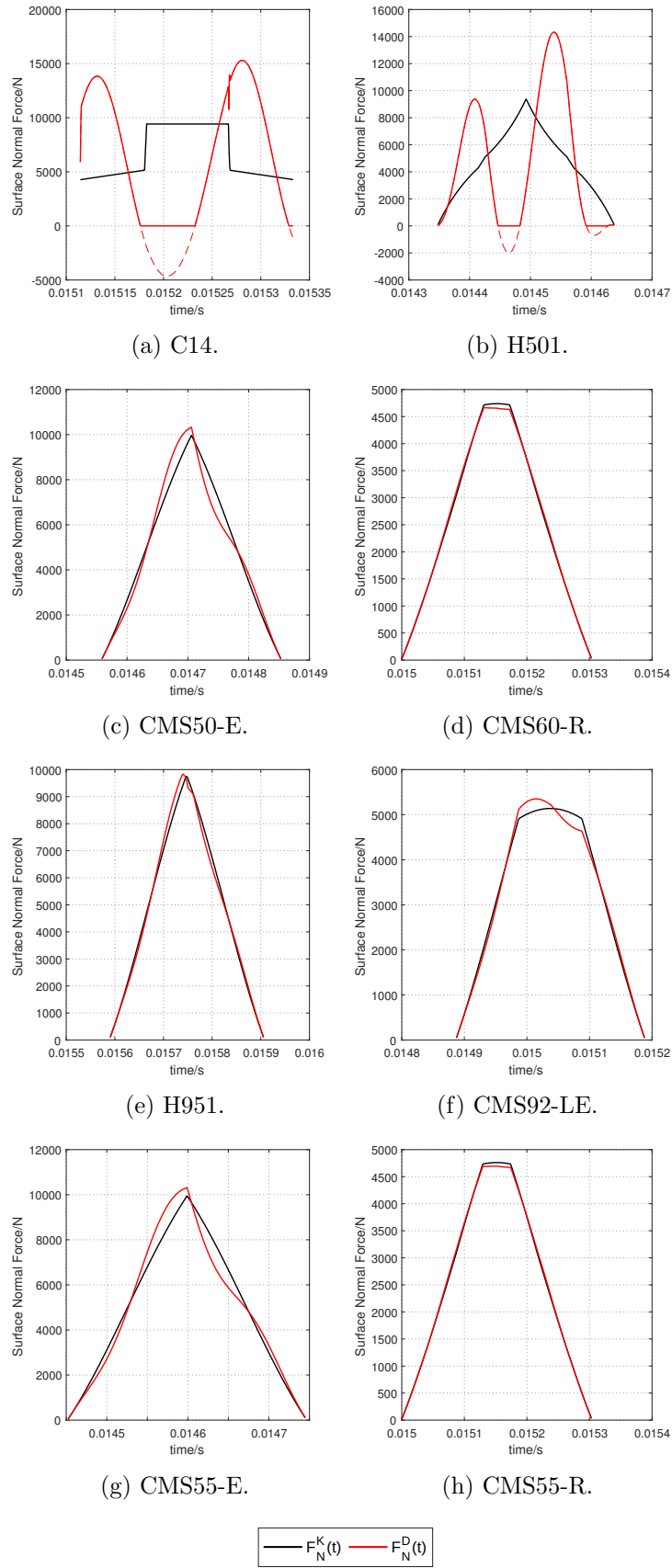


Figure 5.26: Comparison between the static and dynamic normal surface single tooth load $\beta_r = 1$ (K_{11}).

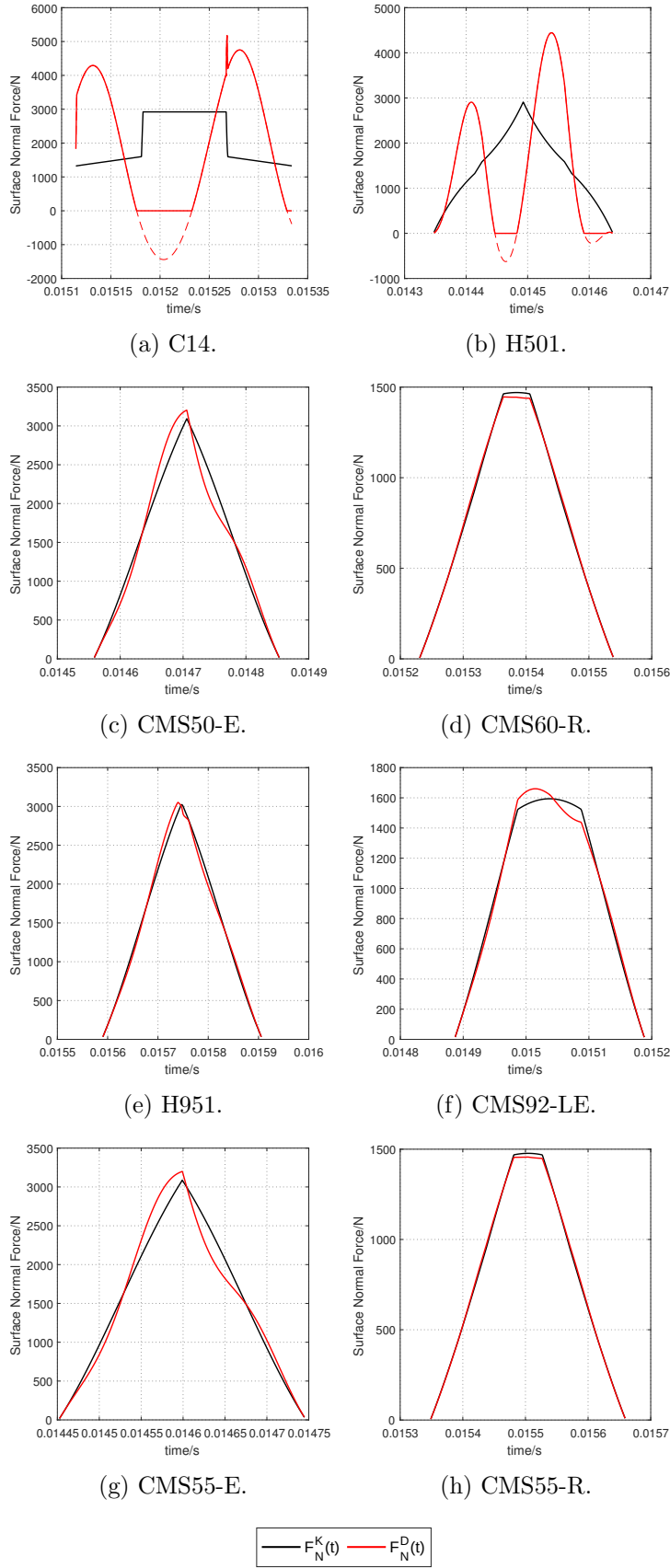


Figure 5.27: Comparison between the static and dynamic normal surface single tooth load $\beta_r = 1$ (K_6).

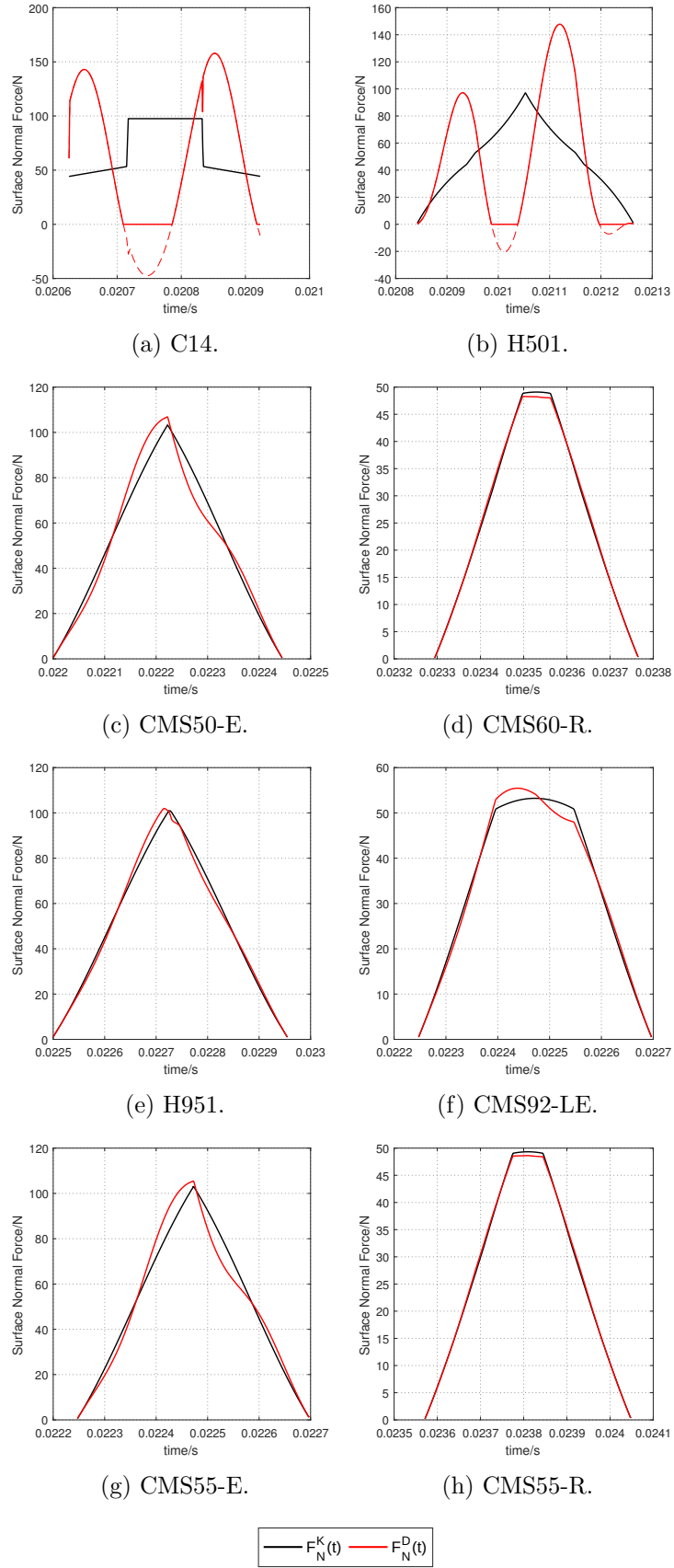


Figure 5.28: Comparison between the static and dynamic normal surface single tooth load $\beta_r = 1$ (K_1).

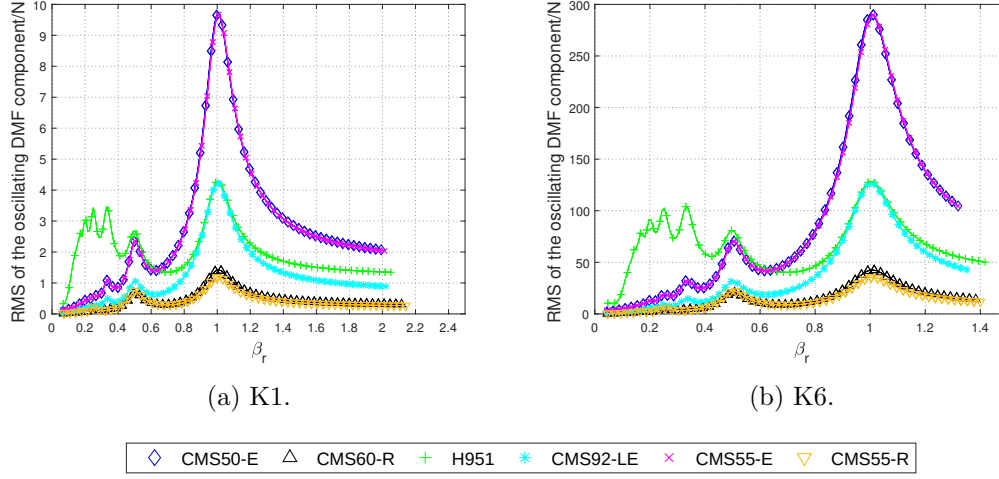


Figure 5.29: RMS of the oscillating DMF component for low fluctuation gears and different torques.

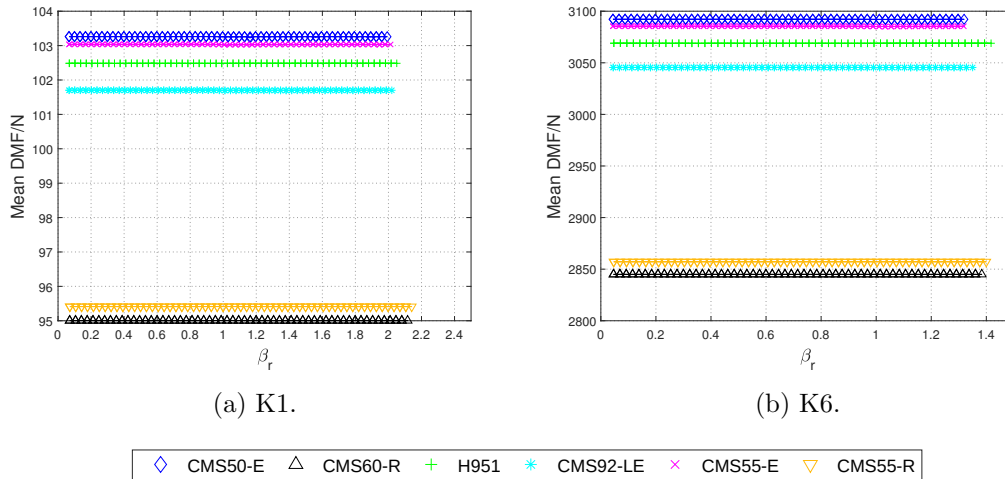


Figure 5.30: Mean DMF for low fluctuation gears and different torques.

5.2.3 Dynamic Contact Analysis

The CMSGears were designed to have an integer overlap ratio since, for now, that was the criteria used to develop this type of gears. However, while under operating conditions, the contact ratio and the overlap ratio can change owing to the load deformation. As any variation in the imposed integer value results in unwanted changes in the mesh stiffness, these effects have to be studied in order to understand their influence on the dynamic performance of the CMSGears.

A contact analysis evaluation was conducted on KISSsoft[®] for all the newly developed gears. The contact ratio (ε_α) increased with the load, so the maximum variation found was for K_{11} . Each group of CMSGears had around the same increase in the contact ratio, 10% for the R-CMSGears, 17% for the E-CMSGears and finally 30% for the CMS92-LE. Therefore, these were the variations that were imposed in the contact ratio for each group of CMSGears.

The effect of the three torques on the overlap ratio (ε_β) was evaluated and there were two cases (K_6 for the two E-CMSGears) where the overlap ratio increased ($\simeq 0.8\%$). For the remaining combinations of CMSGears and torques the overlap ratio presented lower values than the specified by the gear design. The maximum variation found was -2% . As a decrease in the value of the overlap ratio is more harmful than its increase, the variation imposed was of -2% for all CMSGears.

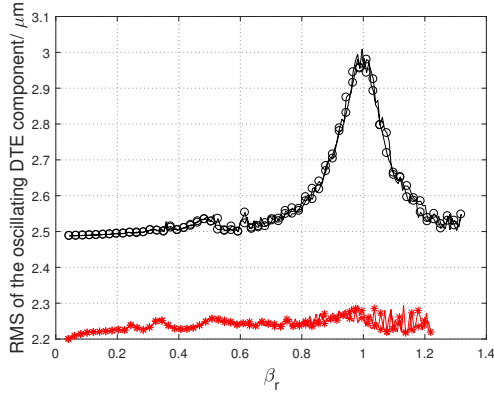
The torque correspondent to the maximum variation in the overlap ratio depended on the gear being analyzed, at least according to KISSsoft[®]'s contact analysis. However, since the contact ratio maximum variations are for K_{11} , that was the torque selected for the analysis of the change in the length of the contact lines. Table 5.4 presents the values to which the contact and the overlap ratios were set to for the simulations.

Table 5.4: Contact and overlap ratios of the contact analysis for the CMSGears

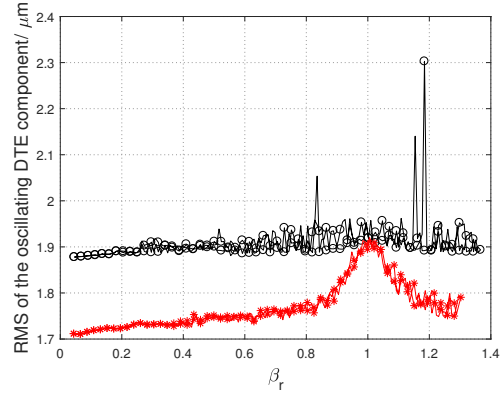
Gear	Contact Analysis		No Contact Analysis	
	ε_α	ε_β	ε_α	ε_β
CMS55-E	1.1700	0.9800	1.0000	1.0000
CMS55-R	1.6401	1.9600	1.4910	2.0000
CMS92-LE	1.3013	1.9600	1.0010	2.0000
CMS50-E	1.1700	0.9800	1.0000	1.0000
CMS60-R	1.6731	1.9600	1.5210	2.0000

In the dynamic model the contact ratio and the overlap ratio were forced to the values established in the table 5.4 so that the correspondent contact line length and resulting mesh stiffness variations would be implemented. The results obtained after running the program for the CMSGears are presented in figures 5.31, 5.32, 5.33, 5.34 and 5.35.

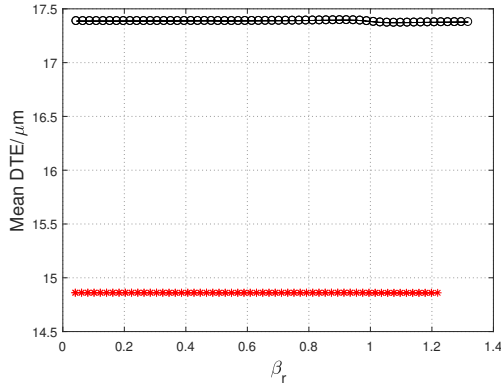
To aid in the interpretation of the results, the mesh stiffness considering the contact analysis modifications was plotted and compared with the design mesh stiffness, attend on figures 5.36, 5.37, 5.38, 5.39 and 5.40.



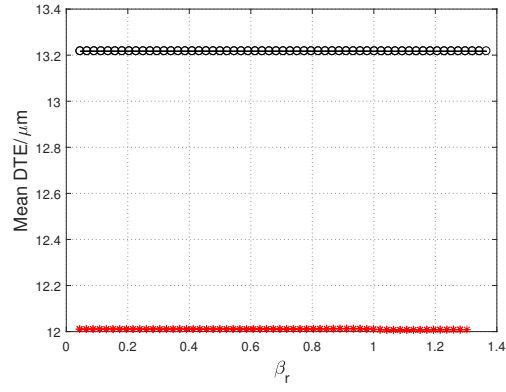
(a) RMS of the oscillating DTE component.



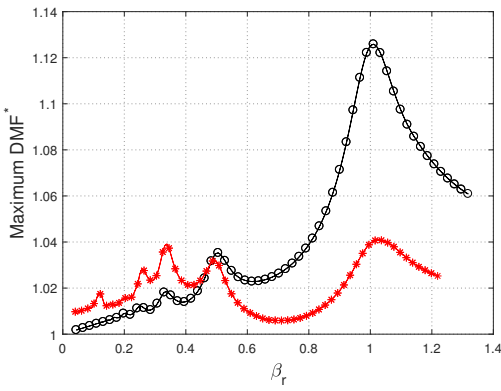
(a) RMS of the oscillating DTE component.



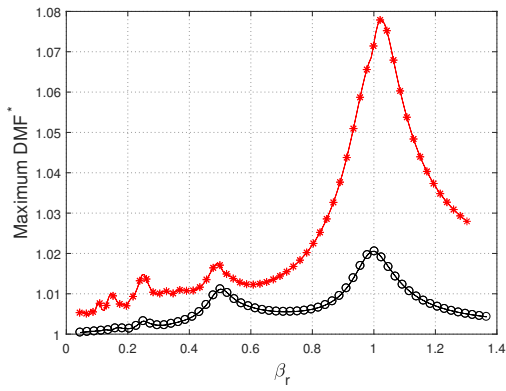
(b) Mean DTE.



(b) Mean DTE.



(c) Maximum DMF*.



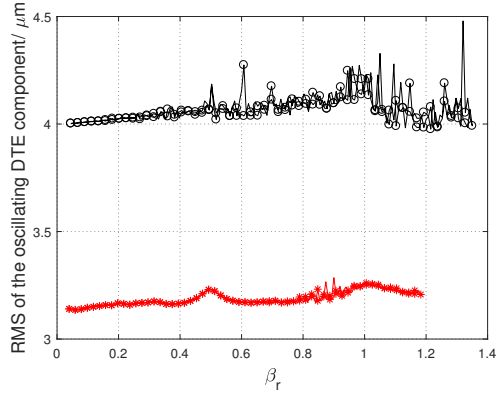
(c) Maximum DMF*.

○ Designed Gear * Load Deformation Gear

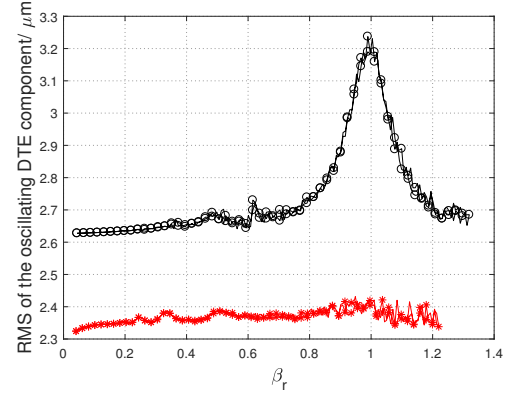
○ Designed Gear * Load Deformation Gear

Figure 5.31: Effect of the contact analysis in the dynamic performance of CMS55-E at K_{11} .

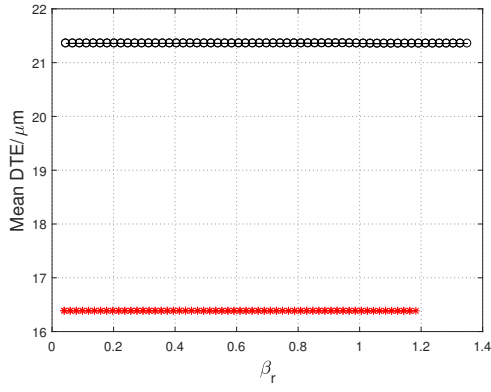
Figure 5.32: Effect of the contact analysis in the dynamic performance of CMS55-R at K_{11} .



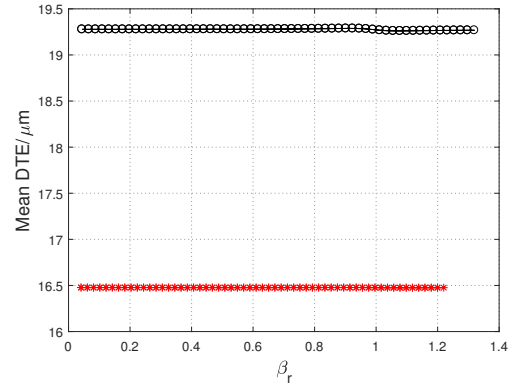
(a) RMS of the oscillating DTE component.



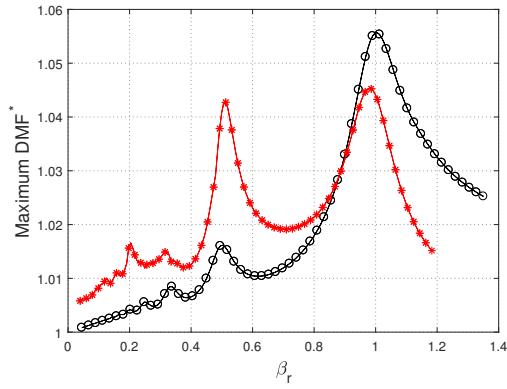
(a) RMS of the oscillating DTE component.



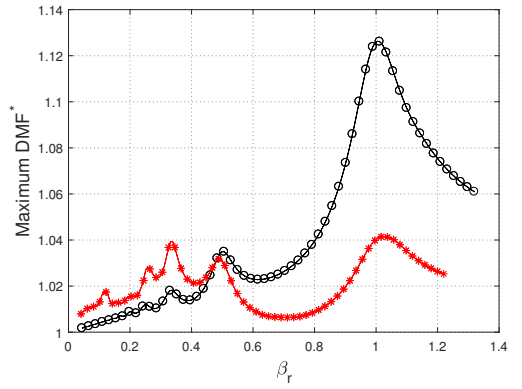
(b) Mean DTE.



(b) Mean DTE.



(c) Maximum DMF*.

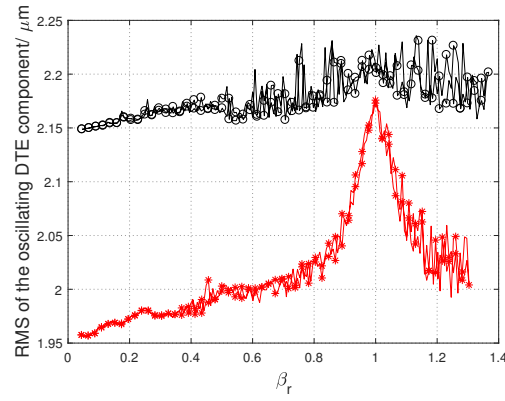


(c) Maximum DMF*.

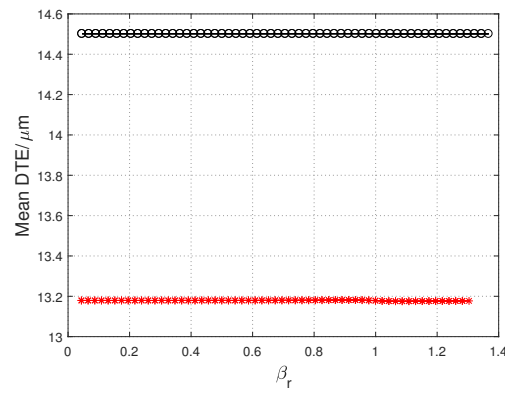


Figure 5.33: Effect of the contact analysis in the dynamic performance of CMS92-LE at K_{11} .

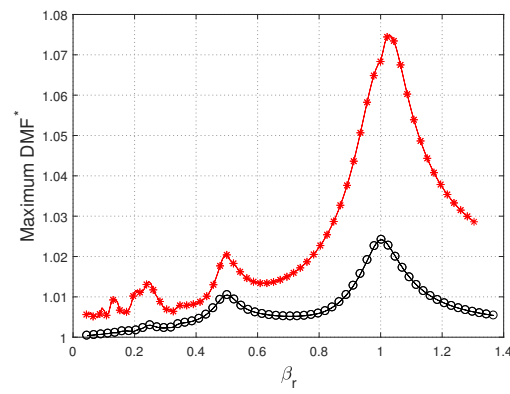
Figure 5.34: Effect of the contact analysis in the dynamic performance of CMS50-E at K_{11} .



(a) RMS of the oscillating DTE component.



(b) Mean DTE.



(c) Maximum DMF*.



Figure 5.35: Effect of the contact analysis in the dynamic performance of CMS60-R at K_{11} .

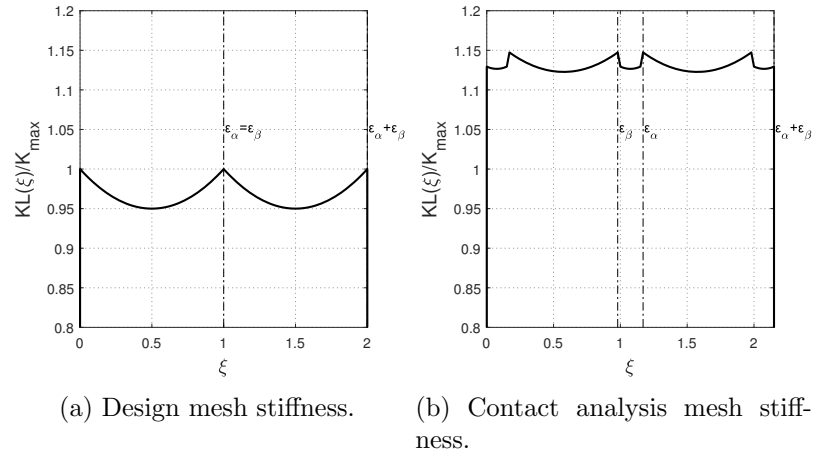


Figure 5.36: Comparison between the design and the contact analysis mesh stiffness for the CMS55-E.

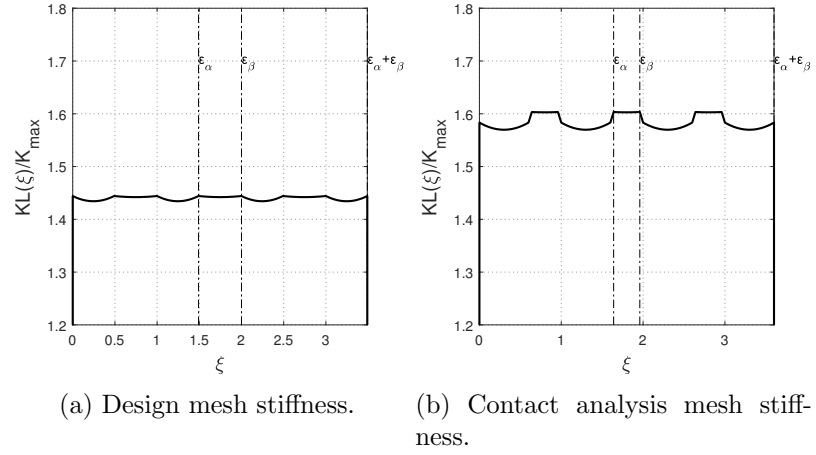


Figure 5.37: Comparison between the design and the contact analysis mesh stiffness for the CMS55-R.

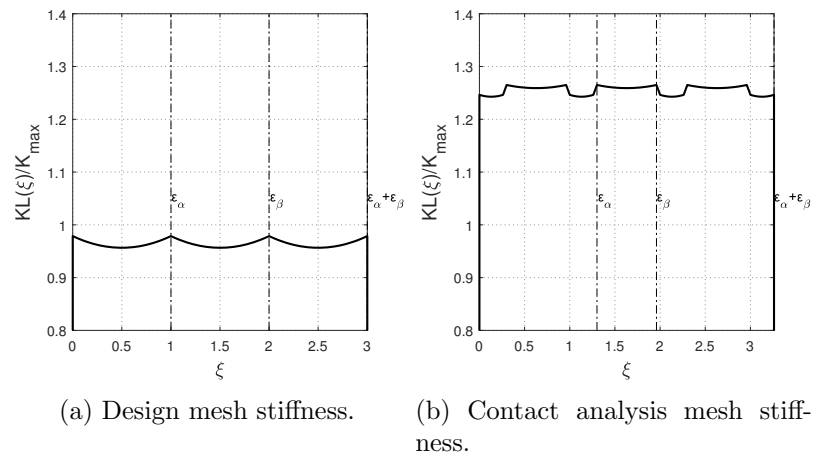


Figure 5.38: Comparison between the design and the contact analysis mesh stiffness for the CMS92-LE.

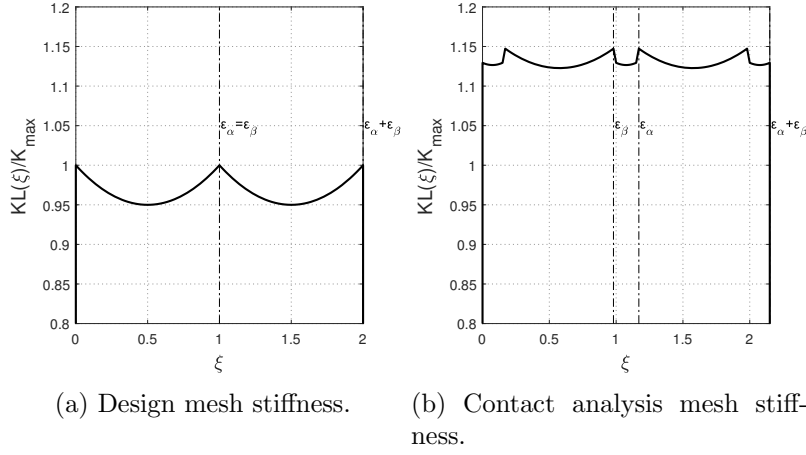


Figure 5.39: Comparison between the design and the contact analysis mesh stiffness for the CMS50-E.

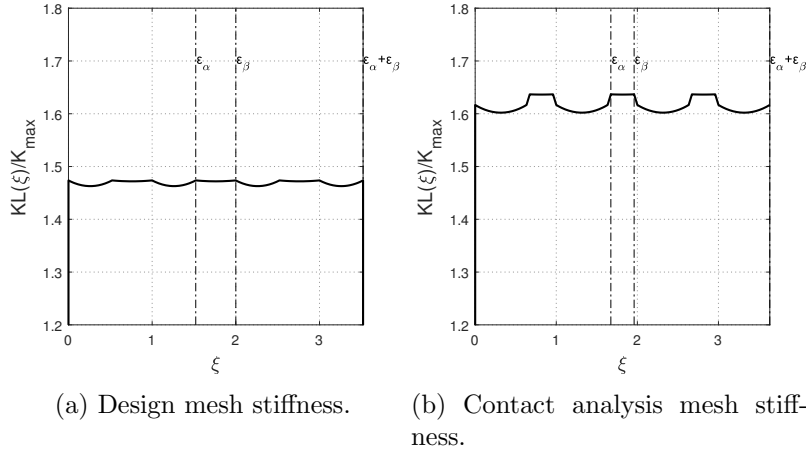


Figure 5.40: Comparison between the design and the contact analysis mesh stiffness for the CMS60-R.

There are two effects that lead to the results shown in figures 5.31 to 5.35: the first is the increase in the value of the mesh stiffness and the second is the increase in the mesh stiffness variation. Depending on the influence of these effects the results can be explained.

For all gears, in general, the RMS of the oscillating DTE component and the mean DTE are both lower when the variations in the contact lines are considered. This results from the fact that there was an increase in the mesh stiffness for all gears. The variations occurred in the mesh stiffness are less significant for the dynamic transmission error, mainly affecting the shape of the RMS curves. The increase in the mesh stiffness is proportional to the decrease in the mean DTE. Regarding the maximum DMF*, the stiffness variations have a much significant effect and will be evaluated for each CMSGear class.

The dynamic behaviors are similar for gears in the same class (E-CMSGears, R-CMSGears and CMS92-LE). Before starting the analysis of each group of gears, it must be clear that all gears have an increase in the mesh stiffness value because of the increase in the contact ratio. Therefore the next analysis will focus on the variations in the mesh stiffness introduced by the change in the contact lines length.

For the E-CMSGears, the mesh stiffness (figures 5.36 and 5.39) shows more but smaller variations. That is the reason why for low values of β_r the maximum DMF* reached is higher and for the area around the main resonance the maximum DMF* is lower. A similar behavior can also be observed for the RMS of the oscillating DTE component. At low frequency ratios the dynamic behavior is dictated by the stiffness and since the mesh stiffness shows more variations, the maximum DMF* is higher. When the area near the resonant speed is reached, the movement is mainly imposed by the system's equivalent mass, however the increase in the mesh stiffness is also reflected by the fact that the highest value of the maximum DMF* is lower.

Figures 5.37 and 5.40 reveal that the mesh stiffness of the R-CMSGears presents deeper variations. This modifications are reflected not only in the shape of RMS (figures 5.34a and 5.35a) but also in the maximum DMF* (figures 5.34c and 5.35c) where the values are superior for all the frequency ratios analyzed. In this case, the rise in the number of variations is overwhelming in relation to the mesh stiffness increase.

Finally, the CMS92-LE contact analysis mesh stiffness, figure 5.38b exhibits more variations but at approximately the same amplitude as in the design mesh stiffness, see figure 5.38a. As the influence of the increase in the mesh stiffness is similar to the elevation of the variations, this gear dynamic performance is a combination of the aforementioned CMSGears classes. For low values of β_r there is an increase in the maximum DMF*. Near $\beta_r = 1$ there is a slight reduction of the maximum DMF*. For this particular gear, for $\beta_r \simeq 0.5$ (resonant submultiple) the maximum DMF* is as high as for $\beta_r = 1$.

Concluding the dynamic contact analysis, the dynamic performance of these gears is not significantly affected by the variation of the contact lines length. All CMSGears exhibit lower RMS of the oscillating DTE component and mean DTE but the effect on the maximum DMF* differs from each group of CMSGears. On one hand, in the E-CMSGear the maximum DMF* of the contact analysis curve is higher for low values of β_r and lower around $\beta_r = 1$. On the other hand, for the R-CMSGears, when the changes in the contact line length are applied, the maximum DMF* increases for all the evaluated range. The modifications in the maximum DMF* for the CMS92-LE are similar to the E-CMSGears but the contact analysis curve for this gear is more affected by the changes in the length of the contact lines. To sum up, the R-CMSGears are more negatively affected by the change in the length of the contact lines than the remaining CMSGears.

Conclusions and Future Work

6.1 Conclusions

The main goal of this dissertation was to develop and study the concept of Constant Mesh Stiffness Gears. The key objectives of this dissertation were achieved.

The constant mesh stiffness gears were developed with success as the mesh stiffness obtained was basically constant. The program that defined these new gear geometries was capable of ensuring the minimum imposed safety factors while taking into account the gear efficiency. Besides the aforementioned, there were no problems with the fulfillment of the gear design principles, crucial for the correct performance of gears. Two kinds of CMSGears were developed, one that was more efficient and another that had higher safety factors.

The dynamic study revealed the huge gap between the CMSGears and the remaining gears studied. The CMSGears easily beat the non-CMSGears when it comes to the evaluation of the dynamic parameters. The root mean square of the oscillating DTE component was not only lower but also exhibited less variations for the CMSGears. Even the average dynamic transmission error was lower for all the new gears. The dynamic transmission error is related to the gear noise and vibration, therefore it is expected that these gears have lower sound pressure level and a much smoother operation. Concerning the dynamic mesh forces, the CMSGears still stand out although in the mean DMF, for certain operating speeds, some of the newly developed gears showed slightly higher values. The dynamic performance of the CMSGears while considering the variations in the contact and overlap ratios due to load deformation reinforced the fact that these gears are dynamically smooth and stable.

6.2 Future Work

This dissertation is the first step towards a whole new gear design and to take advantage of its potential a more detailed investigation must be conducted. In order to improve the CMSGear design and understand to the detail the dynamic phenomenons it would be useful to:

- Develop more CMSGears and compare them with other CMSGears as well as with non-CMSGears;
- Implement in the program that develops the CMSGears the possibility to include tip and root relief and optimize the profile modifications;

- Add rims to the CMSGears, reducing the effective mass of these gears;
- Improve the dynamic model by including more effects, such as, gear backlash, space dependent gear mesh, profile modifications and shaft and bearing stiffness;
- Perform a parametric study to fully comprehend the influence of the mesh stiffness in the CMSGear's dynamics;
- Finally, conduct experimental tests (acoustic, dynamic and power loss measurements) to adjust and validate the dynamic model and check in fact the performance of the CMSGears.

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Appendix A

Sum of the Unbounded Contact Length Ratio for Helical Gears

An example of the sum of the unbounded contact length ratio for helical gears, equation (A.1), is performed for $-2 \leq i \leq 2$ and assuming a overlap ratio equal to one ($\varepsilon_\beta = 1$).

$$\sum_{i=-2}^2 Ul_i^h(\xi) = \frac{1}{\epsilon_\beta} \cdot \left[H(\xi - i) \cdot (\xi - i) - H(\xi - \epsilon_\beta - i) \cdot (\xi - \epsilon_\beta - i) \right. \\ \left. - H(\xi - \epsilon_\alpha - i) \cdot (\xi - \epsilon_\alpha - i) \right. \\ \left. + H(\xi - (\epsilon_\alpha + \epsilon_\beta) - i) \cdot (\xi - (\epsilon_\alpha + \epsilon_\beta) - i) \right] \quad (\text{A.1})$$

$$\sum_{i=-2}^2 Ul_i^h(\xi) = H(\xi + 2)(\xi + 2) - H(\xi - \varepsilon_\beta + 2)(\xi - \varepsilon_\beta + 2) - H(\xi - \varepsilon_\alpha + 2)(\xi - \varepsilon_\alpha + 2) + \\ + H(\xi - (\varepsilon_\alpha + \varepsilon_\beta) + 2)(\xi - (\varepsilon_\alpha + \varepsilon_\beta) + 2) + H(\xi + 1)(\xi + 1) - H(\xi - \varepsilon_\beta + 1)(\xi - \varepsilon_\beta + 1) - \\ - H(\xi - \varepsilon_\alpha + 1)(\xi - \varepsilon_\alpha + 1) + H(\xi - (\varepsilon_\alpha + \varepsilon_\beta) + 1)(\xi - (\varepsilon_\alpha + \varepsilon_\beta) + 1) + \\ + H(\xi)(\xi) - H(\xi - \varepsilon_\beta)(\xi - \varepsilon_\beta) - H(\xi - \varepsilon_\alpha)(\xi - \varepsilon_\alpha) + H(\xi - (\varepsilon_\alpha + \varepsilon_\beta))(\xi - (\varepsilon_\alpha + \varepsilon_\beta)) + \\ + H(\xi - 1)(\xi - 1) - H(\xi - \varepsilon_\beta - 1)(\xi - \varepsilon_\beta - 1) - H(\xi - \varepsilon_\alpha - 1)(\xi - \varepsilon_\alpha - 1) + \\ + H(\xi - (\varepsilon_\alpha + \varepsilon_\beta) - 1)(\xi - (\varepsilon_\alpha + \varepsilon_\beta) - 1) + H(\xi - 2)(\xi - 2) - H(\xi - \varepsilon_\beta - 2)(\xi - \varepsilon_\beta - 2) - \\ - H(\xi - \varepsilon_\alpha - 2)(\xi - \varepsilon_\alpha - 2) + H(\xi - (\varepsilon_\alpha + \varepsilon_\beta) - 2)(\xi - (\varepsilon_\alpha + \varepsilon_\beta) - 2) \quad (\text{A.2})$$

Replacing $\varepsilon_\beta = 1$ in equation (A.2):

$$\sum_{i=-2}^2 Ul_i^h(\xi) = H(\xi + 2)(\xi + 2) - H(\xi + 1)(\xi + 1) - H(\xi - \varepsilon_\alpha + 2)(\xi - \varepsilon_\alpha + 2) + \\ + H(\xi - \varepsilon_\alpha + 1)(\xi - \varepsilon_\alpha + 1) + H(\xi + 1)(\xi + 1) - H(\xi)(\xi) - \\ - H(\xi - \varepsilon_\alpha + 1)(\xi - \varepsilon_\alpha + 1) + H(\xi - \varepsilon_\alpha)(\xi - \varepsilon_\alpha) + \\ + H(\xi)(\xi) - H(\xi - 1)(\xi - 1) - H(\xi - \varepsilon_\alpha)(\xi - \varepsilon_\alpha) + H(\xi - \varepsilon_\alpha - 1)(\xi - \varepsilon_\alpha - 1) + \\ + H(\xi - 1)(\xi - 1) - H(\xi - 2)(\xi - 2) - H(\xi - \varepsilon_\alpha - 1)(\xi - \varepsilon_\alpha - 1) + \\ + H(\xi - \varepsilon_\alpha - 2)(\xi - \varepsilon_\alpha - 2) + H(\xi - 2)(\xi - 2) - H(\xi - 3)(\xi - 3) - \\ - H(\xi - \varepsilon_\alpha - 2)(\xi - \varepsilon_\alpha - 2) + H(\xi - \varepsilon_\alpha - 3)(\xi - \varepsilon_\alpha - 3) \quad (\text{A.3})$$

Notice that not only the terms with the overlap ratio are going to be simplified. Even though the contact ratio does not have an attributed value, some of its terms can be eliminated. Proceeding to all the simplification possible follows equation (A.4).

$$\begin{aligned} \sum_{i=-2}^2 Ul_i^h(\xi) &= H(\xi + 2)(\xi + 2) - H(\xi - \varepsilon_\alpha + 2)(\xi - \varepsilon_\alpha + 2) - \\ &\quad - H(\xi - 3)(\xi - 3) + H(\xi - \varepsilon_\alpha - 3)(\xi - \varepsilon_\alpha - 3) \end{aligned} \quad (\text{A.4})$$

For i to belong to the interval defined in the beginning, the maximum value that the contact ratio can take, according to the definition given at i , is $\varepsilon_\alpha < 2$. From this, it can be concluded that the sum of the unbounded contact length ratio will never reach 3, thus the last two terms of equation (A.4) can be eliminated, which gives equation (A.5).

$$\sum_{i=-2}^2 Ul_i^h(\xi) = H(\xi + 2)(\xi + 2) - H(\xi - \varepsilon_\alpha + 2)(\xi - \varepsilon_\alpha + 2) \quad (\text{A.5})$$

Note that both terms have the same slope but one of them is positive and the other negative. Also, as $\varepsilon_\alpha < 2$ both of them begin before $\xi = 0$. Concluding, during its domain, the sum of the unbounded contact length ratio is constant.

Appendix B

KISSsoft® Reports

B.1 CMS55-E

Name : Unnamed

Changed by: up201306219

on: 21.04.2018

at: 12:43:36

CALCULATION OF A HELICAL GEAR PAIR

Drawing or article number:

Gear 1: 0.000.0

Gear 2: 0.000.0

Calculation method

ISO 6336:2006 Method B

----- GEAR 1 ----- GEAR 2 --

Power (kW)	[P]	60.040	
Speed (1/min)	[n]	1200.0	800.0
Torque (Nm)	[T]	477.8	716.7
Application factor	[KA]		1.25
Required service life (h)	[H]	20000.00	
Gear driving (+) / driven (-)		+	-
Working flank gear 1: Right flank			
Sense of rotation gear 1 clockwise			

1. TOOTH GEOMETRY AND MATERIAL

(geometry calculation according to ISO 21771:2007, DIN ISO 21771)

----- GEAR 1 ----- GEAR 2 --

Center distance (mm)	[a]	91.500	
Centre distance tolerance	ISO 286:2010 Measure js7		
Normal module (mm)	[mn]	3.0000	
Pressure angle at normal section (°)	[alfn]	20.0000	
Helix angle at reference circle (°)	[beta]	15.3177	
Number of teeth	[z]	22	33
Facewidth (mm)	[b]	35.68	35.68
Hand of gear		right	left
Accuracy grade	[Q-ISO 1328:1995]	6	6
Inner diameter (mm)	[di]	0.00	0.00
Inner diameter of gear rim (mm)	[dbi]	0.00	0.00

Material

Gear 1:	16 MnCr 5 (1), Case-carburized steel, case-hardened ISO 6336-5 Figure 9/10 (MQ), Core hardness >=25HRC Jominy J=12mm<HRC28
Gear 2:	16 MnCr 5 (1), Case-carburized steel, case-hardened ISO 6336-5 Figure 9/10 (MQ), Core hardness >=25HRC Jominy J=12mm<HRC28

----- GEAR 1 ----- GEAR 2 --

Surface hardness		HRC 59	HRC 59
Material quality according to ISO 6336:2006 Normal (Life factors ZNT and YNT >=0.85)			
Fatigue strength. tooth root stress (N/mm²)	[σFlim]	430.00	430.00
Fatigue strength for Hertzian pressure (N/mm²)	[σHlim]	1500.00	1500.00
Tensile strength (N/mm²)	[σB]	1000.00	1000.00
Yield point (N/mm²)	[σS]	695.00	695.00
Young's modulus (N/mm²)	[E]	206000	206000
Poisson's ratio	[ν]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.35	0.35

Roughness average value DS, root (µm)	[RAF]	3.00	3.00
Mean roughness height, Rz, flank (µm)	[RZH]	2.50	2.50
Mean roughness height, Rz, root (µm)	[RZF]	20.00	20.00

Gear reference profile 1 :

Reference profile 1.25 / 0.25 / 1.0 ISO 53:1998 Profil C

Dedendum coefficient	[hfP*]	1.250
Root radius factor	[rhofP*]	0.250 (rhofPmax*=0.472)
Addendum coefficient	[haP*]	1.000
Tip radius factor	[rhoaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[alfprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[alfKP]	0.000

not topping

Gear reference profile 2 :

Reference profile 1.25 / 0.25 / 1.0 ISO 53:1998 Profil C

Dedendum coefficient	[hfP*]	1.250
Root radius factor	[rhofP*]	0.250 (rhofPmax*=0.472)
Addendum coefficient	[haP*]	1.000
Tip radius factor	[rhoaP*]	0.000
Protuberance height coefficient	[hprP*]	0.000
Protuberance angle	[alfprP]	0.000
Tip form height coefficient	[hFaP*]	0.000
Ramp angle	[alfKP]	0.000

not topping

Summary of reference profile gears:

Dedendum reference profile	[hfP*]	1.250	1.250
Tooth root radius Refer. profile	[rofpP*]	0.250	0.250
Addendum Reference profile	[haP*]	1.000	1.000
Protuberance height coefficient	[hprP*]	0.000	0.000
Protuberance angle (°)	[alfprP]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[alfKP]	0.000	0.000

Type of profile modification: none (only running-in)

Tip relief (µm)	[Ca]	2.0	2.0
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Lubrication type Oil bath lubrication

Type of oil Oil: ISO-VG 220

Lubricant base Mineral-oil base

Kinem. viscosity oil at 40 °C (mm²/s)	[nu40]	220.00
Kinem. viscosity oil at 100 °C (mm²/s)	[nu100]	17.50
Specific density at 15 °C (kg/dm³)	[roOil]	0.895
Oil temperature (°C)	[TS]	70.000

----- GEAR 1 ----- GEAR 2 --

Overall transmission ratio	[itot]	-1.500
Gear ratio	[u]	1.500
Transverse module (mm)	[mt]	3.110
Pressure angle at pitch circle (°)	[alf]	20.675
Working transverse pressure angle (°)	[alfwt]	28.997
	[alfwt.e/i]	29.017 / 28.978
Working pressure angle at normal section (°)	[alfwn]	28.008

Helix angle at operating pitch circle (°)	[betaw]	16.330		
Base helix angle (°)	[betab]	14.373		
Reference centre distance (mm)	[ad]	85.539		
Sum of profile shift coefficients	[Summexi]	2.3895		
Profile shift coefficient	[x]	1.0924	1.2971	
Tooth thickness (Arc) (module) (module)	[sn*]	2.3660	2.5150	
Tip alteration (mm)	[k*mn]	-1.207	-1.207	
Reference diameter (mm)	[d]	68.431	102.646	
Base diameter (mm)	[db]	64.024	96.036	
Tip diameter (mm)	[da]	78.571	114.015	
(mm)	[da.e/i]	78.571 / 78.561	114.015 / 114.005	
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.010	0.000 / -0.010	
Tip form diameter (mm)	[dFa]	78.571	114.015	
(mm)	[dFa.e/i]	78.571 / 78.561	114.015 / 114.005	
Active tip diameter (mm)	[dNa]	78.571	114.015	
Active tip diameter (mm)	[dNa.e/i]	78.571 / 78.561	114.015 / 114.005	
Operating pitch diameter (mm)	[dw]	73.200	109.800	
(mm)	[dw.e/i]	73.214 / 73.186	109.821 / 109.779	
Root diameter (mm)	[df]	67.485	102.929	
Generating Profile shift coefficient	[xE.e/i]	1.0603/ 1.0420	1.2651/ 1.2468	
Manufactured root diameter with xE (mm)	[df.e/i]	67.293 / 67.183	102.737 / 102.627	
Theoretical tip clearance (mm)	[c]	0.750	0.750	
Effective tip clearance (mm)	[c.e/i]	0.923 / 0.828	0.923 / 0.828	
Active root diameter (mm)	[dNf]	69.585	105.291	
(mm)	[dNf.e/i]	69.621 / 69.557	105.328 / 105.262	
Root form diameter (mm)	[dFf]	68.472	103.971	
(mm)	[dFf.e/i]	68.281 / 68.174	103.763 / 103.646	
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.724 / 0.638	0.841 / 0.749	
Addendum (mm)	[ha=mn*(haP*+x+k)]	5.070	5.684	
(mm)	[ha.e/i]	5.070 / 5.065	5.684 / 5.679	
Dedendum (mm)	[hf=mn*(hfP*-x)]	0.473	-0.141	
(mm)	[hf.e/i]	0.569 / 0.624	-0.045 / 0.010	
Roll angle at dFa (°)	[xsi_dFa.e/i]	40.760 / 40.744	36.664 / 36.653	
Roll angle to dNa (°)	[xsi_dNa.e/i]	40.760 / 40.744	36.664 / 36.653	
Roll angle to dNf (°)	[xsi_dNf.e/i]	24.476 / 24.330	25.807 / 25.711	
Roll angle at dFf (°)	[xsi_dFf.e/i]	21.239 / 20.961	23.443 / 23.257	
Tooth height (mm)	[h]	5.543	5.543	
Virtual gear no. of teeth	[zn]	24.308	36.462	
Normal tooth thickness at tip circle (mm)	[san]	2.324	2.408	
(mm)	[san.e/i]	2.251 / 2.199	2.337 / 2.287	
Normal-tooth thickness on tip form circle (mm)	[sFan]	2.324	2.408	
(mm)	[sFan.e/i]	2.251 / 2.199	2.337 / 2.287	
Normal space width at root circle (mm)	[efn]	1.974	1.989	
(mm)	[efn.e/i]	1.977 / 1.979	1.984 / 1.982	
Max. sliding velocity at tip (m/s)	[vga]	1.054	0.861	
Specific sliding at the tip	[zetaaa]	0.368	0.335	
Specific sliding at the root	[zetaf]	-0.503	-0.583	
Mean specific sliding	[zetam]	0.353		
Sliding factor on tip	[Kga]	0.229	0.187	
Sliding factor on root	[Kgf]	-0.187	-0.229	
Pitch on reference circle (mm)	[pt]	9.772		
Base pitch (mm)	[pbt]	9.143		
Transverse pitch on contact-path (mm)	[pet]	9.143		
Lead height (mm)	[pz]	784.890	1177.335	
Axial pitch (mm)	[px]	35.677		
Length of path of contact (mm)	[ga, e/i]	9.143 (9.180 / 9.089)		

Length T1-A, T2-A (mm)	[T1A, T2A]	13.629(	13.593/	13.675)	30.727(	30.727/	30.718)
Length T1-B (mm)	[T1B, T2B]	13.630(	13.630/	13.675)	30.726(	30.690/	30.718)
Length T1-C (mm)	[T1C, T2C]	17.743(	17.728/	17.757)	26.614(	26.592/	26.636)
Length T1-D (mm)	[T1D, T2D]	22.772(	22.736/	22.764)	21.585(	21.585/	21.628)
Length T1-E (mm)	[T1E, T2E]	22.773(	22.773/	22.764)	21.584(	21.548/	21.628)
Length T1-T2 (mm)	[T1T2]		44.357 (	44.320 /	44.393)		
Diameter of single contact point B (mm)	[d-B]	69.586(	69.586/	69.621)	114.014(	113.975/	114.005)
Diameter of single contact point D (mm)	[d-D]	78.570(	78.528/	78.561)	105.292(	105.292/	105.328)
Addendum contact ratio	[eps]	0.550(	0.552/	0.548)	0.450(	0.452/	0.447)
Minimal length of contact line (mm)	[Lmin]			36.833			
Transverse contact ratio	[eps_a]			1.000			
Transverse contact ratio with allowances	[eps_a.e/m/i]		1.004 /	0.999 /	0.994		
Overlap ratio	[eps_b]			1.000			
Total contact ratio	[eps_g]			2.000			
Total contact ratio with allowances	[eps_g.e/m/i]		2.004 /	1.999 /	1.994		

2. FACTORS OF GENERAL INFLUENCE

		----- GEAR 1 -----	GEAR 2 --
Nominal circum. force at pitch circle (N)	[Ft]		13963.9
Axial force (N)	[Fa]		3824.7
Radial force (N)	[Fr]		5269.6
Normal force (N)	[Fnorm]		15407.4
Nominal circumferential force per mm (N/mm)	[w]		391.40
Only as information: Forces at operating pitch circle:			
Nominal circumferential force (N)	[Ftw]		13054.1
Axial force (N)	[Faw]		3824.7
Radial force (N)	[Frw]		7235.2
Circumferential speed reference circle (m/s)	[v]		4.30
Circumferential speed operating pitch circle (m/s)	[v(dw)]		4.60
Running-in value (μm)	[yp]		0.6
Running-in value (μm)	[yf]		0.6
Correction coefficient	[CM]		0.800
Gear body coefficient	[CR]		1.000
Basic rack factor	[CBS]		0.975
Material coefficient	[E/Est]		1.000
Singular tooth stiffness (N/mm/μm)	[c']		16.077
Meshing stiffness (N/mm/μm)	[cgalf]		16.078
Meshing stiffness (N/mm/μm)	[cgbet]		13.666
Reduced mass (kg/mm)	[mRed]		0.01459
Resonance speed (min-1)	[nE1]		14409
Resonance ratio (-)	[N]		0.083
Subcritical range			
Running-in value (μm)	[ya]		0.6
Bearing distance l of pinion shaft (mm)	[l]		71.400
Distance s of pinion shaft (mm)	[s]		7.140
Outside diameter of pinion shaft (mm)	[dsh]		35.700
Load in accordance with Figure 13, ISO 6336-1:2006 0:a), 1:b), 2:c), 3:d), 4:e)	[-]	4	
Coefficient K' according to Figure 13, ISO 6336-1:2006	[K']	-1.00	
Without support effect			
Tooth trace deviation (active) (μm)	[Fby]		3.61
from deformation of shaft (μm)	[fsh*B1]		3.38
(fsh (μm) = 3.38, B1= 1.00, fHb5 (μm) = 6.00)			

Tooth without tooth trace modification			
Position of Contact pattern: favorable			
from production tolerances (μm)	[fma*B2]	12.02	
(B2= 1.00)			
Tooth trace deviation, theoretical (μm)	[Fbx]	4.25	
Running-in value (μm)	[yb]	0.64	
Dynamic factor	[KV]	1.032	
Face load factor - flank	[KHb]	1.049	
- Tooth root	[KFb]	1.041	
- Scuffing	[KBb]	1.049	
Transverse load factor - flank	[KHa]	1.000	
- Tooth root	[KF _a]	1.000	
- Scuffing	[KB _a]	1.000	
Helical load factor scuffing	[Kbg]	1.003	
Number of load cycles (in mio.)	[NL]	1440.000	960.000

3. TOOTH ROOT STRENGTH

Calculation of Tooth form coefficients according method: B

		----- GEAR 1 -----	GEAR 2 --
Calculated with profile shift	[x]	1.0924	1.2971
Tooth form factor	[YF]	1.32	1.32
Stress correction factor	[YS]	2.62	2.53
Working angle (°)	[alfFen]	31.90	29.90
Bending moment arm (mm)	[hF]	4.24	4.23
Tooth thickness at root (mm)	[sFn]	7.23	7.28
Tooth root radius (mm)	[roF]	0.76	0.85
(hF* = 1.413/ 1.409 sFn* = 2.411/ 2.427 roF* = 0.254/ 0.282)			
(den (mm) =			
82.412/ 120.149 dsFn(mm) = 68.127/ 103.638 alfsFn(°) = 30.00/ 30.00 qs = 4.755/ 4.308)			
Helix angle factor	[Ybet]	0.872	
Deep tooth factor	[YDT]	1.000	
Gear rim factor	[YB]	1.00	1.00
Effective facewidth (mm)	[beff]	35.68	35.68
Nominal stress at tooth root (N/mm²)	[sigF0]	392.23	380.89
Tooth root stress (N/mm²)	[sigF]	526.73	511.50
Permissible bending stress at root of Test-gear			
Notch sensitivity factor	[YdreIT]	1.018	1.015
Surface factor	[YRrelT]	0.957	0.957
size factor (Tooth root)	[YX]	1.000	1.000
Finite life factor	[YNT]	0.884	0.891
	[YdreIT*YRrelT*YX*YNT]	0.861	0.865
Alternating bending factor (mean stress influence coefficient)	[YM]	1.000	1.000
Stress correction factor	[Yst]	2.00	
Yst*sigFlim (N/mm²)	[sigFE]	860.00	860.00
Permissible tooth root stress (N/mm²)	[sigFP=sigFG/SFmin]	528.84	531.45
Limit strength tooth root (N/mm²)	[sigFG]	740.38	744.03
Required safety	[SFmin]	1.40	1.40
Safety for tooth root stress	[SF=sigFG/sigF]	1.41	1.45

Transmittable power (kW)	[kWRating]	60.28	62.38
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4. SAFETY AGAINST PITTING (TOOTH FLANK)

		----- GEAR 1 -----	GEAR 2 --
Zone factor	[ZH]		1.998
Elasticity factor ($\sqrt{N/mm^2}$)	[ZE]		189.812
Contact ratio factor	[Zeps]		1.000
Helix angle factor	[Zbet]		1.018
Effective facewidth (mm)	[beff]		35.68
Nominal contact stress (N/mm^2)	[sigH0]		1192.43
Contact stress at operating pitch circle (N/mm^2)	[sigHw]		1386.96
Single tooth contact factor	[ZB,ZD]	1.00	1.00
Contact stress (N/mm^2)	[sigHB, sigHD]	1386.96	1386.96
Lubrication coefficient at NL	[ZL]	1.020	1.020
Speed coefficient at NL	[ZV]	0.979	0.979
Roughness coefficient at NL	[ZR]	1.016	1.016
Material pairing coefficient at NL	[ZW]	1.000	1.000
Finite life factor	[ZNT]	0.902	0.913
	[ZL * ZV * ZR * ZNT]	0.915	0.927
Limited pitting is permitted:	No		
Size factor (flank)	[ZX]	1.000	1.000
Permissible contact stress (N/mm^2)	[sigHP=sigHG/SHmin]	1372.97	1390.15
Pitting stress limit (N/mm^2)	[sigHG]	1372.97	1390.15
Required safety	[SHmin]	1.00	1.00
Safety factor for contact stress at operating pitch circle			
	[SHw]	0.99	1.00
Safety for stress at single tooth contact	[SHBD=sigHG/sigHBD]	0.99	1.00
(Safety regarding transmittable torque)	[(SHBD)^2]	0.98	1.00
Transmittable power (kW)	[kWRating]	58.83	60.32

4b. MICROPITTING ACCORDING TO ISO/TR 15144-1:2014

Calculation did not run. (Lubricant: Load stage micropitting test is unknown.)

5. SCUFFING LOAD CAPACITY

Calculation method according to ISO TR 13989:2000

Lubrication coefficient (for lubrication type)	[XS]	1.000	
Scuffing test and load stage	[FZGtest]	FZG - Test A / 8.3 / 90 (ISO 14635 - 1)	12
Multiple meshing factor	[Xmp]	1.000	
Relative structure coefficient (Scuffing)	[XWrelT]	1.000	
Thermal contact factor ($N/mm/s^{.5/K}$)	[BM]	13.780	13.780
Relevant tip relief (μm)	[Ca]	2.00	2.00
Optimal tip relief (μm)	[Ceff]	30.43	
Ca taken as optimal in the calculation (0=no, 1=yes)		0	0
Effective facewidth (mm)	[beff]	35.677	
Applicable circumferential force/facewidth (N/mm)	[wBt]	529.522	
Kbg =	1.003, wBt*Kbg =	531.332	

Angle factor ($\epsilon_1:0.550, \epsilon_2:0.450$)	[Xalfbet]	1.098
Flash temperature-criteria		
Lubricant factor	[XL]	0.830
Tooth mass temperature (°C) ($theMi = theoil + XS \cdot 0.47 \cdot Xmp \cdot theflm$)	[theMi]	78.34
Average flash temperature (°C)	[theflm]	17.74
Scuffing temperature (°C)	[theS]	348.80
Coordinate gamma (point of highest temp.) [Gamma.A]=-0.232 [Gamma.E]=0.284	[Gamma]	0.284
Highest contact temp. (°C)	[theB]	115.68
Flash factor ($^{\circ}K \cdot N^{\wedge}-.75 \cdot s^{\wedge}.5 \cdot m^{\wedge}-.5 \cdot mm$)	[XM]	50.058
Approach factor	[XJ]	1.000
Load sharing factor	[XGam]	1.000
Dynamic viscosity (mPa*s)	[etaM]	41.90 (70.0 °C)
Coefficient of friction	[mym]	0.055
Required safety	[SBmin]	2.000
Safety factor for scuffing (flash temperature)	[SB]	6.104
Integral temperature-criteria		
Lubricant factor	[XL]	1.000
Tooth mass temperature (°C) ($theMC = theoil + XS \cdot 0.70 \cdot theflaint$)	[theMC]	81.59
Mean flash temperature (°C)	[theflaint]	16.55
Integral scuffing temperature (°C)	[theSint]	360.78
Flash factor ($^{\circ}K \cdot N^{\wedge}-.75 \cdot s^{\wedge}.5 \cdot m^{\wedge}-.5 \cdot mm$)	[XM]	50.058
Running-in factor (well run in)	[XE]	1.000
Contact ratio factor	[Xeps]	0.459
Dynamic viscosity (mPa*s)	[etaOil]	41.90 (70.0 °C)
Mean coefficient of friction	[mym]	0.056
Geometry factor	[XBE]	0.168
Meshing factor	[XQ]	1.000
Tip relief factor	[XCa]	1.059
Integral tooth flank temperature (°C)	[theint]	106.41
Required safety	[SSmin]	1.800
Safety factor for scuffing (intg.-temp.)	[SSint]	3.390
Safety referring to transmittable torque	[SSL]	7.985

6. MEASUREMENTS FOR TOOTH THICKNESS

		----- Gear 1 ----- Gear 2 --	
		DIN 3967 cd25	DIN 3967 cd25
Tooth thickness deviation		-0.070 / -0.110	-0.070 / -0.110
Tooth thickness allowance (normal section) (mm)	[As.e/i]		
Number of teeth spanned	[k]	5.000	6.000
Base tangent length (no backlash) (mm)	[Wk]	43.120	52.909
Actual base tangent length ('span') (mm)	[Wk.e/i]	43.055 / 43.017	52.843 / 52.806
(mm)	[ΔWk.e/i]	-0.066 / -0.103	-0.066 / -0.103
Diameter of measuring circle (mm)	[dMWk.m]	76.400	108.818
Theoretical diameter of ball/pin (mm)	[DM]	7.033	6.663
Effective diameter of ball/pin (mm)	[DMeff]	7.500	7.000
Radial single-ball measurement backlash free (mm)	[MrK]	43.534	60.696
Radial single-ball measurement (mm)	[MrK.e/i]	43.477 / 43.444	60.633 / 60.597
Diameter of measuring circle (mm)	[dMMr.m]	75.350	110.706

Diametral measurement over two balls without clearance (mm)	[MdK]	87.068	121.262
Diametral two ball measure (mm)	[MdK.e/i]	86.954 / 86.888	121.137 / 121.065
Diametral measurement over pins without clearance (mm)	[MdR]	87.068	121.392
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	86.954 / 86.888	121.266 / 121.195
Measurement over 2 pins (free) according to AGMA 2002 (mm)	[dk2f.e/i]	0.000 / 0.000	121.124 / 121.052
Measurement over 2 pins (transverse) according to AGMA 2002 (mm)	[dk2t.e/i]	0.000 / 0.000	121.392 / 121.320
Measurement over 3 pins (axial) according to AGMA 2002 (mm)	[dk3A.e/i]	86.954 / 86.888	121.266 / 121.195
Chordal tooth thickness (no backlash) (mm)	[sc]	7.087	7.539
Actual chordal tooth thickness (mm)	[sc.e/i]	7.017 / 6.977	7.469 / 7.429
Reference chordal height from da.m (mm)	[ha]	5.239	5.811
Tooth thickness (Arc) (mm)	[sn]	7.098	7.545
(mm)	[sn.e/i]	7.028 / 6.988	7.475 / 7.435
Backlash free center distance (mm)	[aControl.e/i]	91.360 / 91.279	
Backlash free center distance, allowances (mm)	[jta]	-0.140 / -0.221	
dNf.i with aControl (mm)	[dNf0.i]	69.232	104.919
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	0.475	0.578
Tip clearance (mm)	[c0.i(aControl)]	0.625	0.625
Centre distance allowances (mm)	[Aa.e/i]	0.018 / -0.018	
Circumferential backlash from Aa (mm)	[jtw_Aa.e/i]	0.019 / -0.019	
Radial clearance (mm)	[jrw]	0.238 / 0.123	
Circumferential backlash (transverse section) (mm)	[jtw]	0.263 / 0.136	
Normal backlash (mm)	[jnw]	0.239 / 0.123	
Torsional angle at entry with fixed output:			
Entire torsional angle (°)	[j.tSys]	0.4123 / 0.2127	

7. GEAR ACCURACY

		----- GEAR 1 -----	GEAR 2 --
According to ISO 1328-1:1995, ISO 1328-2:1997			
Accuracy grade	[Q]	6	6
Single pitch deviation (μm)	[fptT]	8.50	8.50
Base circle pitch deviation (μm)	[fpbT]	8.00	8.00
Sector pitch deviation over k/8 pitches (μm)	[Fpk/8T]	12.00	15.00
Profile form deviation (μm)	[ffaT]	8.50	8.50
Profile slope deviation (μm)	[fHaT]	7.00	7.00
Total profile deviation (μm)	[FaT]	11.00	11.00
Helix form deviation (μm)	[ffbT]	8.50	8.50
Helix slope deviation (μm)	[fHbT]	8.50	8.50
Total helix deviation (μm)	[FbT]	12.00	12.00
Total cumulative pitch deviation (μm)	[FpT]	27.00	27.00
Runout (μm)	[FrT]	21.00	21.00
Single flank composite, total (μm)	[FisT]	42.00	42.00
Single flank composite, tooth-to-tooth (μm)	[fisT]	15.00	15.00
Radial composite, total (μm)	[FidT]	36.00	36.00
Radial composite, tooth-to-tooth (μm)	[fidT]	14.00	14.00

Axis alignment tolerances (recommendation acc. to ISO TR 10064-3:1996, Quality)

6)

Maximum value for deviation error of axis (μm)	[fSigbet]	12.01 (Fb= 12.00)
Maximum value for inclination error of axes (μm)	[fSigdel]	24.02

8. ADDITIONAL DATA

Maximal possible centre distance (eps_a=1.0)	[aMAX]	91.500	
Mass (kg)	[m]	1.170	2.582
Total mass (kg)	[m]	3.752	
Moment of inertia (system with reference to the drive): calculation without consideration of the exact tooth shape			
single gears ((da+df)/2...di) (kg*m²)	[TraeghMom]	0.00078	0.003797
System ((da+df)/2...di) (kg*m²)	[TraeghMom]	0.002468	
Torsional stiffness at entry with driven force fixed:			
Torsional stiffness (MNm/rad)	[cr]	0.467	
Torsion when subjected to nominal torque (°)	[delcr]	0.059	
Mean coeff. of friction (acc. Niemann)	[mum]	0.062	
Wear sliding coef. by Niemann	[zetw]	0.353	
Gear power loss (kW)	[PVZ]	0.460	
(Meshing efficiency (%))	[etaz]	99.234	
Sound pressure level (according to Masuda)	[dB(A)]	76.9	

9. MODIFICATIONS AND TOOTH FORM DEFINITION

Data for the tooth form calculation :

Data not available.

10. SERVICE LIFE, DAMAGE

Required safety for tooth root	[SFmin]	1.40
Required safety for tooth flank	[SHmin]	1.00

Service life (calculated with required safeties):

System service life (h)	[Hatt]	14376	
Tooth root service life (h)	[HFatt]	2.443e+004	1.351e+005
Tooth flank service life (h)	[HHatt]	1.438e+004	2.156e+004

Damage calculated on the basis of the required service life [H] (20000.0 h)

F1%	F2%	H1%	H2%
81.87	14.81	139.12	92.75

Damage calculated on basis of system service life [Hatt] (14375.8 h)

F1%	F2%	H1%	H2%
58.85	10.64	100.00	66.67

Calculation of the factors required to define reliability R(t) according to B. Bertsche with Weibull distribution:

$R(t) = 100 * \exp(-((t*fac - t_0)/(T - t_0))^b) \%$; t (h)

Gear		fac	b	t0	T	R(H)%
1	Tooth root	72000	1.7	1.698e+009	2.609e+009	100.00
1	Tooth flank	72000	1.3	9.33e+008	4.445e+009	92.24
2	Tooth root	48000	1.7	6.26e+009	9.62e+009	100.00
2	Tooth flank	48000	1.3	9.33e+008	4.445e+009	99.82

Reliability of the configuration for required service life (%) 92.08 (Bertsche)

REMARKS:

- Specifications with [e/i] imply: Maximum [e] and Minimal value [i] with consideration of all tolerances
 - Specifications with [m] imply: Mean value within tolerance
 - For the backlash tolerance, the center distance tolerances and the tooth thickness deviation are taken into account. Shown is the maximal and the minimal backlash corresponding the largest resp. the smallest allowances
- The calculation is done for the operating pitch circle.
- Calculation of Zbet according Corrigendum 1 ISO 6336-2:2008 with $Z_{bet} = 1/(\cos(\beta))^0.5$
 - Details of calculation method:
 - cg according to method B
 - KV according to method B
 - KHb, KFb according method C
 - fma following equation (64), fsh following (57/58), Fbx following (52/53/57)
 - KHa, KFa according to method B
 - The logarithmically interpolated value taken from the values for the fatigue strength and the static strength, based on the number of load cycles, is used for coefficients ZL, ZV, ZR, ZW, ZX, YdreIT, YRreIT and YX..

End of Report

lines: 529

B.2 CMS55-R

Name : Unnamed

Changed by: up201306219

on: 22.04.2018

at: 16:46:08

CALCULATION OF A HELICAL GEAR PAIR

Drawing or article number:

Gear 1: 0.000.0

Gear 2: 0.000.0

Calculation method

ISO 6336:2006 Method B

----- GEAR 1 ----- GEAR 2 --

Power (kW)	[P]	60.040	
Speed (1/min)	[n]	1200.0	800.0
Torque (Nm)	[T]	477.8	716.7
Application factor	[KA]		1.25
Required service life (h)	[H]	20000.00	
Gear driving (+) / driven (-)		+	-
Working flank gear 1: Right flank			
Sense of rotation gear 1 clockwise			

1. TOOTH GEOMETRY AND MATERIAL

(geometry calculation according to ISO 21771:2007, DIN ISO 21771)

----- GEAR 1 ----- GEAR 2 --

Center distance (mm)	[a]	91.500	
Centre distance tolerance	ISO 286:2010 Measure js7		
Normal module (mm)	[mn]	3.0000	
Pressure angle at normal section (°)	[alfn]	20.0000	
Helix angle at reference circle (°)	[beta]	28.1147	
Number of teeth	[z]	22	33
Facewidth (mm)	[b]	40.00	40.00
Hand of gear		right	left
Accuracy grade	[Q-ISO 1328:1995]	6	6
Inner diameter (mm)	[di]	0.00	0.00
Inner diameter of gear rim (mm)	[dbi]	0.00	0.00

Material

Gear 1:	16 MnCr 5 (1), Case-carburized steel, case-hardened
	ISO 6336-5 Figure 9/10 (MQ), Core hardness >=25HRC Jominy J=12mm<HRC28
Gear 2:	16 MnCr 5 (1), Case-carburized steel, case-hardened
	ISO 6336-5 Figure 9/10 (MQ), Core hardness >=25HRC Jominy J=12mm<HRC28

----- GEAR 1 ----- GEAR 2 --

Surface hardness		HRC 59	HRC 59
Material quality according to ISO 6336:2006 Normal (Life factors ZNT and YNT >=0.85)			
Fatigue strength. tooth root stress (N/mm²)	[σFlim]	430.00	430.00
Fatigue strength for Hertzian pressure (N/mm²)	[σHlim]	1500.00	1500.00
Tensile strength (N/mm²)	[σB]	1000.00	1000.00
Yield point (N/mm²)	[σS]	695.00	695.00
Young's modulus (N/mm²)	[E]	206000	206000
Poisson's ratio	[ν]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.35	0.35

Roughness average value DS, root (μm)	[RAF]	3.00	3.00
Mean roughness height, Rz, flank (μm)	[RZH]	2.50	2.50
Mean roughness height, Rz, root (μm)	[RZF]	20.00	20.00

Gear reference profile 1 :

Reference profile 1.25 / 0.25 / 1.0 ISO 53:1998 Profil C

Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[rhofP*]	0.250 (rhofPmax*=0.472)	
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[rhoaP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[alfprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[alfKP]	0.000	

not topping

Gear reference profile 2 :

Reference profile 1.25 / 0.25 / 1.0 ISO 53:1998 Profil C

Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[rhofP*]	0.250 (rhofPmax*=0.472)	
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[rhoaP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[alfprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[alfKP]	0.000	

not topping

Summary of reference profile gears:

Dedendum reference profile	[hfP*]	1.250	1.250
Tooth root radius Refer. profile	[rofpP*]	0.250	0.250
Addendum Reference profile	[haP*]	1.000	1.000
Protuberance height coefficient	[hprP*]	0.000	0.000
Protuberance angle (°)	[alfprP]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[alfKP]	0.000	0.000

Type of profile modification: none (only running-in)

Tip relief (μm)	[Ca]	2.0	2.0
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Lubrication type

Oil bath lubrication

Type of oil

Oil: ISO-VG 220

Lubricant base

Mineral-oil base

Kinem. viscosity oil at 40 °C (mm²/s)	[nu40]	220.00	
Kinem. viscosity oil at 100 °C (mm²/s)	[nu100]	17.50	
Specific density at 15 °C (kg/dm³)	[roOil]	0.895	
Oil temperature (°C)	[TS]	70.000	

----- GEAR 1 ----- GEAR 2 --

Overall transmission ratio	[itot]	-1.500	
Gear ratio	[u]	1.500	
Transverse module (mm)	[mt]	3.401	
Pressure angle at pitch circle (°)	[alf]	22.424	
Working transverse pressure angle (°)	[alfwt]	19.098	
	[alfwt.e/i]	19.129 /	19.066
Working pressure angle at normal section (°)	[alfwn]	17.059	

Helix angle at operating pitch circle (°)	[betaw]	27.594		
Base helix angle (°)	[betab]	26.284		
Reference centre distance (mm)	[ad]	93.537		
Sum of profile shift coefficients	[Summexi]	-0.6324		
Profile shift coefficient	[x]	-0.1998		-0.4326
Tooth thickness (Arc) (module) (module)	[sn*]	1.4254		1.2559
Tip alteration (mm)	[k*mn]	-0.140		-0.140
Reference diameter (mm)	[d]	74.829		112.244
Base diameter (mm)	[db]	69.171		103.757
Tip diameter (mm)	[da]	79.351		115.369
(mm)	[da.e/i]	79.351 /	79.341	115.369 / 115.359
Tip diameter allowances (mm)	[Ada.e/i]	0.000 /	-0.010	0.000 / -0.010
Tip form diameter (mm)	[dFa]	79.351		115.369
(mm)	[dFa.e/i]	79.351 /	79.341	115.369 / 115.359
Active tip diameter (mm)	[dNa]	79.351		115.369
Active tip diameter (mm)	[dNa.e/i]	79.351 /	79.341	115.369 / 115.359
Operating pitch diameter (mm)	[dw]	73.200		109.800
(mm)	[dw.e/i]	73.214 /	73.186	109.821 / 109.779
Root diameter (mm)	[df]	66.131		102.149
Generating Profile shift coefficient	[xE.e/i]	-0.2319/	-0.2502	-0.4646/ -0.4830
Manufactured root diameter with xE (mm)	[df.e/i]	65.938 /	65.828	101.956 / 101.846
Theoretical tip clearance (mm)	[c]	0.750		0.750
Effective tip clearance (mm)	[c.e/i]	0.924 /	0.829	0.924 / 0.829
Active root diameter (mm)	[dNf]	69.811		105.859
(mm)	[dNf.e/i]	69.829 /	69.797	105.884 / 105.838
Root form diameter (mm)	[dFf]	69.671		105.471
(mm)	[dFf.e/i]	69.612 /	69.580	105.382 / 105.332
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.124 /	0.092	0.276 / 0.228
Addendum (mm)	[ha=mn*(haP*+x+k)]	2.261		1.562
(mm)	[ha.e/i]	2.261 /	2.256	1.562 / 1.557
Dedendum (mm)	[hf=mn*(hfP*-x)]	4.349		5.048
(mm)	[hf.e/i]	4.446 /	4.501	5.144 / 5.199
Roll angle at dFa (°)	[xsi_dFa.e/i]	32.207 /	32.190	27.855 / 27.842
Roll angle to dNa (°)	[xsi_dNa.e/i]	32.207 /	32.190	27.855 / 27.842
Roll angle to dNf (°)	[xsi_dNf.e/i]	7.920 /	7.724	11.662 / 11.532
Roll angle at dFf (°)	[xsi_dFf.e/i]	6.480 /	6.242	10.180 / 10.020
Tooth height (mm)	[h]	6.610		6.610
Virtual gear no. of teeth	[zn]	31.027		46.541
Normal tooth thickness at tip circle (mm)	[san]	2.489		2.599
(mm)	[san.e/i]	2.421 /	2.374	2.531 / 2.487
Normal-tooth thickness on tip form circle (mm)	[sFan]	2.489		2.599
(mm)	[sFan.e/i]	2.421 /	2.374	2.531 / 2.487
Normal space width at root circle (mm)	[efn]	0.000		0.000
(mm)	[efn.e/i]	0.000 /	0.000	0.000 / 0.000
Max. sliding velocity at tip (m/s)	[vga]	1.564		1.520
Specific sliding at the tip	[zetaaa]	0.640		0.720
Specific sliding at the root	[zetaf]	-2.566		-1.779
Mean specific sliding	[zetam]		0.679	
Sliding factor on tip	[Kga]	0.340		0.331
Sliding factor on root	[Kgf]	-0.331		-0.340
Pitch on reference circle (mm)	[pt]		10.686	
Base pitch (mm)	[pbt]		9.878	
Transverse pitch on contact-path (mm)	[pet]		9.878	
Lead height (mm)	[pz]	440.001		660.001
Axial pitch (mm)	[px]		20.000	
Length of path of contact (mm)	[ga, e/i]	14.726 (	14.779 /	14.651)

Length T1-A, T2-A (mm)	[T1A, T2A]	4.716(4.662/ 4.781)	25.221(25.221/ 25.210)
Length T1-B (mm)	[T1B, T2B]	9.564(9.564/ 9.553)	20.373(20.320/ 20.437)
Length T1-C (mm)	[T1C, T2C]	11.975(11.953/ 11.996)	17.962(17.930/ 17.994)
Length T1-D (mm)	[T1D, T2D]	14.593(14.540/ 14.658)	15.344(15.344/ 15.332)
Length T1-E (mm)	[T1E, T2E]	19.441(19.441/ 19.431)	10.496(10.442/ 10.559)
Length T1-T2 (mm)	[T1T2]	29.937 (29.883 / 29.990)	
Diameter of single contact point B (mm)	[d-B]	71.767(71.767/ 71.762)	111.471(111.432/ 111.517)
Diameter of single contact point D (mm)	[d-D]	75.077(75.035/ 75.127)	108.200(108.200/ 108.193)
Addendum contact ratio	[eps]	0.756(0.758/ 0.753)	0.735(0.738/ 0.730)
Minimal length of contact line (mm)	[Lmin]	66.508	
Transverse contact ratio	[eps_a]	1.491	
Transverse contact ratio with allowances	[eps_a.e/m/i]	1.496 / 1.490 / 1.483	
Overlap ratio	[eps_b]	2.000	
Total contact ratio	[eps_g]	3.491	
Total contact ratio with allowances	[eps_g.e/m/i]	3.496 / 3.490 / 3.483	

2. FACTORS OF GENERAL INFLUENCE

		----- GEAR 1 -----	GEAR 2 --
Nominal circum. force at pitch circle (N)	[Ft]		12769.8
Axial force (N)	[Fa]		6822.7
Radial force (N)	[Fr]		5269.6
Normal force (N)	[Fnorm]		15407.4
Nominal circumferential force per mm (N/mm)	[w]		319.25
Only as information: Forces at operating pitch circle:			
Nominal circumferential force (N)	[Ftw]		13054.1
Axial force (N)	[Faw]		6822.7
Radial force (N)	[Frw]		4519.8
Circumferential speed reference circle (m/s)	[v]		4.70
Circumferential speed operating pitch circle (m/s)	[v(dw)]		4.60
Running-in value (μm)	[yp]		0.6
Running-in value (μm)	[yf]		0.6
Correction coefficient	[CM]		0.800
Gear body coefficient	[CR]		1.000
Basic rack factor	[CBS]		0.975
Material coefficient	[E/Est]		1.000
Singular tooth stiffness (N/mm/μm)	[c']		10.845
Meshing stiffness (N/mm/μm)	[cgalf]		14.837
Meshing stiffness (N/mm/μm)	[cgbet]		12.611
Reduced mass (kg/mm)	[mRed]		0.01241
Resonance speed (min-1)	[nE1]		15011
Resonance ratio (-)	[N]		0.080
Subcritical range			
Running-in value (μm)	[ya]		0.6
Bearing distance l of pinion shaft (mm)	[l]		80.000
Distance s of pinion shaft (mm)	[s]		8.000
Outside diameter of pinion shaft (mm)	[dsh]		40.000
Load in accordance with Figure 13, ISO 6336-1:2006 0:a), 1:b), 2:c), 3:d), 4:e)	[-]	4	
Coefficient K' according to Figure 13, ISO 6336-1:2006	[K']	-1.00	
Without support effect			
Tooth trace deviation (active) (μm)	[Fby]		3.61
from deformation of shaft (μm)	[fsh*B1]		2.65
(fsh (μm) = 2.65, B1= 1.00, fHb5 (μm) = 6.00)			

Tooth without tooth trace modification			
Position of Contact pattern: favorable			
from production tolerances (μm)	[fma*B2]	12.02	
(B2= 1.00)			
Tooth trace deviation, theoretical (μm)	[Fbx]	4.25	
Running-in value (μm)	[yb]	0.64	
Dynamic factor	[KV]	1.012	
Face load factor - flank	[KHb]	1.056	
- Tooth root	[KFb]	1.047	
- Scuffing	[KBb]	1.056	
Transverse load factor - flank	[KHa]	1.031	
- Tooth root	[KF _a]	1.031	
- Scuffing	[KB _a]	1.031	
Helical load factor scuffing	[Kbg]	1.300	
Number of load cycles (in mio.)	[NL]	1440.000	960.000

3. TOOTH ROOT STRENGTH

Calculation of Tooth form coefficients according method: B

		----- GEAR 1 -----	GEAR 2 --
Calculated with profile shift	[x]	-0.1998	-0.4326
Tooth form factor	[YF]	1.40	1.44
Stress correction factor	[YS]	1.96	1.88
Working angle (°)	[alfFen]	15.05	15.60
Bending moment arm (mm)	[hF]	2.67	2.84
Tooth thickness at root (mm)	[sFn]	5.94	6.04
Tooth root radius (mm)	[roF]	1.61	1.73
(hF* = 0.889/ 0.948 sFn* = 1.979/ 2.014 roF* = 0.537/ 0.575)			
(den (mm) =			
91.894/ 137.490 dsFn(mm) = 67.254/ 103.331 alfsFn(°) = 30.00/ 30.00 qs = 1.841/ 1.751)			
Helix angle factor	[Ybet]	0.766	
Deep tooth factor	[YDT]	1.000	
Gear rim factor	[YB]	1.00	1.00
Effective facewidth (mm)	[beff]	40.00	40.00
Nominal stress at tooth root (N/mm²)	[sigF0]	222.98	220.76
Tooth root stress (N/mm²)	[sigF]	304.61	301.57
Permissible bending stress at root of Test-gear			
Notch sensitivity factor	[YdreIT]	0.993	0.992
Surface factor	[YRrelT]	0.957	0.957
size factor (Tooth root)	[YX]	1.000	1.000
Finite life factor	[YNT]	0.884	0.891
	[YdreIT*YRrelT*YX*YNT]	0.840	0.846
Alternating bending factor (mean stress influence coefficient)	[YM]	1.000	1.000
Stress correction factor	[Yst]	2.00	
Yst*sigFlim (N/mm²)	[sigFE]	860.00	860.00
Permissible tooth root stress (N/mm²)	[sigFP=sigFG/SFmin]	515.90	519.60
Limit strength tooth root (N/mm²)	[sigFG]	722.27	727.44
Required safety	[SFmin]	1.40	1.40
Safety for tooth root stress	[SF=sigFG/sigF]	2.37	2.41

Transmittable power (kW)	[kWRating]	101.69	103.45
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4. SAFETY AGAINST PITTING (TOOTH FLANK)

		----- GEAR 1 -----	GEAR 2 --
Zone factor	[ZH]		2.462
Elasticity factor ($\sqrt{N/mm^2}$)	[ZE]		189.812
Contact ratio factor	[Zeps]		0.819
Helix angle factor	[Zbet]		1.065
Effective facewidth (mm)	[beff]		40.00
Nominal contact stress (N/mm^2)	[sigH0]		1086.69
Contact stress at operating pitch circle (N/mm^2)	[sigHw]		1275.85
Single tooth contact factor	[ZB,ZD]	1.00	1.00
Contact stress (N/mm^2)	[sigHB, sigHD]	1275.85	1275.85
Lubrication coefficient at NL	[ZL]	1.020	1.020
Speed coefficient at NL	[ZV]	0.981	0.981
Roughness coefficient at NL	[ZR]	1.006	1.006
Material pairing coefficient at NL	[ZW]	1.000	1.000
Finite life factor	[ZNT]	0.902	0.913
	[ZL * ZV * ZR * ZNT]	0.908	0.919
Limited pitting is permitted:	No		
Size factor (flank)	[ZX]	1.000	1.000
Permissible contact stress (N/mm^2)	[sigHP=sigHG/SHmin]	1361.43	1378.47
Pitting stress limit (N/mm^2)	[sigHG]	1361.43	1378.47
Required safety	[SHmin]	1.00	1.00
Safety factor for contact stress at operating pitch circle			
	[SHw]	1.07	1.08
Safety for stress at single tooth contact	[SHBD=sigHG/sigHBD]	1.07	1.08
(Safety regarding transmittable torque)	[(SHBD)^2]	1.14	1.17
Transmittable power (kW)	[kWRating]	68.36	70.09

4b. MICROPITTING ACCORDING TO ISO/TR 15144-1:2014

Calculation did not run. (Lubricant: Load stage micropitting test is unknown.)

5. SCUFFING LOAD CAPACITY

Calculation method according to ISO TR 13989:2000

Lubrication coefficient (for lubrication type)	[XS]	1.000	
Scuffing test and load stage	[FZGtest]	FZG - Test A / 8.3 / 90 (ISO 14635 - 1)	12
Multiple meshing factor	[Xmp]	1.000	
Relative structure coefficient (Scuffing)	[XWrelT]	1.000	
Thermal contact factor ($N/mm/s^0.5/K$)	[BM]	13.780	13.780
Relevant tip relief (μm)	[Ca]	2.00	2.00
Optimal tip relief (μm)	[Ceff]	26.90	
Ca taken as optimal in the calculation (0=no, 1=yes)		0	0
Effective facewidth (mm)	[beff]	40.000	
Applicable circumferential force/facewidth (N/mm)	[wBt]	440.062	
Kbg =	1.300, wBt*Kbg =	572.078	

Angle factor ($\epsilon_1:0.756, \epsilon_2:0.735$)	[Xalfbet]	0.942
Flash temperature-criteria		
Lubricant factor	[XL]	0.830
Tooth mass temperature (°C) ($theMi = theoil + XS \cdot 0.47 \cdot Xmp \cdot theflm$)	[theMi]	83.53
Average flash temperature (°C)	[theflm]	28.78
Scuffing temperature (°C)	[theS]	348.80
Coordinate gamma (point of highest temp.) [Gamma.A]=-0.606 [Gamma.E]=0.624	[Gamma]	-0.606
Highest contact temp. (°C)	[theB]	158.38
Flash factor ($^{\circ}K \cdot N^{\wedge}-.75 \cdot s^{\wedge}.5 \cdot m^{\wedge}-.5 \cdot mm$)	[XM]	50.058
Approach factor	[XJ]	1.060
Load sharing factor	[XGam]	0.872
Dynamic viscosity (mPa*s)	[etaM]	41.90 (70.0 °C)
Coefficient of friction	[mym]	0.061
Required safety	[SBmin]	2.000
Safety factor for scuffing (flash temperature)	[SB]	3.155
Integral temperature-criteria		
Lubricant factor	[XL]	1.000
Tooth mass temperature (°C) ($theMC = theoil + XS \cdot 0.70 \cdot theflaint$)	[theMC]	88.57
Mean flash temperature (°C)	[theflaint]	26.53
Integral scuffing temperature (°C)	[theSint]	360.78
Flash factor ($^{\circ}K \cdot N^{\wedge}-.75 \cdot s^{\wedge}.5 \cdot m^{\wedge}-.5 \cdot mm$)	[XM]	50.058
Running-in factor (well run in)	[XE]	1.000
Contact ratio factor	[Xeps]	0.283
Dynamic viscosity (mPa*s)	[etaOil]	41.90 (70.0 °C)
Mean coefficient of friction	[mym]	0.071
Geometry factor	[XBE]	0.376
Meshing factor	[XQ]	1.000
Tip relief factor	[XCa]	1.096
Integral tooth flank temperature (°C)	[theint]	128.37
Required safety	[SSmin]	1.800
Safety factor for scuffing (intg.-temp.)	[SSint]	2.810
Safety referring to transmittable torque	[SSL]	4.982

6. MEASUREMENTS FOR TOOTH THICKNESS

		----- Gear 1 ----- Gear 2 --	
		DIN 3967 cd25	DIN 3967 cd25
Tooth thickness deviation		-0.070 / -0.110	-0.070 / -0.110
Tooth thickness allowance (normal section) (mm)	[As.e/i]		
Number of teeth spanned	[k]	4.000	5.000
Base tangent length (no backlash) (mm)	[Wk]	31.908	40.946
Actual base tangent length ('span') (mm)	[Wk.e/i]	31.842 / 31.804	40.881 / 40.843
(mm)	[ΔWk.e/i]	-0.066 / -0.103	-0.066 / -0.103
Diameter of measuring circle (mm)	[dMWk.m]	74.825	110.035
Theoretical diameter of ball/pin (mm)	[DM]	4.997	4.960
Effective diameter of ball/pin (mm)	[DMeff]	5.000	5.000
Radial single-ball measurement backlash free (mm)	[MrK]	40.147	58.148
Radial single-ball measurement (mm)	[MrK.e/i]	40.054 / 40.000	58.046 / 57.987
Diameter of measuring circle (mm)	[dMMr.m]	73.430	109.517

Diametral measurement over two balls without clearance (mm)	[MdK]	80.294	116.170
Diametral two ball measure (mm)	[MdK.e/i]	80.107 / 79.999	115.966 / 115.849
Diametral measurement over pins without clearance (mm)	[MdR]	80.294	116.296
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	80.107 / 79.999	116.092 / 115.975
Measurement over 2 pins (free) according to AGMA 2002 (mm)	[dk2f.e/i]	0.000 / 0.000	115.918 / 115.800
Measurement over 2 pins (transverse) according to AGMA 2002 (mm)	[dk2t.e/i]	0.000 / 0.000	116.217 / 116.099
Measurement over 3 pins (axial) according to AGMA 2002 (mm)	[dk3A.e/i]	80.107 / 79.999	116.092 / 115.975
Chordal tooth thickness (no backlash) (mm)	[sc]	4.275	3.767
Actual chordal tooth thickness (mm)	[sc.e/i]	4.205 / 4.165	3.697 / 3.657
Reference chordal height from da.m (mm)	[ha]	2.306	1.584
Tooth thickness (Arc) (mm)	[sn]	4.276	3.768
(mm)	[sn.e/i]	4.206 / 4.166	3.698 / 3.658
Backlash free center distance (mm)	[aControl.e/i]	91.273 / 91.142	
Backlash free center distance, allowances (mm)	[jta]	-0.227 / -0.358	
dNf.i with aControl (mm)	[dNf0.i]	69.545	105.440
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	-0.033	0.029
Tip clearance (mm)	[c0.i(aControl)]	0.488	0.488
Centre distance allowances (mm)	[Aa.e/i]	0.018 / -0.018	
Circumferential backlash from Aa (mm)	[jtw_Aa.e/i]	0.012 / -0.012	
Radial clearance (mm)	[jrw]	0.376 / 0.209	
Circumferential backlash (transverse section) (mm)	[jtw]	0.256 / 0.143	
Normal backlash (mm)	[jnw]	0.212 / 0.119	
Torsional angle at entry with fixed output:			
Entire torsional angle (°)	[j.tSys]	0.4009 / 0.2241	

7. GEAR ACCURACY

		----- GEAR 1 -----	GEAR 2 --
According to ISO 1328-1:1995, ISO 1328-2:1997			
Accuracy grade	[Q]	6	6
Single pitch deviation (µm)	[fptT]	8.50	8.50
Base circle pitch deviation (µm)	[fpbT]	7.90	7.90
Sector pitch deviation over k/8 pitches (µm)	[Fpk/8T]	12.00	15.00
Profile form deviation (µm)	[ffaT]	8.50	8.50
Profile slope deviation (µm)	[fHaT]	7.00	7.00
Total profile deviation (µm)	[FaT]	11.00	11.00
Helix form deviation (µm)	[ffbT]	8.50	8.50
Helix slope deviation (µm)	[fHbT]	8.50	8.50
Total helix deviation (µm)	[FbT]	12.00	12.00
Total cumulative pitch deviation (µm)	[FpT]	27.00	27.00
Runout (µm)	[FrT]	21.00	21.00
Single flank composite, total (µm)	[FisT]	38.00	38.00
Single flank composite, tooth-to-tooth (µm)	[fisT]	11.00	11.00
Radial composite, total (µm)	[FidT]	36.00	36.00
Radial composite, tooth-to-tooth (µm)	[fidT]	14.00	14.00

Axis alignment tolerances (recommendation acc. to ISO TR 10064-3:1996, Quality)

6)

Maximum value for deviation error of axis (µm)	[fSigbet]	12.00 (Fb= 12.00)
Maximum value for inclination error of axes (µm)	[fSigdel]	24.00

8. ADDITIONAL DATA

Maximal possible centre distance (eps_a=1.0)	[aMAX]	93.199	
Mass (kg)	[m]	1.302	2.910
Total mass (kg)	[m]	4.211	
Moment of inertia (system with reference to the drive): calculation without consideration of the exact tooth shape			
single gears ((da+df)/2...di) (kg*m ²)	[TraeghMom]	0.0008609	0.004302
System ((da+df)/2...di) (kg*m ²)	[TraeghMom]	0.002773	
Torsional stiffness at entry with driven force fixed:			
Torsional stiffness (MNm/rad)	[cr]	0.617	
Torsion when subjected to nominal torque (°)	[delcr]	0.044	
Mean coeff. of friction (acc. Niemann)	[mum]	0.066	
Wear sliding coef. by Niemann	[zetw]	1.013	
Gear power loss (kW)	[PVZ]	0.651	
(Meshing efficiency (%))	[etaz]	98.916)	
Sound pressure level (according to Masuda)	[dB(A)]	72.6	

9. MODIFICATIONS AND TOOTH FORM DEFINITION

Data for the tooth form calculation :

Data not available.

10. SERVICE LIFE, DAMAGE

Required safety for tooth root	[SFmin]	1.40
Required safety for tooth flank	[SHmin]	1.00

Service life (calculated with required safeties):

System service life (h)	[Hatt]	> 1000000
-------------------------	--------	-----------

Tooth root service life (h)	[HFatt]	1e+006	1e+006
Tooth flank service life (h)	[HHatt]	1e+006	1e+006

Note: The entry 1e+006 h means that the Service life > 1,000,000 h.

Damage calculated on the basis of the required service life [H] (20000.0 h)

F1%	F2%	H1%	H2%
0.00	0.00	0.00	0.00

Calculation of the factors required to define reliability R(t) according to B. Bertsche with Weibull distribution:

$$R(t) = 100 * \exp(-((t^{*fac} - t_0)/(T - t_0))^b) \% ; t (h)$$

Gear		fac	b	t0	T	R(H)%
1	Tooth root	72000	1.7	9.654e+029	1.484e+030	100.00
1	Tooth flank	72000	1.3	9.014e+029	4.295e+030	100.00
2	Tooth root	48000	1.7	9.654e+029	1.484e+030	100.00
2	Tooth flank	48000	1.3	9.014e+029	4.295e+030	100.00

Reliability of the configuration for required service life (%) 100.00 (Bertsche)

REMARKS:

- Specifications with [e/i] imply: Maximum [e] and Minimal value [i] with

consideration of all tolerances

Specifications with [..m] imply: Mean value within tolerance

- For the backlash tolerance, the center distance tolerances and the tooth thickness deviation are taken into account. Shown is the maximal and the minimal backlash corresponding the largest resp. the smallest allowances

The calculation is done for the operating pitch circle.

- Calculation of Zbet according Corrigendum 1 ISO 6336-2:2008 with $Z_{bet} = 1/(\cos(\beta))^0.5$
- Details of calculation method:
 - cg according to method B
 - KV according to method B
 - KHb, KFb according method C
 - fma following equation (64), fsh following (57/58), Fbx following (52/53/57)
 - KHa, KFa according to method B

- The logarithmically interpolated value taken from the values for the fatigue strength and the static strength, based on the number of load cycles, is used for coefficients ZL, ZV, ZR, ZW, ZX, YdreIT, YRreIT and YX..

End of Report

lines: 527

B.3 CMS92-LE

Name : Unnamed

Changed by: up201306219

on: 21.04.2018

at: 12:52:32

CALCULATION OF A HELICAL GEAR PAIR

Drawing or article number:

Gear 1: 0.000.0

Gear 2: 0.000.0

Calculation method

ISO 6336:2006 Method B

----- GEAR 1 ----- GEAR 2 --

Power (kW)	[P]	60.040	
Speed (1/min)	[n]	1200.0	807.3
Torque (Nm)	[T]	477.8	710.2
Application factor	[KA]		1.25
Required service life (h)	[H]	20000.00	
Gear driving (+) / driven (-)		+	-
Working flank gear 1: Right flank			
Sense of rotation gear 1 clockwise			

1. TOOTH GEOMETRY AND MATERIAL

(geometry calculation according to ISO 21771:2007, DIN ISO 21771)

----- GEAR 1 ----- GEAR 2 --

Center distance (mm)	[a]	91.500	
Centre distance tolerance	ISO 286:2010 Measure js7		
Normal module (mm)	[mn]	1.7500	
Pressure angle at normal section (°)	[alfn]	20.0000	
Helix angle at reference circle (°)	[beta]	21.6909	
Number of teeth	[z]	37	55
Facewidth (mm)	[b]	29.75	29.75
Hand of gear		right	left
Accuracy grade	[Q-ISO 1328:1995]	6	6
Inner diameter (mm)	[di]	0.00	0.00
Inner diameter of gear rim (mm)	[dbi]	0.00	0.00

Material

Gear 1:

16 MnCr 5 (1), Case-carburized steel, case-hardened
ISO 6336-5 Figure 9/10 (MQ), Core hardness >=25HRC Jominy J=12mm<HRC28

Gear 2:

16 MnCr 5 (1), Case-carburized steel, case-hardened
ISO 6336-5 Figure 9/10 (MQ), Core hardness >=25HRC Jominy J=12mm<HRC28

----- GEAR 1 ----- GEAR 2 --

Surface hardness		HRC 59	HRC 59
Material quality according to ISO 6336:2006 Normal (Life factors ZNT and YNT >=0.85)			
Fatigue strength. tooth root stress (N/mm²)	[σFlim]	430.00	430.00
Fatigue strength for Hertzian pressure (N/mm²)	[σHlim]	1500.00	1500.00
Tensile strength (N/mm²)	[σB]	1000.00	1000.00
Yield point (N/mm²)	[σS]	695.00	695.00
Young's modulus (N/mm²)	[E]	206000	206000
Poisson's ratio	[ν]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.35	0.35

Roughness average value DS, root (µm)	[RAF]	3.00	3.00
Mean roughness height, Rz, flank (µm)	[RZH]	2.50	2.50
Mean roughness height, Rz, root (µm)	[RZF]	20.00	20.00

Gear reference profile 1 :

Reference profile 1.25 / 0.25 / 1.0 ISO 53:1998 Profil C

Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[rhofP*]	0.250 (rhofPmax*=0.472)	
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[rhoaP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[alfprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[alfKP]	0.000	

not topping

Gear reference profile 2 :

Reference profile 1.25 / 0.25 / 1.0 ISO 53:1998 Profil C

Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[rhofP*]	0.250 (rhofPmax*=0.472)	
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[rhoaP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[alfprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[alfKP]	0.000	

not topping

Summary of reference profile gears:

Dedendum reference profile	[hfP*]	1.250	1.250
Tooth root radius Refer. profile	[rofpP*]	0.250	0.250
Addendum Reference profile	[haP*]	1.000	1.000
Protuberance height coefficient	[hprP*]	0.000	0.000
Protuberance angle (°)	[alfprP]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[alfKP]	0.000	0.000

Type of profile modification: none (only running-in)

Tip relief (µm)	[Ca]	2.0	2.0
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Lubrication type Oil bath lubrication

Type of oil Oil: ISO-VG 220

Lubricant base Mineral-oil base

Kinem. viscosity oil at 40 °C (mm²/s)	[nu40]	220.00	
Kinem. viscosity oil at 100 °C (mm²/s)	[nu100]	17.50	
Specific density at 15 °C (kg/dm³)	[roOil]	0.895	
Oil temperature (°C)	[TS]	70.000	

----- GEAR 1 ----- GEAR 2 --

Overall transmission ratio	[itot]	-1.486	
Gear ratio	[u]	1.486	
Transverse module (mm)	[mt]	1.883	
Pressure angle at pitch circle (°)	[alf]	21.391	
Working transverse pressure angle (°)	[alfwt]	28.164	
	[alfwt.e/i]	28.184 / 28.143	
Working pressure angle at normal section (°)	[alfwn]	26.271	

Helix angle at operating pitch circle (°)	[betaw]	22.787		
Base helix angle (°)	[betab]	20.323		
Reference centre distance (mm)	[ad]	86.634		
Sum of profile shift coefficients	[Summexi]	3.2177		
Profile shift coefficient	[x]	1.3702	1.8475	
Tooth thickness (Arc) (module) (module)	[sn*]	2.5682	2.9157	
Tip alteration (mm)	[k*mn]	-0.766	-0.766	
Reference diameter (mm)	[d]	69.684	103.585	
Base diameter (mm)	[db]	64.884	96.449	
Tip diameter (mm)	[da]	76.448	112.019	
(mm)	[da.e/i]	76.448 / 76.438	112.019 / 112.009	
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.010	0.000 / -0.010	
Tip form diameter (mm)	[dFa]	76.448	112.019	
(mm)	[dFa.e/i]	76.448 / 76.438	112.019 / 112.009	
Active tip diameter (mm)	[dNa]	76.448	112.019	
Active tip diameter (mm)	[dNa.e/i]	76.448 / 76.438	112.019 / 112.009	
Operating pitch diameter (mm)	[dw]	73.598	109.402	
(mm)	[dw.e/i]	73.612 / 73.584	109.423 / 109.381	
Root diameter (mm)	[df]	70.105	105.676	
Generating Profile shift coefficient	[xE.e/i]	1.3153 / 1.2839	1.7926 / 1.7612	
Manufactured root diameter with xE (mm)	[df.e/i]	69.913 / 69.803	105.484 / 105.374	
Theoretical tip clearance (mm)	[c]	0.438	0.438	
Effective tip clearance (mm)	[c.e/i]	0.612 / 0.517	0.612 / 0.517	
Active root diameter (mm)	[dNf]	71.235	106.835	
(mm)	[dNf.e/i]	71.274 / 71.205	106.875 / 106.803	
Root form diameter (mm)	[dFf]	70.726	106.470	
(mm)	[dFf.e/i]	70.518 / 70.401	106.248 / 106.121	
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.437 / 0.343	0.377 / 0.278	
Addendum (mm)	[ha=mn*(haP*+x+k)]	3.382	4.217	
(mm)	[ha.e/i]	3.382 / 3.377	4.217 / 4.212	
Dedendum (mm)	[hf=mn*(hfP*-x)]	-0.210	-1.046	
(mm)	[hf.e/i]	-0.114 / -0.059	-0.950 / -0.895	
Roll angle at dFa (°)	[xsi_dFa.e/i]	35.699 / 35.682	33.844 / 33.833	
Roll angle to dNa (°)	[xsi_dNa.e/i]	35.699 / 35.682	33.844 / 33.833	
Roll angle to dNf (°)	[xsi_dNf.e/i]	26.047 / 25.898	27.350 / 27.251	
Roll angle at dFf (°)	[xsi_dFf.e/i]	24.390 / 24.124	26.474 / 26.295	
Tooth height (mm)	[h]	3.172	3.171	
Virtual gear no. of teeth	[zn]	45.282	67.311	
Normal tooth thickness at tip circle (mm)	[san]	1.536	1.465	
(mm)	[san.e/i]	1.466 / 1.417	1.396 / 1.348	
Normal-tooth thickness on tip form circle (mm)	[sFan]	1.536	1.465	
(mm)	[sFan.e/i]	1.466 / 1.417	1.396 / 1.348	
Normal space width at root circle (mm)	[efn]	1.166	1.225	
(mm)	[efn.e/i]	1.161 / 1.158	1.214 / 1.209	
Max. sliding velocity at tip (m/s)	[vga]	0.598	0.561	
Specific sliding at the tip	[zetaa]	0.235	0.233	
Specific sliding at the root	[zetaf]	-0.304	-0.308	
Mean specific sliding	[zetam]	0.234		
Sliding factor on tip	[Kga]	0.129	0.121	
Sliding factor on root	[Kgf]	-0.121	-0.129	
Pitch on reference circle (mm)	[pt]	5.917		
Base pitch (mm)	[pbt]	5.509		
Transverse pitch on contact-path (mm)	[pet]	5.509		
Lead height (mm)	[pz]	550.375	818.125	
Axial pitch (mm)	[px]	14.875		
Length of path of contact (mm)	[ga, e/i]	5.512 (5.549 / 5.456)		

Length T1-A, T2-A (mm)	[T1A, T2A]	14.701(	14.664/	14.748)	28.486(	28.486/	28.476)
Length T1-B (mm)	[T1B, T2B]	14.704(	14.704/	14.748)	28.483(	28.446/	28.476)
Length T1-C (mm)	[T1C, T2C]	17.369(	17.354/	17.384)	25.818(	25.796/	25.841)
Length T1-D (mm)	[T1D, T2D]	20.210(	20.173/	20.204)	22.977(	22.977/	23.020)
Length T1-E (mm)	[T1E, T2E]	20.214(	20.214/	20.204)	22.974(	22.937/	23.020)
Length T1-T2 (mm)	[T1T2]	43.187 (43.150 / 43.224)					
Diameter of single contact point B (mm)	[d-B]	71.238(	71.238/	71.274)	112.016(	111.978/	112.009)
Diameter of single contact point D (mm)	[d-D]	76.445(	76.405/	76.438)	106.837(	106.837/	106.875)
Addendum contact ratio	[eps]	0.516(	0.519/	0.512)	0.484(	0.488/	0.478)
Minimal length of contact line (mm)	[Lmin]	31.743					
Transverse contact ratio	[eps_a]	1.001					
Transverse contact ratio with allowances	[eps_a.e/m/i]	1.007 / 0.999 / 0.990					
Overlap ratio	[eps_b]	2.000					
Total contact ratio	[eps_g]	3.001					
Total contact ratio with allowances	[eps_g.e/m/i]	3.007 / 2.999 / 2.990					

2. FACTORS OF GENERAL INFLUENCE

		----- GEAR 1 -----	GEAR 2 --
Nominal circum. force at pitch circle (N)	[Ft]		13712.7
Axial force (N)	[Fa]		5454.4
Radial force (N)	[Fr]		5371.4
Normal force (N)	[Fnorm]		15704.8
Nominal circumferential force per mm (N/mm)	[w]		460.93
Only as information: Forces at operating pitch circle:			
Nominal circumferential force (N)	[Ftw]		12983.5
Axial force (N)	[Faw]		5454.4
Radial force (N)	[Frw]		6951.1
Circumferential speed reference circle (m/s)	[v]		4.38
Circumferential speed operating pitch circle (m/s)	[v(dw)]		4.62
Running-in value (μm)	[yp]		0.5
Running-in value (μm)	[yf]		0.5
Correction coefficient	[CM]		0.800
Gear body coefficient	[CR]		1.000
Basic rack factor	[CBS]		0.975
Material coefficient	[E/Est]		1.000
Singular tooth stiffness (N/mm/μm)	[c']		15.034
Meshing stiffness (N/mm/μm)	[cgalf]		15.041
Meshing stiffness (N/mm/μm)	[cgbet]		12.785
Reduced mass (kg/mm)	[mRed]		0.01448
Resonance speed (min-1)	[nE1]		8317
Resonance ratio (-)	[N]		0.144
Subcritical range			
Running-in value (μm)	[ya]		0.5
Bearing distance l of pinion shaft (mm)	[l]		59.500
Distance s of pinion shaft (mm)	[s]		5.950
Outside diameter of pinion shaft (mm)	[dsh]		29.750
Load in accordance with Figure 13, ISO 6336-1:2006 0:a), 1:b), 2:c), 3:d), 4:e)	[-]	4	
Coefficient K' according to Figure 13, ISO 6336-1:2006	[K']	-1.00	
Without support effect			
Tooth trace deviation (active) (μm)	[Fby]		3.61
from deformation of shaft (μm)	[fsh*B1]		4.43
(fsh (μm) = 4.43, B1= 1.00, fHb5 (μm) = 6.00)			

Tooth without tooth trace modification			
Position of Contact pattern: favorable			
from production tolerances (μm)	[fma*B2]	12.02	
(B2= 1.00)			
Tooth trace deviation, theoretical (μm)	[Fbx]	4.25	
Running-in value (μm)	[yb]	0.64	
Dynamic factor	[KV]	1.022	
Face load factor - flank	[KHb]	1.039	
- Tooth root	[KFb]	1.035	
- Scuffing	[KBb]	1.039	
Transverse load factor - flank	[KHa]	1.000	
- Tooth root	[KF _a]	1.000	
- Scuffing	[KB _a]	1.000	
Helical load factor scuffing	[Kbg]	1.283	
Number of load cycles (in mio.)	[NL]	1440.000	968.727

3. TOOTH ROOT STRENGTH

Calculation of Tooth form coefficients according method: B

		----- GEAR 1 -----	GEAR 2 --
Calculated with manufacturing profile shift	[xE.e]	1.3153	1.7926
Tooth form factor	[YF]	1.26	1.28
Stress correction factor	[YS]	2.62	2.22
Working angle (°)	[alfFen]	28.02	27.41
Bending moment arm (mm)	[hF]	2.29	2.28
Tooth thickness at root (mm)	[sFn]	4.24	4.20
Tooth root radius (mm)	[roF]	0.49	0.72
(hF* = 1.309/ 1.305 sFn* = 2.422/ 2.400 roF* = 0.280/ 0.414)			
(den (mm) =			
85.303/ 125.550 dsFn(mm) = 70.341/ 105.998 alfsFn(°) = 30.00/ 30.00 qs = 4.322/ 2.902)			
Helix angle factor	[Ybet]	0.819	
Deep tooth factor	[YDT]	1.000	
Gear rim factor	[YB]	1.00	1.00
Effective facewidth (mm)	[beff]	29.75	29.75
Nominal stress at tooth root (N/mm ²)	[sigF0]	710.11	614.70
Tooth root stress (N/mm ²)	[sigF]	938.95	812.80
Permissible bending stress at root of Test-gear			
Notch sensitivity factor	[YdreIT]	1.015	1.004
Surface factor	[YRrelT]	0.957	0.957
size factor (Tooth root)	[YX]	1.000	1.000
Finite life factor	[YNT]	0.884	0.891
	[YdreIT*YRrelT*YX*YNT]	0.858	0.855
Alternating bending factor (mean stress influence coefficient)	[YM]	1.000	1.000
Stress correction factor	[Yst]	2.00	
Yst*sigFlim (N/mm ²)	[sigFE]	860.00	860.00
Permissible tooth root stress (N/mm ²)	[sigFP=sigFG/SFmin]	546.73	544.85
Limit strength tooth root (N/mm ²)	[sigFG]	738.09	735.55
Required safety	[SFmin]	1.35	1.35
Safety for tooth root stress	[SF=sigFG/sigF]	0.79	0.90

Transmittable power (kW)	[kWRating]	34.96	40.25
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4. SAFETY AGAINST PITTING (TOOTH FLANK)

		----- GEAR 1 -----	GEAR 2 --
Zone factor	[ZH]		2.010
Elasticity factor ($\sqrt{N/mm^2}$)	[ZE]		189.812
Contact ratio factor	[Zeps]		1.000
Helix angle factor	[Zbet]		1.037
Effective facewidth (mm)	[beff]		29.75
Nominal contact stress (N/mm ²)	[sigH0]		1316.24
Contact stress at operating pitch circle (N/mm ²)	[sigHw]		1516.73
Single tooth contact factor	[ZB,ZD]	1.00	1.00
Contact stress (N/mm ²)	[sigHB, sigHD]	1516.73	1516.73
Lubrication coefficient at NL	[ZL]	1.020	1.020
Speed coefficient at NL	[ZV]	0.979	0.979
Roughness coefficient at NL	[ZR]	1.016	1.016
Material pairing coefficient at NL	[ZW]	1.000	1.000
Finite life factor	[ZNT]	0.902	0.913
	[ZL * ZV * ZR * ZNT]	0.915	0.926
Limited pitting is permitted:	No		
Size factor (flank)	[ZX]	1.000	1.000
Permissible contact stress (N/mm ²)	[sigHP=sigHG/SHmin]	1407.82	1425.04
Pitting stress limit (N/mm ²)	[sigHG]	1372.62	1389.41
Required safety	[SHmin]	0.97	0.97
Safety factor for contact stress at operating pitch circle			
	[SHw]	0.90	0.92
Safety for stress at single tooth contact	[SHBD=sigHG/sigHBD]	0.90	0.92
(Safety regarding transmittable torque)	[(SHBD)^2]	0.82	0.84
Transmittable power (kW)	[kWRating]	51.73	53.00

4b. MICROPITTING ACCORDING TO ISO/TR 15144-1:2014

Calculation did not run. (Lubricant: Load stage micropitting test is unknown.)

5. SCUFFING LOAD CAPACITY

Calculation method according to ISO TR 13989:2000

Lubrication coefficient (for lubrication type)	[XS]	1.000	
Scuffing test and load stage	[FZGtest]	FZG - Test A / 8.3 / 90 (ISO 14635 - 1)	12
Multiple meshing factor	[Xmp]	1.000	
Relative structure coefficient (Scuffing)	[XWrelT]	1.000	
Thermal contact factor (N/mm/s ^{0.5} /K)	[BM]	13.780	13.780
Relevant tip relief (μm)	[Ca]	2.00	2.00
Optimal tip relief (μm)	[Ceff]	38.31	
Ca taken as optimal in the calculation (0=no, 1=yes)		0	0
Effective facewidth (mm)	[beff]	29.750	
Applicable circumferential force/facewidth (N/mm)	[wBt]	612.044	
Kbg =	1.283, wBt*Kbg =	785.180	

Angle factor ($\epsilon_1:0.516$, $\epsilon_2:0.484$)	[Xalfbet]	1.079
Flash temperature-criteria		
Lubricant factor	[XL]	0.830
Tooth mass temperature (°C) ($theMi = theoil + XS \cdot 0.47 \cdot Xmp \cdot theflm$)	[theMi]	75.92
Average flash temperature (°C)	[theflm]	12.60
Scuffing temperature (°C)	[theS]	348.80
Coordinate gamma (point of highest temp.) [Gamma.A]=-0.154 [Gamma.E]=0.164	[Gamma]	-0.154
Highest contact temp. (°C)	[theB]	102.00
Flash factor ($^{\circ}K \cdot N^{\wedge}-.75 \cdot s^{\wedge}.5 \cdot m^{\wedge}-.5 \cdot mm$)	[XM]	50.058
Approach factor	[XJ]	1.082
Load sharing factor	[XGam]	1.000
Dynamic viscosity (mPa*s)	[etaM]	41.90 (70.0 °C)
Coefficient of friction	[mym]	0.057
Required safety	[SBmin]	2.000
Safety factor for scuffing (flash temperature)	[SB]	8.712
Integral temperature-criteria		
Lubricant factor	[XL]	1.000
Tooth mass temperature (°C) ($theMC = theoil + XS \cdot 0.70 \cdot theflaint$)	[theMC]	80.35
Mean flash temperature (°C)	[theflaint]	14.78
Integral scuffing temperature (°C)	[theSint]	360.78
Flash factor ($^{\circ}K \cdot N^{\wedge}-.75 \cdot s^{\wedge}.5 \cdot m^{\wedge}-.5 \cdot mm$)	[XM]	50.058
Running-in factor (well run in)	[XE]	1.000
Contact ratio factor	[Xeps]	0.484
Dynamic viscosity (mPa*s)	[etaOil]	41.90 (70.0 °C)
Mean coefficient of friction	[mym]	0.060
Geometry factor	[XBE]	0.098
Meshing factor	[XQ]	1.000
Tip relief factor	[XCa]	1.051
Integral tooth flank temperature (°C)	[theint]	102.52
Required safety	[SSmin]	1.800
Safety factor for scuffing (intg.-temp.)	[SSint]	3.519
Safety referring to transmittable torque	[SSL]	8.940

6. MEASUREMENTS FOR TOOTH THICKNESS

		----- Gear 1 ----- Gear 2 --	
		DIN 3967 cd25	DIN 3967 cd25
Tooth thickness deviation		-0.070 /	-0.110
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.070 /	-0.110
Number of teeth spanned	[k]	8.000	11.000
Base tangent length (no backlash) (mm)	[Wk]	41.505	58.119
Actual base tangent length ('span') (mm)	[Wk.e/i]	41.439 /	41.401
(mm)	[ΔWk.e/i]	-0.066 /	-0.103
Diameter of measuring circle (mm)	[dMWk.m]	75.622	110.744
Theoretical diameter of ball/pin (mm)	[DM]	3.734	3.764
Effective diameter of ball/pin (mm)	[DMeff]	3.750	4.000
Radial single-ball measurement backlash free (mm)	[MrK]	40.016	58.137
Radial single-ball measurement (mm)	[MrK.e/i]	39.949 /	39.911
Diameter of measuring circle (mm)	[dMMr.m]	74.332	110.232

Diametral measurement over two balls without clearance (mm)	[MdK]	79.963	116.227
Diametral two ball measure (mm)	[MdK.e/i]	79.829 / 79.753	116.090 / 116.011
Diametral measurement over pins without clearance (mm)	[MdR]	80.031	116.273
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	79.898 / 79.821	116.136 / 116.057
Measurement over 2 pins (free) according to AGMA 2002 (mm)	[dk2f.e/i]	79.813 / 79.737	116.080 / 116.001
Measurement over 2 pins (transverse) according to AGMA 2002 (mm)	[dk2t.e/i]	79.965 / 79.888	116.181 / 116.102
Measurement over 3 pins (axial) according to AGMA 2002 (mm)	[dk3A.e/i]	79.898 / 79.821	116.136 / 116.057
Chordal tooth thickness (no backlash) (mm)	[sc]	4.492	5.101
Actual chordal tooth thickness (mm)	[sc.e/i]	4.422 / 4.382	5.031 / 4.991
Reference chordal height from da.m (mm)	[ha]	3.442	4.269
Tooth thickness (Arc) (mm)	[sn]	4.494	5.102
(mm)	[sn.e/i]	4.424 / 4.384	5.032 / 4.992
Backlash free center distance (mm)	[aControl.e/i]	91.351 / 91.265	
Backlash free center distance, allowances (mm)	[jta]	-0.149 / -0.235	
dNf.i with aControl (mm)	[dNf0.i]	70.829	106.409
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	0.155	0.081
Tip clearance (mm)	[c0.i(aControl)]	0.300	0.300
Centre distance allowances (mm)	[Aa.e/i]	0.018 / -0.018	
Circumferential backlash from Aa (mm)	[jtw_Aa.e/i]	0.019 / -0.019	
Radial clearance (mm)	[jrw]	0.252 / 0.132	
Circumferential backlash (transverse section) (mm)	[jtw]	0.269 / 0.140	
Normal backlash (mm)	[jnw]	0.235 / 0.123	
Torsional angle at entry with fixed output:			
Entire torsional angle (°)	[j.tSys]	0.4185 / 0.2186	

7. GEAR ACCURACY

		----- GEAR 1 -----	GEAR 2 --
According to ISO 1328-1:1995, ISO 1328-2:1997			
Accuracy grade	[Q]	6	6
Single pitch deviation (µm)	[fptT]	7.50	7.50
Base circle pitch deviation (µm)	[fpbT]	7.00	7.00
Sector pitch deviation over k/8 pitches (µm)	[Fpk/8T]	12.00	13.00
Profile form deviation (µm)	[ffaT]	6.50	6.50
Profile slope deviation (µm)	[fHaT]	5.50	5.50
Total profile deviation (µm)	[FaT]	8.50	8.50
Helix form deviation (µm)	[ffbT]	8.50	8.50
Helix slope deviation (µm)	[fHbT]	8.50	8.50
Total helix deviation (µm)	[FbT]	12.00	12.00
Total cumulative pitch deviation (µm)	[FpT]	26.00	26.00
Runout (µm)	[FrT]	21.00	21.00
Single flank composite, total (µm)	[FisT]	36.00	36.00
Single flank composite, tooth-to-tooth (µm)	[fisT]	10.00	10.00
Radial composite, total (µm)	[FidT]	31.00	31.00
Radial composite, tooth-to-tooth (µm)	[fidT]	9.50	9.50

Axis alignment tolerances (recommendation acc. to ISO TR 10064-3:1996, Quality)

6)

Maximum value for deviation error of axis (µm)	[fSigbet]	12.00 (Fb= 12.00)
Maximum value for inclination error of axes (µm)	[fSigdel]	24.00

8. ADDITIONAL DATA

Maximal possible centre distance (eps_a=1.0)	[aMAX]	91.501	
Mass (kg)	[m]	0.982	2.168
Total mass (kg)	[m]	3.150	
Moment of inertia (system with reference to the drive): calculation without consideration of the exact tooth shape			
single gears ((da+df)/2...di) (kg*m²)	[TraeghMom]	0.0006593	0.00321
System ((da+df)/2...di) (kg*m²)	[TraeghMom]	0.002086	
Torsional stiffness at entry with driven force fixed:			
Torsional stiffness (MNm/rad)	[cr]	0.379	
Torsion when subjected to nominal torque (°)	[delcr]	0.072	
Mean coeff. of friction (acc. Niemann)	[mum]	0.064	
Wear sliding coef. by Niemann	[zetw]	0.234	
Gear power loss (kW)	[PVZ]	0.289	
(Meshing efficiency (%))	[etaz]	99.518)	
Sound pressure level (according to Masuda)	[dB(A)]	75.5	
Indications for the manufacturing by wire cutting:			
Deviation from theoretical tooth trace (µm)	[WireErr]	501.2	337.6
Permissible deviation (µm)	[Fb/2]	6.0	6.0

9. MODIFICATIONS AND TOOTH FORM DEFINITION

Data for the tooth form calculation :
Data not available.

10. SERVICE LIFE, DAMAGE

Required safety for tooth root	[SFmin]	1.35	
Required safety for tooth flank	[SHmin]	0.97	
Service life (calculated with required safeties):			
System service life (h)	[Hatt]	2.490	
Tooth root service life (h)	[HFatt]	2.49	7.074
Tooth flank service life (h)	[HHatt]	1762	2620
Damage calculated on the basis of the required service life [H] (20000.0 h)			
F1%	F2%	H1%	H2%
9999.99	9999.99	1134.76	763.39
Damage calculated on basis of system service life [Hatt] (2.5 h)			
F1%	F2%	H1%	H2%
100.00	35.19	0.14	0.10

Calculation of the factors required to define reliability R(t) according to B. Bertsche with Weibull distribution:

$$R(t) = 100 * \exp(-((t*fac - t_0)/(T - t_0))^b) \% ; t (h)$$

Gear		fac	b	t0	T	R(H)%
1	Tooth root	72000	1.7	1.731e+005	2.659e+005	0.00
1	Tooth flank	72000	1.3	1.144e+008	5.45e+008	1.34

2	Tooth root	48436	1.7	3.308e+005	5.084e+005	0.00
2	Tooth flank	48436	1.3	1.144e+008	5.45e+008	8.74

Reliability of the configuration for required service life (%) 0.00 (Bertsche)

REMARKS:

- Specifications with [e/i] imply: Maximum [e] and Minimal value [i] with consideration of all tolerances
- Specifications with [m] imply: Mean value within tolerance
- For the backlash tolerance, the center distance tolerances and the tooth thickness deviation are taken into account. Shown is the maximal and the minimal backlash corresponding the largest resp. the smallest allowances
- The calculation is done for the operating pitch circle.
- Calculation of Zbet according Corrigendum 1 ISO 6336-2:2008 with $Z_{bet} = 1/(\cos(\beta))^0.5$
- Details of calculation method:
 - cg according to method B
 - KV according to method B
 - KHb, KFb according method C
 - fma following equation (64), fsh following (57/58), Fbx following (52/53/57)
 - KHa, KFa according to method B
- The logarithmically interpolated value taken from the values for the fatigue strength and the static strength, based on the number of load cycles, is used for coefficients ZL, ZV, ZR, ZW, ZX, YdreIT, YRreIT and YX..

End of Report

lines: 533

B.4 CMS50-E

Name : Unnamed

Changed by: up201306219

on: 21.04.2018

at: 12:56:08

CALCULATION OF A HELICAL GEAR PAIR

Drawing or article number:

Gear 1: 0.000.0

Gear 2: 0.000.0

Calculation method

ISO 6336:2006 Method B

----- GEAR 1 ----- GEAR 2 --

Power (kW)	[P]	60.040	
Speed (1/min)	[n]	1200.0	800.0
Torque (Nm)	[T]	477.8	716.7
Application factor	[KA]		1.25
Required service life (h)	[H]	20000.00	
Gear driving (+) / driven (-)		+	-
Working flank gear 1: Right flank			
Sense of rotation gear 1 clockwise			

1. TOOTH GEOMETRY AND MATERIAL

(geometry calculation according to ISO 21771:2007, DIN ISO 21771)

----- GEAR 1 ----- GEAR 2 --

Center distance (mm)	[a]	91.500	
Centre distance tolerance	ISO 286:2010 Measure js7		
Normal module (mm)	[mn]	3.2500	
Pressure angle at normal section (°)	[alfn]	20.0000	
Helix angle at reference circle (°)	[beta]	18.2219	
Number of teeth	[z]	20	30
Facewidth (mm)	[b]	32.65	32.65
Hand of gear		right	left
Accuracy grade	[Q-ISO 1328:1995]	6	6
Inner diameter (mm)	[di]	0.00	0.00
Inner diameter of gear rim (mm)	[dbi]	0.00	0.00

Material

Gear 1:

16 MnCr 5 (1), Case-carburized steel, case-hardened
ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ Jominy J=12mm<HRC28

Gear 2:

16 MnCr 5 (1), Case-carburized steel, case-hardened
ISO 6336-5 Figure 9/10 (MQ), Core hardness $\geq 25\text{HRC}$ Jominy J=12mm<HRC28

----- GEAR 1 ----- GEAR 2 --

Surface hardness		HRC 59	HRC 59
Material quality according to ISO 6336:2006 Normal (Life factors ZNT and YNT ≥ 0.85)			
Fatigue strength. tooth root stress (N/mm ²)	[σFlim]	430.00	430.00
Fatigue strength for Hertzian pressure (N/mm ²)	[σHlim]	1500.00	1500.00
Tensile strength (N/mm ²)	[σB]	1000.00	1000.00
Yield point (N/mm ²)	[σS]	695.00	695.00
Young's modulus (N/mm ²)	[E]	206000	206000
Poisson's ratio	[ν]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.35	0.35

Roughness average value DS, root (µm)	[RAF]	3.00	3.00
Mean roughness height, Rz, flank (µm)	[RZH]	2.50	2.50
Mean roughness height, Rz, root (µm)	[RZF]	20.00	20.00

Gear reference profile 1 :

Reference profile 1.25 / 0.25 / 1.0 ISO 53:1998 Profil C

Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[rhofP*]	0.250 (rhofPmax*=0.472)	
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[rhoaP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[alfprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[alfKP]	0.000	

not topping

Gear reference profile 2 :

Reference profile 1.25 / 0.25 / 1.0 ISO 53:1998 Profil C

Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[rhofP*]	0.250 (rhofPmax*=0.472)	
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[rhoaP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[alfprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[alfKP]	0.000	

not topping

Summary of reference profile gears:

Dedendum reference profile	[hfP*]	1.250	1.250
Tooth root radius Refer. profile	[rofpP*]	0.250	0.250
Addendum Reference profile	[haP*]	1.000	1.000
Protuberance height coefficient	[hprP*]	0.000	0.000
Protuberance angle (°)	[alfprP]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[alfKP]	0.000	0.000

Type of profile modification: none (only running-in)

Tip relief (µm)	[Ca]	2.0	2.0
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Lubrication type Oil bath lubrication

Type of oil Oil: ISO-VG 220

Lubricant base Mineral-oil base

Kinem. viscosity oil at 40 °C (mm²/s)	[nu40]	220.00	
Kinem. viscosity oil at 100 °C (mm²/s)	[nu100]	17.50	
Specific density at 15 °C (kg/dm³)	[roOil]	0.895	
Oil temperature (°C)	[TS]	70.000	

----- GEAR 1 ----- GEAR 2 --

Overall transmission ratio	[itot]	-1.500	
Gear ratio	[u]	1.500	
Transverse module (mm)	[mt]	3.422	
Pressure angle at pitch circle (°)	[alf]	20.966	
Working transverse pressure angle (°)	[alfwt]	29.195	
	[alfwt.e/i]	29.215 / 29.175	
Working pressure angle at normal section (°)	[alfwn]	27.791	

Helix angle at operating pitch circle (°)	[betaw]	19.400		
Base helix angle (°)	[betab]	17.088		
Reference centre distance (mm)	[ad]	85.540		
Sum of profile shift coefficients	[Summexi]	2.1953		
Profile shift coefficient	[x]	1.0110	1.1843	
Tooth thickness (Arc) (module) (module)	[sn*]	2.3067	2.4329	
Tip alteration (mm)	[k*mn]	-1.174	-1.174	
Reference diameter (mm)	[d]	68.432	102.648	
Base diameter (mm)	[db]	63.901	95.851	
Tip diameter (mm)	[da]	79.155	114.497	
(mm)	[da.e/i]	79.155 / 79.145	114.497 / 114.487	
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.010	0.000 / -0.010	
Tip form diameter (mm)	[dFa]	79.155	114.497	
(mm)	[dFa.e/i]	79.155 / 79.145	114.497 / 114.487	
Active tip diameter (mm)	[dNa]	79.155	114.497	
Active tip diameter (mm)	[dNa.e/i]	79.155 / 79.145	114.497 / 114.487	
Operating pitch diameter (mm)	[dw]	73.200	109.800	
(mm)	[dw.e/i]	73.214 / 73.186	109.821 / 109.779	
Root diameter (mm)	[df]	66.878	102.220	
Generating Profile shift coefficient	[xE.e/i]	0.9814/ 0.9645	1.1547/ 1.1378	
Manufactured root diameter with xE (mm)	[df.e/i]	66.686 / 66.576	102.028 / 101.918	
Theoretical tip clearance (mm)	[c]	0.812	0.812	
Effective tip clearance (mm)	[c.e/i]	0.986 / 0.891	0.986 / 0.891	
Active root diameter (mm)	[dNf]	69.231	104.872	
(mm)	[dNf.e/i]	69.266 / 69.203	104.908 / 104.842	
Root form diameter (mm)	[dFf]	67.959	103.303	
(mm)	[dFf.e/i]	67.778 / 67.676	103.104 / 102.991	
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.795 / 0.713	0.958 / 0.869	
Addendum (mm)	[ha=mn*(haP*+x+k)]	5.362	5.925	
(mm)	[ha.e/i]	5.362 / 5.357	5.925 / 5.920	
Dedendum (mm)	[hf=mn*(hfP*-x)]	0.777	0.214	
(mm)	[hf.e/i]	0.873 / 0.928	0.310 / 0.365	
Roll angle at dFa (°)	[xsi_dFa.e/i]	41.885 / 41.870	37.436 / 37.425	
Roll angle to dNa (°)	[xsi_dNa.e/i]	41.885 / 41.870	37.436 / 37.425	
Roll angle to dNf (°)	[xsi_dNf.e/i]	23.965 / 23.820	25.488 / 25.392	
Roll angle at dFf (°)	[xsi_dFf.e/i]	20.259 / 19.984	22.706 / 22.522	
Tooth height (mm)	[h]	6.139	6.139	
Virtual gear no. of teeth	[zn]	23.046	34.569	
Normal tooth thickness at tip circle (mm)	[san]	2.427	2.550	
(mm)	[san.e/i]	2.354 / 2.302	2.479 / 2.429	
Normal-tooth thickness on tip form circle (mm)	[sFan]	2.427	2.550	
(mm)	[sFan.e/i]	2.354 / 2.302	2.479 / 2.429	
Normal space width at root circle (mm)	[efn]	2.150	2.142	
(mm)	[efn.e/i]	2.157 / 2.162	2.141 / 2.141	
Max. sliding velocity at tip (m/s)	[vga]	1.153	0.950	
Specific sliding at the tip	[zetaaa]	0.393	0.362	
Specific sliding at the root	[zetaf]	-0.567	-0.647	
Mean specific sliding	[zetam]	0.379		
Sliding factor on tip	[Kga]	0.251	0.206	
Sliding factor on root	[Kgf]	-0.206	-0.251	
Pitch on reference circle (mm)	[pt]	10.749		
Base pitch (mm)	[pbt]	10.038		
Transverse pitch on contact-path (mm)	[pet]	10.038		
Lead height (mm)	[pz]	653.038	979.556	
Axial pitch (mm)	[px]	32.652		
Length of path of contact (mm)	[ga, e/i]	10.038 (10.074 /	9.985)	

Length T1-A, T2-A (mm)	[T1A, T2A]	13.319(	13.283/	13.364)	31.313(	31.313/	31.304)
Length T1-B (mm)	[T1B, T2B]	13.320(	13.320/	13.364)	31.313(	31.277/	31.304)
Length T1-C (mm)	[T1C, T2C]	17.853(	17.839/	17.867)	26.779(	26.758/	26.801)
Length T1-D (mm)	[T1D, T2D]	23.357(	23.321/	23.349)	21.276(	21.276/	21.320)
Length T1-E (mm)	[T1E, T2E]	23.357(	23.357/	23.349)	21.275(	21.239/	21.320)
Length T1-T2 (mm)	[T1T2]		44.632 (	44.596 /	44.668)		
Diameter of single contact point B (mm)	[d-B]	69.231(	69.231/	69.266)	114.497(	114.457/	114.487)
Diameter of single contact point D (mm)	[d-D]	79.155(	79.112/	79.145)	104.872(	104.872/	104.908)
Addendum contact ratio	[eps]	0.548(	0.550/	0.546)	0.452(	0.454/	0.449)
Minimal length of contact line (mm)	[Lmin]			34.162			
Transverse contact ratio	[eps_a]			1.000			
Transverse contact ratio with allowances	[eps_a.e/m/i]		1.004 /	0.999 /	0.995		
Overlap ratio	[eps_b]			1.000			
Total contact ratio	[eps_g]			2.000			
Total contact ratio with allowances	[eps_g.e/m/i]		2.004 /	1.999 /	1.995		

2. FACTORS OF GENERAL INFLUENCE

		----- GEAR 1 -----	GEAR 2 --
Nominal circum. force at pitch circle (N)	[Ft]		13963.7
Axial force (N)	[Fa]		4596.9
Radial force (N)	[Fr]		5350.7
Normal force (N)	[Fnorm]		15644.4
Nominal circumferential force per mm (N/mm)	[w]		427.66
Only as information: Forces at operating pitch circle:			
Nominal circumferential force (N)	[Ftw]		13054.1
Axial force (N)	[Faw]		4596.9
Radial force (N)	[Frw]		7294.2
Circumferential speed reference circle (m/s)	[v]		4.30
Circumferential speed operating pitch circle (m/s)	[v(dw)]		4.60
Running-in value (μm)	[yp]		0.6
Running-in value (μm)	[yf]		0.6
Correction coefficient	[CM]		0.800
Gear body coefficient	[CR]		1.000
Basic rack factor	[CBS]		0.975
Material coefficient	[E/Est]		1.000
Singular tooth stiffness (N/mm/μm)	[c']		15.666
Meshing stiffness (N/mm/μm)	[cgalf]		15.666
Meshing stiffness (N/mm/μm)	[cgbet]		13.316
Reduced mass (kg/mm)	[mRed]		0.01462
Resonance speed (min-1)	[nE1]		15629
Resonance ratio (-)	[N]		0.077
Subcritical range			
Running-in value (μm)	[ya]		0.6
Bearing distance l of pinion shaft (mm)	[l]		65.300
Distance s of pinion shaft (mm)	[s]		6.530
Outside diameter of pinion shaft (mm)	[dsh]		32.650
Load in accordance with Figure 13, ISO 6336-1:2006 0:a), 1:b), 2:c), 3:d), 4:e)	[-]	4	
Coefficient K' according to Figure 13, ISO 6336-1:2006	[K']	-1.00	
Without support effect			
Tooth trace deviation (active) (μm)	[Fby]		3.61
from deformation of shaft (μm)	[fsh*B1]		3.91
(fsh (μm) = 3.91, B1= 1.00, fHb5 (μm) = 6.00)			

Tooth without tooth trace modification			
Position of Contact pattern: favorable			
from production tolerances (μm)	[fma*B2]	12.02	
(B2= 1.00)			
Tooth trace deviation, theoretical (μm)	[Fbx]	4.25	
Running-in value (μm)	[yb]	0.64	
Dynamic factor	[KV]	1.028	
Face load factor - flank	[KHb]	1.044	
- Tooth root	[KFb]	1.035	
- Scuffing	[KBb]	1.044	
Transverse load factor - flank	[KHa]	1.000	
- Tooth root	[KF _a]	1.000	
- Scuffing	[KB _a]	1.000	
Helical load factor scuffing	[Kbg]	1.003	
Number of load cycles (in mio.)	[NL]	1440.000	960.000

3. TOOTH ROOT STRENGTH

Calculation of Tooth form coefficients according method: B

		----- GEAR 1 -----	GEAR 2 --
Calculated with profile shift	[x]	1.0110	1.1843
Tooth form factor	[YF]	1.33	1.33
Stress correction factor	[YS]	2.62	2.60
Working angle (°)	[alfFen]	31.70	29.68
Bending moment arm (mm)	[hF]	4.57	4.57
Tooth thickness at root (mm)	[sFn]	7.78	7.86
Tooth root radius (mm)	[roF]	0.81	0.85
(hF* = 1.405/ 1.408 sFn* = 2.393/ 2.419 roF* = 0.250/ 0.262)			
(den (mm) =			
84.602/ 123.251 dsFn(mm) = 67.575/ 102.968 alfsFn(°) = 30.00/ 30.00 qs = 4.786/ 4.620)			
Helix angle factor	[Ybet]	0.848	
Deep tooth factor	[YDT]	1.000	
Gear rim factor	[YB]	1.00	1.00
Effective facewidth (mm)	[beff]	32.65	32.65
Nominal stress at tooth root (N/mm²)	[sigF0]	389.59	386.40
Tooth root stress (N/mm²)	[sigF]	518.36	514.12
Permissible bending stress at root of Test-gear			
Notch sensitivity factor	[YdreIT]	1.019	1.017
Surface factor	[YRrelT]	0.957	0.957
size factor (Tooth root)	[YX]	1.000	1.000
Finite life factor	[YNT]	0.884	0.891
	[YdreIT*YRrelT*YX*YNT]	0.861	0.867
Alternating bending factor (mean stress influence coefficient)	[YM]	1.000	1.000
Stress correction factor	[Yst]	2.00	
Yst*sigFlim (N/mm²)	[sigFE]	860.00	860.00
Permissible tooth root stress (N/mm²)	[sigFP=sigFG/SFmin]	528.96	532.65
Limit strength tooth root (N/mm²)	[sigFG]	740.54	745.71
Required safety	[SFmin]	1.40	1.40
Safety for tooth root stress	[SF=sigFG/sigF]	1.43	1.45

Transmittable power (kW)	[kWRating]	61.27	62.20
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4. SAFETY AGAINST PITTING (TOOTH FLANK)

		----- GEAR 1 -----	GEAR 2 --
Zone factor	[ZH]		1.981
Elasticity factor ($\sqrt{N/mm^2}$)	[ZE]		189.812
Contact ratio factor	[Zeps]		1.000
Helix angle factor	[Zbet]		1.026
Effective facewidth (mm)	[beff]		32.65
Nominal contact stress (N/mm^2)	[sigH0]		1245.00
Contact stress at operating pitch circle (N/mm^2)	[sigHw]		1441.82
Single tooth contact factor	[ZB,ZD]	1.00	1.00
Contact stress (N/mm^2)	[sigHB, sigHD]	1441.83	1441.82
Lubrication coefficient at NL	[ZL]	1.020	1.020
Speed coefficient at NL	[ZV]	0.979	0.979
Roughness coefficient at NL	[ZR]	1.017	1.017
Material pairing coefficient at NL	[ZW]	1.000	1.000
Finite life factor	[ZNT]	0.902	0.913
	[ZL * ZV * ZR * ZNT]	0.915	0.927
Limited pitting is permitted:	No		
Size factor (flank)	[ZX]	1.000	1.000
Permissible contact stress (N/mm^2)	[sigHP=sigHG/SHmin]	1373.20	1390.38
Pitting stress limit (N/mm^2)	[sigHG]	1373.20	1390.38
Required safety	[SHmin]	1.00	1.00
Safety factor for contact stress at operating pitch circle			
	[SHw]	0.95	0.96
Safety for stress at single tooth contact	[SHBD=sigHG/sigHBD]	0.95	0.96
(Safety regarding transmittable torque)	[(SHBD)^2]	0.91	0.93
Transmittable power (kW)	[kWRating]	54.46	55.83

4b. MICROPITTING ACCORDING TO ISO/TR 15144-1:2014

Calculation did not run. (Lubricant: Load stage micropitting test is unknown.)

5. SCUFFING LOAD CAPACITY

Calculation method according to ISO TR 13989:2000

Lubrication coefficient (for lubrication type)	[XS]	1.000	
Scuffing test and load stage	[FZGtest]	FZG - Test A / 8.3 / 90 (ISO 14635 - 1)	12
Multiple meshing factor	[Xmp]	1.000	
Relative structure coefficient (Scuffing)	[XWrelT]	1.000	
Thermal contact factor ($N/mm/s^{.5/K}$)	[BM]	13.780	13.780
Relevant tip relief (μm)	[Ca]	2.00	2.00
Optimal tip relief (μm)	[Ceff]	34.12	
Ca taken as optimal in the calculation (0=no, 1=yes)		0	0
Effective facewidth (mm)	[beff]	32.652	
Applicable circumferential force/facewidth (N/mm)	[wBt]	573.559	
Kbg =	1.003, wBt*Kbg =	574.998	

Angle factor ($\epsilon_1:0.548$, $\epsilon_2:0.452$)	[Xalfbet]	1.098
Flash temperature-criteria		
Lubricant factor	[XL]	0.830
Tooth mass temperature (°C) ($theMi = theoil + XS \cdot 0.47 \cdot Xmp \cdot theflm$)	[theMi]	79.81
Average flash temperature (°C)	[theflm]	20.88
Scuffing temperature (°C)	[theS]	348.80
Coordinate gamma (point of highest temp.) [Gamma.A]=-0.254 [Gamma.E]=0.308	[Gamma]	0.308
Highest contact temp. (°C)	[theB]	123.54
Flash factor ($^{\circ}K \cdot N^{\wedge}-.75 \cdot s^{\wedge}.5 \cdot m^{\wedge}-.5 \cdot mm$)	[XM]	50.058
Approach factor	[XJ]	1.000
Load sharing factor	[XGam]	1.000
Dynamic viscosity (mPa*s)	[etaM]	41.90 (70.0 °C)
Coefficient of friction	[mym]	0.056
Required safety	[SBmin]	2.000
Safety factor for scuffing (flash temperature)	[SB]	5.207
Integral temperature-criteria		
Lubricant factor	[XL]	1.000
Tooth mass temperature (°C) ($theMC = theoil + XS \cdot 0.70 \cdot theflaint$)	[theMC]	83.50
Mean flash temperature (°C)	[theflaint]	19.29
Integral scuffing temperature (°C)	[theSint]	360.78
Flash factor ($^{\circ}K \cdot N^{\wedge}-.75 \cdot s^{\wedge}.5 \cdot m^{\wedge}-.5 \cdot mm$)	[XM]	50.058
Running-in factor (well run in)	[XE]	1.000
Contact ratio factor	[Xeps]	0.460
Dynamic viscosity (mPa*s)	[etaOil]	41.90 (70.0 °C)
Mean coefficient of friction	[mym]	0.056
Geometry factor	[XBE]	0.182
Meshing factor	[XQ]	1.000
Tip relief factor	[XCa]	1.057
Integral tooth flank temperature (°C)	[theint]	112.43
Required safety	[SSmin]	1.800
Safety factor for scuffing (intg.-temp.)	[SSint]	3.209
Safety referring to transmittable torque	[SSL]	6.852

6. MEASUREMENTS FOR TOOTH THICKNESS

		----- Gear 1 ----- Gear 2 --	
		DIN 3967 cd25	DIN 3967 cd25
Tooth thickness deviation		-0.070 / -0.110	-0.070 / -0.110
Tooth thickness allowance (normal section) (mm)	[As.e/i]		
Number of teeth spanned	[k]	4.000	6.000
Base tangent length (no backlash) (mm)	[Wk]	36.882	56.983
Actual base tangent length ('span') (mm)	[Wk.e/i]	36.816 / 36.779	56.917 / 56.880
(mm)	[ΔWk.e/i]	-0.066 / -0.103	-0.066 / -0.103
Diameter of measuring circle (mm)	[dMWk.m]	72.942	110.206
Theoretical diameter of ball/pin (mm)	[DM]	7.500	7.079
Effective diameter of ball/pin (mm)	[DMeff]	7.500	7.500
Radial single-ball measurement backlash free (mm)	[MrK]	43.247	61.074
Radial single-ball measurement (mm)	[MrK.e/i]	43.189 / 43.155	61.011 / 60.975
Diameter of measuring circle (mm)	[dMMr.m]	74.870	110.729

Diametral measurement over two balls without clearance (mm)	[MdK]	86.495	122.148
Diametral two ball measure (mm)	[MdK.e/i]	86.378 / 86.310	122.023 / 121.951
Diametral measurement over pins without clearance (mm)	[MdR]	86.495	122.148
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	86.378 / 86.310	122.023 / 121.951
Measurement over 3 pins (axial) according to AGMA 2002 (mm)	[dk3A.e/i]	86.378 / 86.310	122.023 / 121.951
Chordal tooth thickness (no backlash) (mm)	[sc]	7.485	7.900
Actual chordal tooth thickness (mm)	[sc.e/i]	7.415 / 7.375	7.830 / 7.790
Reference chordal height from da.m (mm)	[ha]	5.545	6.060
Tooth thickness (Arc) (mm)	[sn]	7.497	7.907
(mm)	[sn.e/i]	7.427 / 7.387	7.837 / 7.797
Backlash free center distance (mm)	[aControl.e/i]	91.359 / 91.277	
Backlash free center distance, allowances (mm)	[jta]	-0.141 / -0.223	
dNf.i with aControl (mm)	[dNf0.i]	68.884	104.503
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	0.553	0.700
Tip clearance (mm)	[c0.i(aControl)]	0.686	0.686
Centre distance allowances (mm)	[Aa.e/i]	0.018 / -0.018	
Circumferential backlash from Aa (mm)	[jtw_Aa.e/i]	0.020 / -0.020	
Radial clearance (mm)	[jrw]	0.240 / 0.124	
Circumferential backlash (transverse section) (mm)	[jtw]	0.267 / 0.138	
Normal backlash (mm)	[jnw]	0.239 / 0.123	
Torsional angle at entry with fixed output:			
Entire torsional angle (°)	[j.tSys]	0.4185/0.2162	

7. GEAR ACCURACY

		----- GEAR 1 -----	GEAR 2 --
According to ISO 1328-1:1995, ISO 1328-2:1997			
Accuracy grade	[Q]	6	6
Single pitch deviation (μm)	[fptT]	8.50	8.50
Base circle pitch deviation (μm)	[fpbT]	7.90	7.90
Sector pitch deviation over k/8 pitches (μm)	[Fpk/8T]	12.00	13.00
Profile form deviation (μm)	[ffaT]	8.50	8.50
Profile slope deviation (μm)	[fHaT]	7.00	7.00
Total profile deviation (μm)	[FaT]	11.00	11.00
Helix form deviation (μm)	[ffbT]	8.50	8.50
Helix slope deviation (μm)	[fHbT]	8.50	8.50
Total helix deviation (μm)	[FbT]	12.00	12.00
Total cumulative pitch deviation (μm)	[FpT]	27.00	27.00
Runout (μm)	[FrT]	21.00	21.00
Single flank composite, total (μm)	[FisT]	42.00	42.00
Single flank composite, tooth-to-tooth (μm)	[fisT]	15.00	15.00
Radial composite, total (μm)	[FidT]	36.00	36.00
Radial composite, tooth-to-tooth (μm)	[fidT]	14.00	14.00

Axis alignment tolerances (recommendation acc. to ISO TR 10064-3:1996, Quality)

6)

Maximum value for deviation error of axis (μm)	[fSigbet]	12.00 (Fb= 12.00)
Maximum value for inclination error of axes (μm)	[fSigdel]	24.00

8. ADDITIONAL DATA

Mass (kg)	[m]	1.071	2.358
Total mass (kg)	[m]	3.428	
Moment of inertia (system with reference to the drive): calculation without consideration of the exact tooth shape			
single gears ((da+df)/2...di) (kg*m ²)	[TraeghMom]	0.0007134	0.00346
System ((da+df)/2...di) (kg*m ²)	[TraeghMom]	0.002251	
Torsional stiffness at entry with driven force fixed:			
Torsional stiffness (MNm/rad)	[cr]	0.415	
Torsion when subjected to nominal torque (°)	[delcr]	0.066	
Mean coeff. of friction (acc. Niemann)	[mum]	0.063	
Wear sliding coef. by Niemann	[zetw]	0.379	
Gear power loss (kW)	[PVZ]	0.519	
(Meshing efficiency (%))	[etaz]	99.135	
Sound pressure level (according to Masuda)	[dB(A)]	76.3	

9. MODIFICATIONS AND TOOTH FORM DEFINITION

Data for the tooth form calculation :

Data not available.

10. SERVICE LIFE, DAMAGE

Required safety for tooth root	[SFmin]	1.40
Required safety for tooth flank	[SHmin]	1.00

Service life (calculated with required safeties):

System service life (h)	[Hatt]	4080
-------------------------	--------	------

Tooth root service life (h)	[HFatt]	5.494e+004	1.171e+005
Tooth flank service life (h)	[HHatt]	4080	6120

Damage calculated on the basis of the required service life [H] (20000.0 h)

F1%	F2%	H1%	H2%
36.40	17.07	490.17	326.78

Damage calculated on basis of system service life [Hatt] (4080.2 h)

F1%	F2%	H1%	H2%
7.43	3.48	100.00	66.67

Calculation of the factors required to define reliability R(t) according to B. Bertsche with Weibull distribution:

$$R(t) = 100 * \exp(-((t^{*fac} - t_0)/(T - t_0))^b) \% ; t (h)$$

Gear		fac	b	t0	T	R(H)%
1	Tooth root	72000	1.7	3.819e+009	5.869e+009	100.00
1	Tooth flank	72000	1.3	2.648e+008	1.262e+009	28.98
2	Tooth root	48000	1.7	5.428e+009	8.342e+009	100.00
2	Tooth flank	48000	1.3	2.648e+008	1.262e+009	53.48

Reliability of the configuration for required service life (%) 15.50 (Bertsche)

REMARKS:

- Specifications with [e/i] imply: Maximum [e] and Minimal value [i] with consideration of all tolerances
- Specifications with [m] imply: Mean value within tolerance
- For the backlash tolerance, the center distance tolerances and the tooth thickness deviation are taken into account. Shown is the maximal and the minimal backlash corresponding

the largest resp. the smallest allowances

The calculation is done for the operating pitch circle.

- Calculation of Zbet according Corrigendum 1 ISO 6336-2:2008 with $Z_{bet} = 1/(\cos(\beta))^0.5$
- Details of calculation method:
 - cg according to method B
 - KV according to method B
 - KHb, KFb according method C
 - fma following equation (64), fsh following (57/58), Fbx following (52/53/57)
 - KHa, KFa according to method B
- The logarithmically interpolated value taken from the values for the fatigue strength and the static strength, based on the number of load cycles, is used for coefficients ZL, ZV, ZR, ZW, ZX, YdreIT, YRreIT and YX..

End of Report

lines: 524

B.5 CMS60-R

Name : Unnamed

Changed by: up201306219

on: 22.04.2018

at: 12:10:02

CALCULATION OF A HELICAL GEAR PAIR

Drawing or article number:

Gear 1: 0.000.0

Gear 2: 0.000.0

Calculation method

ISO 6336:2006 Method B

----- GEAR 1 ----- GEAR 2 --

Power (kW)	[P]	60.040	
Speed (1/min)	[n]	1200.0	800.0
Torque (Nm)	[T]	477.8	716.7
Application factor	[KA]		1.25
Required service life (h)	[H]	20000.00	
Gear driving (+) / driven (-)		+	-
Working flank gear 1: Right flank			
Sense of rotation gear 1 clockwise			

1. TOOTH GEOMETRY AND MATERIAL

(geometry calculation according to ISO 21771:2007, DIN ISO 21771)

----- GEAR 1 ----- GEAR 2 --

Center distance (mm)	[a]	91.500	
Centre distance tolerance	ISO 286:2010 Measure js7		
Normal module (mm)	[mn]	2.7500	
Pressure angle at normal section (°)	[alfn]	20.0000	
Helix angle at reference circle (°)	[beta]	28.6452	
Number of teeth	[z]	24	36
Facewidth (mm)	[b]	36.04	36.04
Hand of gear		right	left
Accuracy grade	[Q-ISO 1328:1995]	6	6
Inner diameter (mm)	[di]	0.00	0.00
Inner diameter of gear rim (mm)	[dbi]	0.00	0.00

Material

Gear 1:

16 MnCr 5 (1), Case-carburized steel, case-hardened
ISO 6336-5 Figure 9/10 (MQ), Core hardness >=25HRC Jominy J=12mm<HRC28

Gear 2:

16 MnCr 5 (1), Case-carburized steel, case-hardened
ISO 6336-5 Figure 9/10 (MQ), Core hardness >=25HRC Jominy J=12mm<HRC28

----- GEAR 1 ----- GEAR 2 --

Surface hardness		HRC 59	HRC 59
Material quality according to ISO 6336:2006 Normal (Life factors ZNT and YNT >=0.85)			
Fatigue strength. tooth root stress (N/mm ²)	[σFlim]	430.00	430.00
Fatigue strength for Hertzian pressure (N/mm ²)	[σHlim]	1500.00	1500.00
Tensile strength (N/mm ²)	[σB]	1000.00	1000.00
Yield point (N/mm ²)	[σS]	695.00	695.00
Young's modulus (N/mm ²)	[E]	206000	206000
Poisson's ratio	[ν]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.35	0.35

Roughness average value DS, root (µm)	[RAF]	3.00	3.00
Mean roughness height, Rz, flank (µm)	[RZH]	2.50	2.50
Mean roughness height, Rz, root (µm)	[RZF]	20.00	20.00

Gear reference profile 1 :

Reference profile 1.25 / 0.25 / 1.0 ISO 53:1998 Profil C

Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[rhofP*]	0.250 (rhofPmax*=0.472)	
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[rhoaP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[alfprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[alfKP]	0.000	

not topping

Gear reference profile 2 :

Reference profile 1.25 / 0.25 / 1.0 ISO 53:1998 Profil C

Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[rhofP*]	0.250 (rhofPmax*=0.472)	
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[rhoaP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[alfprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[alfKP]	0.000	

not topping

Summary of reference profile gears:

Dedendum reference profile	[hfP*]	1.250	1.250
Tooth root radius Refer. profile	[rofpP*]	0.250	0.250
Addendum Reference profile	[haP*]	1.000	1.000
Protuberance height coefficient	[hprP*]	0.000	0.000
Protuberance angle (°)	[alfprP]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[alfKP]	0.000	0.000

Type of profile modification: none (only running-in)

Tip relief (µm)	[Ca]	2.0	2.0
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Lubrication type Oil bath lubrication

Type of oil Oil: ISO-VG 220

Lubricant base Mineral-oil base

Kinem. viscosity oil at 40 °C (mm²/s)	[nu40]	220.00	
Kinem. viscosity oil at 100 °C (mm²/s)	[nu100]	17.50	
Specific density at 15 °C (kg/dm³)	[roOil]	0.895	
Oil temperature (°C)	[TS]	70.000	

----- GEAR 1 ----- GEAR 2 --

Overall transmission ratio	[itot]	-1.500	
Gear ratio	[u]	1.500	
Transverse module (mm)	[mt]	3.134	
Pressure angle at pitch circle (°)	[alf]	22.525	
Working transverse pressure angle (°)	[alfwt]	18.376	
	[alfwt.e/i]	18.409 /	18.343
Working pressure angle at normal section (°)	[alfwn]	16.347	

Helix angle at operating pitch circle (°)	[betaw]	27.999		
Base helix angle (°)	[betab]	26.774		
Reference centre distance (mm)	[ad]	94.006		
Sum of profile shift coefficients	[Summexi]	-0.8342		
Profile shift coefficient	[x]	-0.2944		-0.5398
Tooth thickness (Arc) (module) (module)	[sn*]	1.3565		1.1778
Tip alteration (mm)	[k*mn]	-0.212		-0.212
Reference diameter (mm)	[d]	75.205		112.807
Base diameter (mm)	[db]	69.467		104.201
Tip diameter (mm)	[da]	78.661		114.914
(mm)	[da.e/i]	78.661 /	78.651	114.914 / 114.904
Tip diameter allowances (mm)	[Ada.e/i]	0.000 /	-0.010	0.000 / -0.010
Tip form diameter (mm)	[dFa]	78.661		114.914
(mm)	[dFa.e/i]	78.661 /	78.651	114.914 / 114.904
Active tip diameter (mm)	[dNa]	78.661		114.914
Active tip diameter (mm)	[dNa.e/i]	78.661 /	78.651	114.914 / 114.904
Operating pitch diameter (mm)	[dw]	73.200		109.800
(mm)	[dw.e/i]	73.214 /	73.186	109.821 / 109.779
Root diameter (mm)	[df]	66.710		102.963
Generating Profile shift coefficient	[xE.e/i]	-0.3294/	-0.3493	-0.5748/ -0.5948
Manufactured root diameter with xE (mm)	[df.e/i]	66.518 /	66.408	102.771 / 102.661
Theoretical tip clearance (mm)	[c]	0.688		0.688
Effective tip clearance (mm)	[c.e/i]	0.861 /	0.766	0.861 / 0.766
Active root diameter (mm)	[dNf]	70.079		106.254
(mm)	[dNf.e/i]	70.097 /	70.065	106.280 / 106.233
Root form diameter (mm)	[dFf]	70.048		106.081
(mm)	[dFf.e/i]	69.985 /	69.951	105.988 / 105.936
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.073 /	0.040	0.172 / 0.122
Addendum (mm)	[ha=mn*(haP*+x+k)]	1.728		1.053
(mm)	[ha.e/i]	1.728 /	1.723	1.053 / 1.048
Dedendum (mm)	[hf=mn*(hfP*-x)]	4.247		4.922
(mm)	[hf.e/i]	4.343 /	4.398	5.018 / 5.073
Roll angle at dFa (°)	[xsi_dFa.e/i]	30.438 /	30.421	26.640 / 26.627
Roll angle to dNa (°)	[xsi_dNa.e/i]	30.438 /	30.421	26.640 / 26.627
Roll angle to dNf (°)	[xsi_dNf.e/i]	7.734 /	7.531	11.503 / 11.369
Roll angle at dFf (°)	[xsi_dFf.e/i]	7.008 /	6.772	10.655 / 10.498
Tooth height (mm)	[h]	5.975		5.976
Virtual gear no. of teeth	[zn]	34.309		51.464
Normal tooth thickness at tip circle (mm)	[san]	2.404		2.467
(mm)	[san.e/i]	2.336 /	2.290	2.400 / 2.356
Normal-tooth thickness on tip form circle (mm)	[sFan]	2.404		2.467
(mm)	[sFan.e/i]	2.336 /	2.290	2.400 / 2.356
Normal space width at root circle (mm)	[efn]	0.000		0.000
(mm)	[efn.e/i]	0.000 /	0.000	0.000 / 0.000
Max. sliding velocity at tip (m/s)	[vga]	1.448		1.449
Specific sliding at the tip	[zetaa]	0.624		0.714
Specific sliding at the root	[zetaf]	-2.495		-1.663
Mean specific sliding	[zetam]		0.669	
Sliding factor on tip	[Kga]	0.315		0.315
Sliding factor on root	[Kgf]	-0.315		-0.315
Pitch on reference circle (mm)	[pt]		9.844	
Base pitch (mm)	[pbt]		9.093	
Transverse pitch on contact-path (mm)	[pet]		9.093	
Lead height (mm)	[pz]	432.524		648.786
Axial pitch (mm)	[px]		18.022	
Length of path of contact (mm)	[ga, e/i]	13.831 (	13.887 /	13.753)

Length T1-A, T2-A (mm)	[T1A, T2A]	4.621(	4.565/	4.688)	24.225(	24.225/	24.213)
Length T1-B (mm)	[T1B, T2B]	9.359(	9.359/	9.348)	19.487(	19.431/	19.553)
Length T1-C (mm)	[T1C, T2C]	11.538(	11.516/	11.560)	17.307(	17.274/	17.341)
Length T1-D (mm)	[T1D, T2D]	13.714(	13.659/	13.782)	15.132(	15.132/	15.120)
Length T1-E (mm)	[T1E, T2E]	18.452(	18.452/	18.441)	10.394(	10.338/	10.460)
Length T1-T2 (mm)	[T1T2]		28.846 (	28.790 /	28.901)		
Diameter of single contact point B (mm)	[d-B]	71.945(	71.945/	71.939)	111.251(	111.212/	111.298)
Diameter of single contact point D (mm)	[d-D]	74.686(	74.645/	74.736)	108.507(	108.507/	108.500)
Addendum contact ratio	[eps]	0.760(	0.763/	0.757)	0.761(	0.764/	0.756)
Minimal length of contact line (mm)	[Lmin]			61.407			
Transverse contact ratio	[eps_a]			1.521			
Transverse contact ratio with allowances	[eps_a.e/m/i]		1.527 /	1.520 /	1.512		
Overlap ratio	[eps_b]			2.000			
Total contact ratio	[eps_g]			3.521			
Total contact ratio with allowances	[eps_g.e/m/i]		3.527 /	3.520 /	3.512		

2. FACTORS OF GENERAL INFLUENCE

		----- GEAR 1 -----	GEAR 2 --
Nominal circum. force at pitch circle (N)	[Ft]		12706.1
Axial force (N)	[Fa]		6940.6
Radial force (N)	[Fr]		5269.6
Normal force (N)	[Fnorm]		15407.4
Nominal circumferential force per mm (N/mm)	[w]		352.52
Only as information: Forces at operating pitch circle:			
Nominal circumferential force (N)	[Ftw]		13054.1
Axial force (N)	[Faw]		6940.6
Radial force (N)	[Frw]		4336.5
Circumferential speed reference circle (m/s)	[v]		4.73
Circumferential speed operating pitch circle (m/s)	[v(dw)]		4.60
Running-in value (μm)	[yp]		0.6
Running-in value (μm)	[yf]		0.6
Correction coefficient	[CM]		0.800
Gear body coefficient	[CR]		1.000
Basic rack factor	[CBS]		0.975
Material coefficient	[E/Est]		1.000
Singular tooth stiffness (N/mm/μm)	[c']		10.661
Meshing stiffness (N/mm/μm)	[cgalf]		14.826
Meshing stiffness (N/mm/μm)	[cgbet]		12.602
Reduced mass (kg/mm)	[mRed]		0.01230
Resonance speed (min-1)	[nE1]		13814
Resonance ratio (-)	[N]		0.087
Subcritical range			
Running-in value (μm)	[ya]		0.6
Bearing distance l of pinion shaft (mm)	[l]		72.100
Distance s of pinion shaft (mm)	[s]		7.210
Outside diameter of pinion shaft (mm)	[dsh]		36.050
Load in accordance with Figure 13, ISO 6336-1:2006 0:a), 1:b), 2:c), 3:d), 4:e)	[-]	4	
Coefficient K' according to Figure 13, ISO 6336-1:2006	[K']	-1.00	
Without support effect			
Tooth trace deviation (active) (μm)	[Fby]		3.61
from deformation of shaft (μm)	[fsh*B1]		3.16
(fsh (μm) = 3.16, B1= 1.00, fHb5 (μm) = 6.00)			

Tooth without tooth trace modification			
Position of Contact pattern: favorable			
from production tolerances (μm)	[fma*B2]	12.02	
(B2= 1.00)			
Tooth trace deviation, theoretical (μm)	[Fbx]	4.25	
Running-in value (μm)	[yb]	0.64	
Dynamic factor	[KV]	1.012	
Face load factor - flank	[KHb]	1.051	
- Tooth root	[KFb]	1.042	
- Scuffing	[KBb]	1.051	
Transverse load factor - flank	[KHa]	1.020	
- Tooth root	[KF _a]	1.020	
- Scuffing	[KB _a]	1.020	
Helical load factor scuffing	[Kbg]	1.300	
Number of load cycles (in mio.)	[NL]	1440.000	960.000

3. TOOTH ROOT STRENGTH

Calculation of Tooth form coefficients according method: B

		----- GEAR 1 -----	GEAR 2 --
Calculated with profile shift	[x]	-0.2944	-0.5398
Tooth form factor	[YF]	1.37	1.38
Stress correction factor	[YS]	1.93	1.87
Working angle (°)	[alfFen]	14.12	14.90
Bending moment arm (mm)	[hF]	2.36	2.48
Tooth thickness at root (mm)	[sFn]	5.42	5.51
Tooth root radius (mm)	[roF]	1.55	1.65
(hF* = 0.857/ 0.903 sFn* = 1.969/ 2.005 roF* = 0.563/ 0.599)			
(den (mm) =			
92.551/ 138.736 dsFn(mm) = 67.780/ 104.082 alfsFn(°) = 30.00/ 30.00 qs = 1.750/ 1.673)			
Helix angle factor	[Ybet]	0.761	
Deep tooth factor	[YDT]	1.000	
Gear rim factor	[YB]	1.00	1.00
Effective facewidth (mm)	[beff]	36.04	36.04
Nominal stress at tooth root (N/mm²)	[sigF0]	257.68	253.02
Tooth root stress (N/mm²)	[sigF]	346.53	340.27
Permissible bending stress at root of Test-gear			
Notch sensitivity factor	[YdreIT]	0.992	0.992
Surface factor	[YRrelT]	0.957	0.957
size factor (Tooth root)	[YX]	1.000	1.000
Finite life factor	[YNT]	0.884	0.891
	[YdreIT*YRrelT*YX*YNT]	0.839	0.845
Alternating bending factor (mean stress influence coefficient)	[YM]	1.000	1.000
Stress correction factor	[Yst]	2.00	
Yst*sigFlim (N/mm²)	[sigFE]	860.00	860.00
Permissible tooth root stress (N/mm²)	[sigFP=sigFG/SFmin]	515.39	519.15
Limit strength tooth root (N/mm²)	[sigFG]	721.55	726.81
Required safety	[SFmin]	1.40	1.40
Safety for tooth root stress	[SF=sigFG/sigF]	2.08	2.14

Transmittable power (kW)	[kWRating]	89.30	91.60
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4. SAFETY AGAINST PITTING (TOOTH FLANK)

		----- GEAR 1 -----	GEAR 2 --
Zone factor	[ZH]		2.510
Elasticity factor ($\sqrt{N/mm^2}$)	[ZE]		189.812
Contact ratio factor	[Zeps]		0.811
Helix angle factor	[Zbet]		1.067
Effective facewidth (mm)	[beff]		36.04
Nominal contact stress (N/mm ²)	[sigH0]		1152.54
Contact stress at operating pitch circle (N/mm ²)	[sigHw]		1342.06
Single tooth contact factor	[ZB,ZD]	1.00	1.00
Contact stress (N/mm ²)	[sigHB, sigHD]	1342.06	1342.06
Lubrication coefficient at NL	[ZL]	1.020	1.020
Speed coefficient at NL	[ZV]	0.981	0.981
Roughness coefficient at NL	[ZR]	1.005	1.005
Material pairing coefficient at NL	[ZW]	1.000	1.000
Finite life factor	[ZNT]	0.902	0.913
	[ZL * ZV * ZR * ZNT]	0.907	0.918
Limited pitting is permitted:	No		
Size factor (flank)	[ZX]	1.000	1.000
Permissible contact stress (N/mm ²)	[sigHP=sigHG/SHmin]	1360.24	1377.26
Pitting stress limit (N/mm ²)	[sigHG]	1360.24	1377.26
Required safety	[SHmin]	1.00	1.00
Safety factor for contact stress at operating pitch circle			
	[SHw]	1.01	1.03
Safety for stress at single tooth contact	[SHBD=sigHG/sigHBD]	1.01	1.03
(Safety regarding transmittable torque)	[(SHBD)^2]	1.03	1.05
Transmittable power (kW)	[kWRating]	61.68	63.23

4b. MICROPITTING ACCORDING TO ISO/TR 15144-1:2014

Calculation did not run. (Lubricant: Load stage micropitting test is unknown.)

5. SCUFFING LOAD CAPACITY

Calculation method according to ISO TR 13989:2000

Lubrication coefficient (for lubrication type)	[XS]	1.000	
Scuffing test and load stage	[FZGtest]	FZG - Test A / 8.3 / 90 (ISO 14635 - 1)	12
Multiple meshing factor	[Xmp]	1.000	
Relative structure coefficient (Scuffing)	[XWrelT]	1.000	
Thermal contact factor (N/mm/s ^{0.5} /K)	[BM]	13.780	13.780
Relevant tip relief (μm)	[Ca]	2.00	2.00
Optimal tip relief (μm)	[Ceff]	29.72	
Ca taken as optimal in the calculation (0=no, 1=yes)		0	0
Effective facewidth (mm)	[beff]	36.044	
Applicable circumferential force/facewidth (N/mm)	[wBt]	477.987	
Kbg =	1.300, wBt*Kbg =	621.383	

Angle factor ($\epsilon_1:0.760, \epsilon_2:0.761$)	[Xalfbet]	0.930
Flash temperature-criteria		
Lubricant factor	[XL]	0.830
Tooth mass temperature (°C) ($theMi = theoil + XS \cdot 0.47 \cdot Xmp \cdot theflm$)	[theMi]	84.17
Average flash temperature (°C)	[theflm]	30.16
Scuffing temperature (°C)	[theS]	348.80
Coordinate gamma (point of highest temp.) [Gamma.A]=-0.600 [Gamma.E]=0.599	[Gamma]	-0.600
Highest contact temp. (°C)	[theB]	163.99
Flash factor ($^{\circ}K \cdot N^{\wedge}-.75 \cdot s^{\wedge}.5 \cdot m^{\wedge}-.5 \cdot mm$)	[XM]	50.058
Approach factor	[XJ]	1.069
Load sharing factor	[XGam]	0.855
Dynamic viscosity (mPa*s)	[etaM]	41.90 (70.0 °C)
Coefficient of friction	[mym]	0.063
Required safety	[SBmin]	2.000
Safety factor for scuffing (flash temperature)	[SB]	2.966
Integral temperature-criteria		
Lubricant factor	[XL]	1.000
Tooth mass temperature (°C) ($theMC = theoil + XS \cdot 0.70 \cdot theflaint$)	[theMC]	89.37
Mean flash temperature (°C)	[theflaint]	27.67
Integral scuffing temperature (°C)	[theSint]	360.78
Flash factor ($^{\circ}K \cdot N^{\wedge}-.75 \cdot s^{\wedge}.5 \cdot m^{\wedge}-.5 \cdot mm$)	[XM]	50.058
Running-in factor (well run in)	[XE]	1.000
Contact ratio factor	[Xeps]	0.280
Dynamic viscosity (mPa*s)	[etaOil]	41.90 (70.0 °C)
Mean coefficient of friction	[mym]	0.074
Geometry factor	[XBE]	0.360
Meshing factor	[XQ]	1.000
Tip relief factor	[XCa]	1.093
Integral tooth flank temperature (°C)	[theint]	130.87
Required safety	[SSmin]	1.800
Safety factor for scuffing (intg.-temp.)	[SSint]	2.757
Safety referring to transmittable torque	[SSL]	4.777

6. MEASUREMENTS FOR TOOTH THICKNESS

		----- Gear 1 ----- Gear 2 --	
		DIN 3967 cd25	DIN 3967 cd25
Tooth thickness deviation		-0.070 / -0.110	-0.070 / -0.110
Tooth thickness allowance (normal section) (mm)	[As.e/i]		
Number of teeth spanned	[k]	4.000	5.000
Base tangent length (no backlash) (mm)	[Wk]	29.199	37.526
Actual base tangent length ('span') (mm)	[Wk.e/i]	29.134 / 29.096	37.460 / 37.422
(mm)	[ΔWk.e/i]	-0.066 / -0.103	-0.066 / -0.103
Diameter of measuring circle (mm)	[dMWk.m]	74.171	109.431
Theoretical diameter of ball/pin (mm)	[DM]	4.554	4.548
Effective diameter of ball/pin (mm)	[DMeff]	5.000	5.000
Radial single-ball measurement backlash free (mm)	[MrK]	40.646	58.794
Radial single-ball measurement (mm)	[MrK.e/i]	40.557 / 40.505	58.696 / 58.640
Diameter of measuring circle (mm)	[dMMr.m]	74.355	110.745

Diametral measurement over two balls without clearance (mm)	[MdK]	81.292	117.587
Diametral two ball measure (mm)	[MdK.e/i]	81.113 / 81.010	117.392 / 117.279
Diametral measurement over pins without clearance (mm)	[MdR]	81.292	117.587
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	81.113 / 81.010	117.392 / 117.279
Measurement over 3 pins (axial) according to AGMA 2002 (mm)	[dk3A.e/i]	81.113 / 81.010	117.392 / 117.279
Chordal tooth thickness (no backlash) (mm)	[sc]	3.729	3.239
Actual chordal tooth thickness (mm)	[sc.e/i]	3.659 / 3.619	3.169 / 3.129
Reference chordal height from da.m (mm)	[ha]	1.762	1.069
Tooth thickness (Arc) (mm)	[sn]	3.730	3.239
(mm)	[sn.e/i]	3.660 / 3.620	3.169 / 3.129
Backlash free center distance (mm)	[aControl.e/i]	91.263 / 91.126	
Backlash free center distance, allowances (mm)	[jta]	-0.237 / -0.374	
dNf.i with aControl (mm)	[dNf0.i]	69.801	105.807
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	-0.092	-0.090
Tip clearance (mm)	[c0.i(aControl)]	0.410	0.410
Centre distance allowances (mm)	[Aa.e/i]	0.018 / -0.018	
Circumferential backlash from Aa (mm)	[jtw_Aa.e/i]	0.012 / -0.012	
Radial clearance (mm)	[jrw]	0.392 / 0.219	
Circumferential backlash (transverse section) (mm)	[jtw]	0.256 / 0.144	
Normal backlash (mm)	[jnw]	0.211 / 0.118	
Torsional angle at entry with fixed output:			
Entire torsional angle (°)	[j.tSys]	0.4002 / 0.2249	

7. GEAR ACCURACY

		----- GEAR 1 -----	GEAR 2 --
According to ISO 1328-1:1995, ISO 1328-2:1997			
Accuracy grade	[Q]	6	6
Single pitch deviation (μm)	[fptT]	8.50	8.50
Base circle pitch deviation (μm)	[fpbT]	7.90	7.90
Sector pitch deviation over k/8 pitches (μm)	[Fpk/8T]	13.00	15.00
Profile form deviation (μm)	[ffaT]	8.50	8.50
Profile slope deviation (μm)	[fHaT]	7.00	7.00
Total profile deviation (μm)	[FaT]	11.00	11.00
Helix form deviation (μm)	[ffbT]	8.50	8.50
Helix slope deviation (μm)	[fHbT]	8.50	8.50
Total helix deviation (μm)	[FbT]	12.00	12.00
Total cumulative pitch deviation (μm)	[FpT]	27.00	27.00
Runout (μm)	[FrT]	21.00	21.00
Single flank composite, total (μm)	[FisT]	38.00	38.00
Single flank composite, tooth-to-tooth (μm)	[fisT]	11.00	11.00
Radial composite, total (μm)	[FidT]	36.00	36.00
Radial composite, tooth-to-tooth (μm)	[fidT]	14.00	14.00

Axis alignment tolerances (recommendation acc. to ISO TR 10064-3:1996, Quality)

6)

Maximum value for deviation error of axis (μm)	[fSigbet]	12.00 (Fb= 12.00)
Maximum value for inclination error of axes (μm)	[fSigdel]	24.00

8. ADDITIONAL DATA

Maximal possible centre distance (eps_a=1.0)	[aMAX]	93.102	
Mass (kg)	[m]	1.171	2.631
Total mass (kg)	[m]	3.802	
Moment of inertia (system with reference to the drive): calculation without consideration of the exact tooth shape			
single gears ((da+df)/2...di) (kg*m²)	[TraeghMom]	0.0007734	0.003902
System ((da+df)/2...di) (kg*m²)	[TraeghMom]	0.004676	
Torsional stiffness at entry with driven force fixed:			
Torsional stiffness (MNm/rad)	[cr]	0.563	
Torsion when subjected to nominal torque (°)	[delcr]	0.049	
Mean coeff. of friction (acc. Niemann)	[mum]	0.068	
Wear sliding coef. by Niemann	[zetw]	1.018	
Gear power loss (kW)	[PVZ]	0.633	
(Meshing efficiency (%))	[etaz]	98.945)	
Sound pressure level (according to Masuda)	[dB(A)]	72.4	

9. MODIFICATIONS AND TOOTH FORM DEFINITION

Data for the tooth form calculation :

Data not available.

10. SERVICE LIFE, DAMAGE

Required safety for tooth root	[SFmin]	1.40
Required safety for tooth flank	[SHmin]	1.00

Service life (calculated with required safeties):

System service life (h)	[Hatt]	31020
-------------------------	--------	-------

Tooth root service life (h)	[HFatt]	1e+006	1e+006
Tooth flank service life (h)	[HHatt]	3.102e+004	4.653e+004

Note: The entry 1e+006 h means that the Service life > 1,000,000 h.

Damage calculated on the basis of the required service life [H] (20000.0 h)

F1%	F2%	H1%	H2%
0.00	0.00	64.48	42.98

Damage calculated on basis of system service life [Hatt] (31019.6 h)

F1%	F2%	H1%	H2%
0.00	0.00	100.00	66.67

Calculation of the factors required to define reliability R(t) according to B. Bertsche with Weibull distribution:

$$R(t) = 100 * \text{Exp}(-((t^{\text{fac}} - t_0)/(T - t_0))^b) \% ; t (h)$$

Gear		fac	b	t0	T	R(H)%
1	Tooth root	72000	1.7	9.654e+029	1.484e+030	100.00
1	Tooth flank	72000	1.3	2.013e+009	9.592e+009	100.00
2	Tooth root	48000	1.7	9.654e+029	1.484e+030	100.00
2	Tooth flank	48000	1.3	2.013e+009	9.592e+009	100.00

Reliability of the configuration for required service life (%) 100.00 (Bertsche)

REMARKS:

- Specifications with [e/i] imply: Maximum [e] and Minimal value [i] with consideration of all tolerances
- Specifications with [m] imply: Mean value within tolerance

- For the backlash tolerance, the center distance tolerances and the tooth thickness deviation are taken into account. Shown is the maximal and the minimal backlash corresponding the largest resp. the smallest allowances
The calculation is done for the operating pitch circle.
- Calculation of Zbet according Corrigendum 1 ISO 6336-2:2008 with $Z_{bet} = 1/(\cos(\beta))^0.5$
- Details of calculation method:
 - cg according to method B
 - KV according to method B
 - KHb, KFb according method C
 - fma following equation (64), fsh following (57/58), Fbx following (52/53/57)
 - KHa, KFa according to method B
- The logarithmically interpolated value taken from the values for the fatigue strength and the static strength, based on the number of load cycles, is used for coefficients ZL, ZV, ZR, ZW, ZX, YdreIT, YRrelT and YX..

End of Report	lines:	526
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B.6 C14

Name : Unnamed

Changed by: up201306219

on: 21.04.2018

at: 12:04:29

Important hint: At least one warning has occurred during the calculation:

1-> Gear 1: The specific sliding at the root [zeta_f] is less than -3.00.

2-> Gear 1:

Check that roughness values (Flank), Ra value and Rz value lie outside the usual range specified in DIN 4768

CALCULATION OF A CYLINDRICAL SPUR GEAR PAIR

Drawing or article number:

Gear 1: 0.000.0

Gear 2: 0.000.0

Calculation method ISO 6336:2006 Method B

	----- GEAR 1 -----	GEAR 2 --
Power (kW)	[P]	60.040
Speed (1/min)	[n]	1200.0 800.0
Torque (Nm)	[T]	477.8 716.7
Application factor	[KA]	1.25
Required service life (h)	[H]	20000.00
Gear driving (+) / driven (-)	+	-
Working flank gear 1: Right flank		
Sense of rotation gear 1 clockwise		

1. TOOTH GEOMETRY AND MATERIAL

(geometry calculation according to ISO 21771:2007, DIN ISO 21771)

	----- GEAR 1 -----	GEAR 2 --
Center distance (mm)	[a]	91.500
Centre distance tolerance	ISO 286:2010 Measure js7	
Normal module (mm)	[mn]	4.5000
Pressure angle at normal section (°)	[alfn]	20.0000
Helix angle at reference circle (°)	[beta]	0.0000
Number of teeth	[z]	16 24
Facewidth (mm)	[b]	14.00 14.00
Hand of gear		Spur gear
Accuracy grade	[Q-ISO 1328:1995]	6 6
Inner diameter (mm)	[di]	0.00 0.00
Inner diameter of gear rim (mm)	[dbi]	0.00 0.00

Material

Gear 1: 16 MnCr 5 (1), Case-carburized steel, case-hardened
ISO 6336-5 Figure 9/10 (MQ), Core hardness >=25HRC Jominy J=12mm<HRC28

Gear 2: 16 MnCr 5 (1), Case-carburized steel, case-hardened
ISO 6336-5 Figure 9/10 (MQ), Core hardness >=25HRC Jominy J=12mm<HRC28

----- GEAR 1 ----- GEAR 2 --

Surface hardness		HRC 59	HRC 59
Material quality according to ISO 6336:2006 Normal (Life factors ZNT and YNT ≥ 0.85)			
Fatigue strength, tooth root stress (N/mm ²)	[σ_{Flim}]	430.00	430.00
Fatigue strength for Hertzian pressure (N/mm ²)	[σ_{Hlim}]	1500.00	1500.00
Tensile strength (N/mm ²)	[σ_B]	1000.00	1000.00
Yield point (N/mm ²)	[σ_S]	695.00	695.00
Young's modulus (N/mm ²)	[E]	206000	206000
Poisson's ratio	[ν]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.70	0.70
Roughness average value DS, root (μm)	[RAF]	3.00	3.00
Mean roughness height, Rz, flank (μm)	[RZH]	2.50	2.50
Mean roughness height, Rz, root (μm)	[RZF]	20.00	20.00

Gear reference profile	1 :		
Reference profile	1.25 / 0.38 / 1.0 ISO 53:1998 Profil A		
Dedendum coefficient	[hfP*]		1.250
Root radius factor	[rhofP*]		0.380 (rhofPmax*=0.472)
Addendum coefficient	[haP*]		1.000
Tip radius factor	[rhoaP*]		0.000
Protuberance height coefficient	[hprP*]		0.000
Protuberance angle	[alfprP]		0.000
Tip form height coefficient	[hFaP*]		0.000
Ramp angle	[alfKP]		0.000
		not topping	

Gear reference profile	2 :		
Reference profile	1.25 / 0.38 / 1.0 ISO 53:1998 Profil A		
Dedendum coefficient	[hfP*]		1.250
Root radius factor	[rhofP*]		0.380 (rhofPmax*=0.472)
Addendum coefficient	[haP*]		1.000
Tip radius factor	[rhoaP*]		0.000
Protuberance height coefficient	[hprP*]		0.000
Protuberance angle	[alfprP]		0.000
Tip form height coefficient	[hFaP*]		0.000
Ramp angle	[alfKP]		0.000
		not topping	

Summary of reference profile gears:

Dedendum reference profile	[hfP*]	1.250	1.250
Tooth root radius Refer. profile	[rofP*]	0.380	0.380
Addendum Reference profile	[haP*]	1.000	1.000
Protuberance height coefficient	[hprP*]	0.000	0.000
Protuberance angle (°)	[alfprP]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[alfKP]	0.000	0.000

Type of profile modification: none (only running-in)

Tip relief (μm)	[Ca]	2.0	2.0
------------------------------	------	-----	-----

Lubrication type Oil bath lubrication

Type of oil Oil: ISO-VG 220

Lubricant base Mineral-oil base

Kinem. viscosity oil at 40 °C (mm ² /s)	[ν_{40}]	220.00	
Kinem. viscosity oil at 100 °C (mm ² /s)	[ν_{100}]	17.50	
Specific density at 15 °C (kg/dm ³)	[ρ_{Oil}]	0.895	
Oil temperature (°C)	[TS]	70.000	

	----- GEAR 1 -----	GEAR 2 --	
Overall transmission ratio	[itot]	-1.500	
Gear ratio	[u]	1.500	
Transverse module (mm)	[mt]	4.500	
Pressure angle at pitch circle (°)	[alft]	20.000	
Working transverse pressure angle (°)	[alfwt]	22.439	
	[alfwt.e/i]	22.465 / 22.412	
Working pressure angle at normal section (°)	[alfwn]	22.439	
Helix angle at operating pitch circle (°)	[betaw]	0.000	
Base helix angle (°)	[betab]	0.000	
Reference centre distance (mm)	[ad]	90.000	
Sum of profile shift coefficients	[Summexi]	0.3532	
Profile shift coefficient	[x]	0.1817	0.1715
Tooth thickness (Arc) (module) (module)	[sn*]	1.7031	1.6956
Tip alteration (mm)	[k*mn]	-0.089	-0.089
Reference diameter (mm)	[d]	72.000	108.000
Base diameter (mm)	[db]	67.658	101.487
Tip diameter (mm)	[da]	82.457	118.365
(mm)	[da.e/i]	82.457 / 82.447	118.365 / 118.355
Tip diameter allowances (mm)	[Ada.e/i]	0.000 / -0.010	0.000 / -0.010
Tip form diameter (mm)	[dFa]	82.457	118.365
(mm)	[dFa.e/i]	82.457 / 82.447	118.365 / 118.355
Active tip diameter (mm)	[dNa]	82.457	118.365
Active tip diameter (mm)	[dNa.e/i]	82.457 / 82.447	118.365 / 118.355
Operating pitch diameter (mm)	[dw]	73.200	109.800
(mm)	[dw.e/i]	73.214 / 73.186	109.821 / 109.779
Root diameter (mm)	[df]	62.385	98.293
Generating Profile shift coefficient	[xE.e/i]	0.1603/ 0.1481	0.1501/ 0.1379
Manufactured root diameter with xE (mm)	[df.e/i]	62.193 / 62.083	98.101 / 97.991
Theoretical tip clearance (mm)	[c]	1.125	1.125
Effective tip clearance (mm)	[c.e/i]	1.298 / 1.203	1.298 / 1.203
Active root diameter (mm)	[dNf]	68.245	103.998
(mm)	[dNf.e/i]	68.260 / 68.233	104.022 / 103.978
Root form diameter (mm)	[dFf]	67.729	102.609
(mm)	[dFf.e/i]	67.705 / 67.694	102.528 / 102.483
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.283 / 0.264	0.770 / 0.725
Addendum (mm)	[ha=mn*(haP*+x+k)]	5.229	5.183
(mm)	[ha.e/i]	5.229 / 5.224	5.183 / 5.178
Dedendum (mm)	[hf=mn*(hfP*-x)]	4.807	4.853
(mm)	[hf.e/i]	4.904 / 4.958	4.949 / 5.004
Roll angle at dFa (°)	[xsi_dFa.e/i]	39.915 / 39.900	34.391 / 34.380
Roll angle to dNa (°)	[xsi_dNa.e/i]	39.915 / 39.900	34.391 / 34.380
Roll angle to dNf (°)	[xsi_dNf.e/i]	7.660 / 7.488	12.886 / 12.773
Roll angle at dFf (°)	[xsi_dFf.e/i]	2.144 / 1.872	8.228 / 8.047
Tooth height (mm)	[h]	10.036	10.036
Virtual gear no. of teeth	[zn]	16.000	24.000
Normal tooth thickness at tip circle (mm)	[san]	2.735	3.067
(mm)	[san.e/i]	2.661 / 2.609	2.996 / 2.947
Normal-tooth thickness on tip form circle (mm)	[sFan]	2.735	3.067
(mm)	[sFan.e/i]	2.661 / 2.609	2.996 / 2.947
Normal space width at root circle (mm)	[efn]	0.000	0.000
(mm)	[efn.e/i]	0.000 / 0.000	0.000 / 0.000
Max. sliding velocity at tip (m/s)	[vga]	2.010	1.990
Specific sliding at the tip	[zetaa]	0.679	0.780
Specific sliding at the root	[zetaf]	-3.546	-2.112

Mean specific sliding	[zetam]	0.729	
Sliding factor on tip	[Kga]	0.437	0.433
Sliding factor on root	[Kgf]	-0.433	-0.437
Pitch on reference circle (mm)	[pt]	14.137	
Base pitch (mm)	[pbt]	13.285	
Transverse pitch on contact-path (mm)	[pet]	13.285	
Length of path of contact (mm)	[ga, e/i]	19.100 (19.146 / 19.036)	
Length T1-A, T2-A (mm)	[T1A, T2A]	4.467(4.421/ 4.523)	30.458(30.458/ 30.448)
Length T1-B (mm)	[T1B, T2B]	10.282(10.282/ 10.274)	24.643(24.597/ 24.697)
Length T1-C (mm)	[T1C, T2C]	13.970(13.952/ 13.988)	20.955(20.928/ 20.983)
Length T1-D (mm)	[T1D, T2D]	17.752(17.706/ 17.807)	17.173(17.173/ 17.164)
Length T1-E (mm)	[T1E, T2E]	23.567(23.567/ 23.558)	11.358(11.312/ 11.413)
Length T1-T2 (mm)	[T1T2]	34.925 (34.879 / 34.971)	
Diameter of single contact point B (mm)	[d-B]	70.714(70.714/ 70.709)	112.821(112.781/ 112.869)
Diameter of single contact point D (mm)	[d-D]	76.407(76.365/ 76.459)	107.141(107.141/ 107.135)
Addendum contact ratio	[eps]	0.722(0.724/ 0.720)	0.715(0.717/ 0.713)
Minimal length of contact line (mm)	[Lmin]	14.000	
Transverse contact ratio	[eps_a]	1.438	
Transverse contact ratio with allowances	[eps_a.e/m/i]	1.441 / 1.437 / 1.433	
Overlap ratio	[eps_b]	0.000	
Total contact ratio	[eps_g]	1.438	
Total contact ratio with allowances	[eps_g.e/m/i]	1.441 / 1.437 / 1.433	

2. FACTORS OF GENERAL INFLUENCE

	----- GEAR 1 -----	GEAR 2 --
Nominal circum. force at pitch circle (N)	[Ft]	13271.7
Axial force (N)	[Fa]	0.0
Radial force (N)	[Fr]	4830.5
Normal force (N)	[Fnorm]	14123.4
Nominal circumferential force per mm (N/mm)	[w]	947.98
Only as information: Forces at operating pitch circle:		
Nominal circumferential force (N)	[Ftw]	13054.1
Axial force (N)	[Faw]	0.0
Radial force (N)	[Frw]	5390.9
Circumferential speed reference circle (m/s)	[v]	4.52
Circumferential speed operating pitch circle (m/s)	[v(dw)]	4.60
Running-in value (μm)	[yp]	0.6
Running-in value (μm)	[yf]	0.8
Correction coefficient	[CM]	0.800
Gear body coefficient	[CR]	1.000
Basic rack factor	[CBS]	0.975
Material coefficient	[E/Est]	1.000
Singular tooth stiffness (N/mm/μm)	[c']	12.305
Meshing stiffness (N/mm/μm)	[cgalf]	16.345
Meshing stiffness (N/mm/μm)	[cgbet]	13.893
Reduced mass (kg/mm)	[mRed]	0.01275
Resonance speed (min-1)	[nE1]	21371
Resonance ratio (-)	[N]	0.056
Subcritical range		
Running-in value (μm)	[ya]	0.6
Bearing distance l of pinion shaft (mm)	[l]	28.000
Distance s of pinion shaft (mm)	[s]	2.800
Outside diameter of pinion shaft (mm)	[dsh]	14.000

Load in accordance with Figure 13, ISO 6336-1:2006	[-]	4	
0:a), 1:b), 2:c), 3:d), 4:e)			
Coefficient K' according to Figure 13, ISO 6336-1:2006	[K']	-1.00	
Without support effect			
Tooth trace deviation (active) (μm)	[Fby]	7.38	
from deformation of shaft (μm)	[fsh*B1]	10.66	
(fsh (μm) = 10.66, B1= 1.00, fHb5 (μm) = 5.50)			
Tooth without tooth trace modification			
Position of Contact pattern: favorable			
from production tolerances (μm)	[fma*B2]	10.61	
(B2= 1.00)			
Tooth trace deviation, theoretical (μm)	[Fbx]	8.68	
Running-in value (μm)	[yb]	1.30	
Dynamic factor	[KV]	1.016	
Face load factor - flank	[KHb]	1.043	
- Tooth root	[KFb]	1.029	
- Scuffing	[KBb]	1.043	
Transverse load factor - flank	[KHa]	1.000	
- Tooth root	[KFα]	1.000	
- Scuffing	[KBα]	1.000	
Helical load factor scuffing	[Kbg]	1.000	
Number of load cycles (in mio.)	[NL]	1440.000	960.000

3. TOOTH ROOT STRENGTH

Calculation of Tooth form coefficients according method: B

		----- GEAR 1 -----	GEAR 2 --
Calculated with profile shift	[x]	0.1817	0.1715
Tooth form factor	[YF]	1.71	1.60
Stress correction factor	[YS]	1.84	1.90
Working angle (°)	[alfFn]	23.11	22.92
Bending moment arm (mm)	[hF]	5.13	5.34
Tooth thickness at root (mm)	[sFn]	8.91	9.40
Tooth root radius (mm)	[roF]	2.33	2.28
(hF* = 1.140/ 1.187 sFn* = 1.980/ 2.089 roF* = 0.519/ 0.507)			
(den (mm) = 76.407/ 112.821 dsFn(mm) = 63.923/ 99.951 alfsFn(°) = 30.00/ 30.00 qs = 1.908/ 2.061)			
Helix angle factor	[Ybet]	1.000	
Deep tooth factor	[YDT]	1.000	
Gear rim factor	[YB]	1.00	1.00
Effective facewidth (mm)	[beff]	14.00	14.00
Nominal stress at tooth root (N/mm²)	[sigF0]	661.97	641.83
Tooth root stress (N/mm²)	[sigF]	865.50	839.15
Permissible bending stress at root of Test-gear			
Notch sensitivity factor	[YdreIT]	0.994	0.996
Surface factor	[YRrelT]	0.957	0.957
size factor (Tooth root)	[YX]	1.000	1.000
Finite life factor	[YNT]	0.884	0.891
	[YdreIT*YRrelT*YX*YNT]	0.840	0.849

Alternating bending factor (mean stress influence coefficient)	[YM]	1.000	1.000
Stress correction factor	[Yst]	2.00	
Yst*sigFlim (N/mm ²)	[sigFE]	860.00	860.00
Permissible tooth root stress (N/mm ²)	[sigFP=sigFG/SFmin]	516.27	521.31
Limit strength tooth root (N/mm ²)	[sigFG]	722.78	729.84
Required safety	[SFmin]	1.40	1.40
Safety for tooth root stress	[SF=sigFG/sigF]	0.84	0.87
Transmittable power (kW)	[kWRating]	35.81	37.30

4. SAFETY AGAINST PITTING (TOOTH FLANK)

		----- GEAR 1 -----	GEAR 2 --
Zone factor	[ZH]		2.342
Elasticity factor ($\sqrt{N/mm^2}$)	[ZE]		189.812
Contact ratio factor	[Zeps]		0.924
Helix angle factor	[Zbet]		1.000
Effective facewidth (mm)	[beff]		14.00
Nominal contact stress (N/mm ²)	[sigH0]		1924.44
Contact stress at operating pitch circle (N/mm ²)	[sigHw]		2214.62
Single tooth contact factor	[ZB,ZD]	1.07	1.00
Contact stress (N/mm ²)	[sigHB, sigHD]	2380.41	2214.62
Lubrication coefficient at NL	[ZL]	1.020	1.020
Speed coefficient at NL	[ZV]	0.980	0.980
Roughness coefficient at NL	[ZR]	1.010	1.010
Material pairing coefficient at NL	[ZW]	1.000	1.000
Finite life factor	[ZNT]	0.902	0.913
	[ZL * ZV * ZR * ZNT]	0.911	0.922
Limited pitting is permitted:	No		
Size factor (flank)	[ZX]	1.000	1.000
Permissible contact stress (N/mm ²)	[sigHP=sigHG/SHmin]	1365.82	1382.92
Pitting stress limit (N/mm ²)	[sigHG]	1365.82	1382.92
Required safety	[SHmin]	1.00	1.00
Safety factor for contact stress at operating pitch circle			
	[SHw]	0.62	0.62
Safety for stress at single tooth contact	[SHBD=sigHG/sigHBD]	0.57	0.62
(Safety regarding transmittable torque)	[(SHBD)^2]	0.33	0.39
Transmittable power (kW)	[kWRating]	19.77	23.41

4b. MICROPITTING ACCORDING TO ISO/TR 15144-1:2014

Calculation did not run. (Lubricant: Load stage micropitting test is unknown.)

5. SCUFFING LOAD CAPACITY

Calculation method according to ISO TR 13989:2000

Lubrication coefficient (for lubrication type)	[XS]	1.000	
Scuffing test and load stage	[FZGtest]	FZG - Test A / 8.3 / 90 (ISO 14635 - 1)	12
Multiple meshing factor	[Xmp]	1.000	
Relative structure coefficient (Scuffing)	[XWrelT]	1.000	

Thermal contact factor (N/mm/s ^{0.5} /K)	[BM]	13.780	13.780
Relevant tip relief (μm)	[Ca]	2.00	2.00
Optimal tip relief (μm)	[Ce _{ff}]	96.30	
Ca taken as optimal in the calculation (0=no, 1=yes)		0	0
Effective facewidth (mm)	[be _{ff}]	14.000	
Applicable circumferential force/facewidth (N/mm)	[wB _t]	1255.423	
K _{bg} = 1.000, wB _t *K _{bg} = 1255.423			
Angle factor (ε ₁ :0.722, ε ₂ :0.715)	[X _{alfbet}]	1.013	

Flash temperature-criteria

Lubricant factor	[XL]	0.830	
Tooth mass temperature (°C) (the _{Mi} = the _{oil} + X _S *0.47*X _{mp} *the _{flm})	[the _{Mi}]	110.45	
Average flash temperature (°C)	[the _{flm}]	86.06	
Scuffing temperature (°C)	[the _S]	348.80	
Coordinate gamma (point of highest temp.) [Gamma.A]=-0.680 [Gamma.E]=0.687	[Gamma]	-0.680	
Highest contact temp. (°C)	[the _B]	264.51	
Flash factor (°K*N ^{-0.75} *s ^{-0.5} *m ^{-0.5})	[X _M]	50.058	
Approach factor	[X _J]	1.232	
Load sharing factor	[X _{Gam}]	0.333	
Dynamic viscosity (mPa*s)	[eta _M]	41.90 (70.0 °C)	
Coefficient of friction	[m _{ym}]	0.085	
Required safety	[S _{Bmin}]	2.000	
Safety factor for scuffing (flash temperature)	[S _B]	1.433	

Integral temperature-criteria

Lubricant factor	[XL]	1.000	
Tooth mass temperature (°C) (the _{MC} = the _{oil} + X _S *0.70*the _{flaint})	[the _{MC}]	126.37	
Mean flash temperature (°C)	[the _{flaint}]	80.53	
Integral scuffing temperature (°C)	[the _{Sint}]	360.78	
Flash factor (°K*N ^{-0.75} *s ^{-0.5} *m ^{-0.5})	[X _M]	50.058	
Running-in factor (well run in)	[X _E]	1.000	
Contact ratio factor	[X _{eps}]	0.297	
Dynamic viscosity (mPa*s)	[eta _{Oil}]	41.90 (70.0 °C)	
Mean coefficient of friction	[m _{ym}]	0.094	
Geometry factor	[X _{BE}]	0.419	
Meshing factor	[X _Q]	1.000	
Tip relief factor	[X _{Ca}]	1.064	
Integral tooth flank temperature (°C)	[the _{int}]	247.17	
Required safety	[S _{Smin}]	1.800	
Safety factor for scuffing (intg.-temp.)	[S _{Sint}]	1.460	
Safety referring to transmittable torque	[S _{SL}]	1.641	

6. MEASUREMENTS FOR TOOTH THICKNESS

		----- Gear 1 -----	Gear 2 --
Tooth thickness deviation		DIN 3967 cd25	DIN 3967 cd25
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.070 / -0.110	-0.070 / -0.110
Number of teeth spanned	[k]	3.000	3.000
Base tangent length (no backlash) (mm)	[W _k]	34.779	35.252
Actual base tangent length ('span') (mm)	[W _{k.e/i}]	34.713 / 34.676	35.186 / 35.149
(mm)	[ΔW _{k.e/i}]	-0.066 / -0.103	-0.066 / -0.103

Diameter of measuring circle (mm)	[dMWk.m]	76.035	107.407
Theoretical diameter of ball/pin (mm)	[DM]	8.300	7.995
Effective diameter of ball/pin (mm)	[DMeff]	9.000	8.000
Radial single-ball measurement backlash free (mm)	[MrK]	43.848	60.403
Radial single-ball measurement (mm)	[MrK.e/i]	43.783 / 43.746	60.328 / 60.285
Diameter of measuring circle (mm)	[dMMr.m]	74.366	109.383
Diametral measurement over two balls without clearance (mm)	[MdK]	87.696	120.807
Diametral two ball measure (mm)	[MdK.e/i]	87.567 / 87.493	120.656 / 120.569
Diametral measurement over pins without clearance (mm)	[MdR]	87.696	120.807
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	87.567 / 87.493	120.656 / 120.569
Measurement over 3 pins (axial) according to AGMA 2002 (mm)	[dk3A.e/i]	87.567 / 87.493	120.656 / 120.569
Chordal tooth thickness (no backlash) (mm)	[sc]	7.649	7.624
Actual chordal tooth thickness (mm)	[sc.e/i]	7.579 / 7.539	7.554 / 7.514
Reference chordal height from da.m (mm)	[ha]	5.430	5.315
Tooth thickness (Arc) (mm)	[sn]	7.664	7.630
(mm)	[sn.e/i]	7.594 / 7.554	7.560 / 7.520
Backlash free center distance (mm)	[aControl.e/i]	91.327 / 91.227	
Backlash free center distance, allowances (mm)	[jta]	-0.173 / -0.273	
dNf.i with aControl (mm)	[dNf0.i]	68.071	103.692
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	0.183	0.582
Tip clearance (mm)	[c0.i(aControl)]	0.948	0.948
Centre distance allowances (mm)	[Aa.e/i]	0.018 / -0.018	
Circumferential backlash from Aa (mm)	[jtw_Aa.e/i]	0.014 / -0.014	
Radial clearance (mm)	[jrw]	0.291 / 0.156	
Circumferential backlash (transverse section) (mm)	[jtw]	0.238 / 0.128	
Normal backlash (mm)	[jnw]	0.224 / 0.120	
Torsional angle at entry with fixed output:			
Entire torsional angle (°)	[j.tSys]	0.3728 / 0.2002	

7. GEAR ACCURACY

----- GEAR 1 ----- GEAR 2 --

According to ISO 1328-1:1995, ISO 1328-2:1997

Accuracy grade	[Q]	6	6
Single pitch deviation (μm)	[fptT]	9.00	9.00
Base circle pitch deviation (μm)	[fpbT]	8.50	8.50
Sector pitch deviation over k/8 pitches (μm)	[Fpk/8T]	14.00	16.00
Profile form deviation (μm)	[ffaT]	10.00	10.00
Profile slope deviation (μm)	[fHaT]	8.50	8.50
Total profile deviation (μm)	[FaT]	13.00	13.00
Helix form deviation (μm)	[ffbT]	7.50	7.50
Helix slope deviation (μm)	[fHbT]	7.50	7.50
Total helix deviation (μm)	[FbT]	11.00	11.00
Total cumulative pitch deviation (μm)	[FpT]	28.00	28.00
Runout (μm)	[FrT]	22.00	22.00
Single flank composite, total (μm)	[FisT]	49.00	49.00
Single flank composite, tooth-to-tooth (μm)	[fisT]	22.00	22.00
Radial composite, total (μm)	[FidT]	44.00	44.00
Radial composite, tooth-to-tooth (μm)	[fidT]	22.00	22.00

Axis alignment tolerances (recommendation acc. to ISO TR 10064-3:1996, Quality)

6)

Maximum value for deviation error of axis (μm)	[fSigbet]	11.00 (Fb= 11.00)
Maximum value for inclination error of axes (μm)	[fSigdel]	22.00

8. ADDITIONAL DATA

Mass (kg)	[m]	0.452	1.010
Total mass (kg)	[m]	1.462	
Moment of inertia (system with reference to the drive): calculation without consideration of the exact tooth shape			
single gears ((da+df)/2...di) (kg*m²)	[TraeghMom]	0.000296	0.001482
System ((da+df)/2...di) (kg*m²)	[TraeghMom]	0.0009547	
Torsional stiffness at entry with driven force fixed:			
Torsional stiffness (MNm/rad)	[cr]	0.219	
Torsion when subjected to nominal torque (°)	[delcr]	0.125	
Mean coeff. of friction (acc. Niemann)	[mum]	0.095	
Wear sliding coef. by Niemann	[zetw]	1.048	
Gear power loss (kW)	[PVZ]	1.109	
(Meshing efficiency (%))	[etaz]	98.152	
Sound pressure level (according to Masuda)	[dB(A)]	77.6	

9. MODIFICATIONS AND TOOTH FORM DEFINITION

Data for the tooth form calculation :

Data not available.

10. SERVICE LIFE, DAMAGE

Required safety for tooth root	[SFmin]	1.40
Required safety for tooth flank	[SHmin]	1.00

Service life (calculated with required safeties):

System service life (h)	[Hatt]	1.231
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Tooth root service life (h)	[HFatt]	1.231	2.731
Tooth flank service life (h)	[HHatt]	1.551	6.163

Damage calculated on the basis of the required service life [H] (20000.0 h)

F1%	F2%	H1%	H2%
9999.99	9999.99	9999.99	9999.99

Damage calculated on basis of system service life [Hatt] (1.2 h)

F1%	F2%	H1%	H2%
100.00	45.06	79.33	19.97

Calculation of the factors required to define reliability R(t) according to B. Bertsche with Weibull distribution:

$$R(t) = 100 * \exp(-((t*fac - t_0)/(T - t_0))^b) \% ; t (h)$$

Gear		fac	b	t0	T	R(H)%
1	Tooth root	72000	1.7	8.554e+004	1.315e+005	0.00
1	Tooth flank	72000	1.3	1.007e+005	4.797e+005	0.00
2	Tooth root	48000	1.7	1.265e+005	1.945e+005	0.00
2	Tooth flank	48000	1.3	2.667e+005	1.27e+006	0.00

Reliability of the configuration for required service life (%) 0.00 (Bertsche)

REMARKS:

- Specifications with [e/i] imply: Maximum [e] and Minimal value [i] with consideration of all tolerances
- Specifications with [m] imply: Mean value within tolerance
- For the backlash tolerance, the center distance tolerances and the tooth thickness deviation are taken into account. Shown is the maximal and the minimal backlash corresponding the largest resp. the smallest allowances
- The calculation is done for the operating pitch circle.
- Details of calculation method:
 - cg according to method B
 - KV according to method B
 - KHb, KFb according method C
 - fma following equation (64), fsh following (57/58), Fbx following (52/53/57)
 - KHa, KFa according to method B
- The logarithmically interpolated value taken from the values for the fatigue strength and the static strength, based on the number of load cycles, is used for coefficients ZL, ZV, ZR, ZW, ZX, YdreIT, YRreIT and YX..

End of Report	lines:	521
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B.7 H951

Name : Unnamed

Changed by: up201306219

on: 23.04.2018

at: 10:14:32

Important hint: At least one warning has occurred during the calculation:

1-> Transverse contact ratio is < 1 !

CALCULATION OF A HELICAL GEAR PAIR

Drawing or article number:

Gear 1: 0.000.0

Gear 2: 0.000.0

Calculation method ISO 6336:2006 Method B

	----- GEAR 1 -----	GEAR 2 --
Power (kW)	[P]	60.040
Speed (1/min)	[n]	1200.0 800.0
Torque (Nm)	[T]	477.8 716.7
Application factor	[KA]	1.25
Required service life (h)	[H]	20000.00
Gear driving (+) / driven (-)	+	-
Working flank gear 1: Right flank		
Sense of rotation gear 1 clockwise		

1. TOOTH GEOMETRY AND MATERIAL

(geometry calculation according to ISO 21771:2007, DIN ISO 21771)

	----- GEAR 1 -----	GEAR 2 --
Center distance (mm)	[a]	91.500
Centre distance tolerance	ISO 286:2010 Measure js7	
Normal module (mm)	[mn]	1.7500
Pressure angle at normal section (°)	[alfn]	20.0000
Helix angle at reference circle (°)	[beta]	15.0000
Number of teeth	[z]	38 57
Facewidth (mm)	[b]	23.00 23.00
Hand of gear	right	left
Accuracy grade	[Q-ISO 1328:1995]	6 6
Inner diameter (mm)	[di]	0.00 0.00
Inner diameter of gear rim (mm)	[dbi]	0.00 0.00

Material

Gear 1: 16 MnCr 5 (1), Case-carburized steel, case-hardened
ISO 6336-5 Figure 9/10 (MQ), Core hardness >=25HRC Jominy J=12mm<HRC28

Gear 2: 16 MnCr 5 (1), Case-carburized steel, case-hardened
ISO 6336-5 Figure 9/10 (MQ), Core hardness >=25HRC Jominy J=12mm<HRC28

	----- GEAR 1 -----	GEAR 2 --
Surface hardness	HRC 59	HRC 59
Material quality according to ISO 6336:2006 Normal (Life factors ZNT and YNT >=0.85)		
Fatigue strength. tooth root stress (N/mm²)	[σFlim]	430.00 430.00

Fatigue strength for Hertzian pressure (N/mm ²)	[σHlim]	1500.00	1500.00
Tensile strength (N/mm ²)	[σB]	1000.00	1000.00
Yield point (N/mm ²)	[σS]	695.00	695.00
Young's modulus (N/mm ²)	[E]	206000	206000
Poisson's ratio	[ν]	0.300	0.300
Roughness average value DS, flank (μm)	[RAH]	0.35	0.35
Roughness average value DS, root (μm)	[RAF]	3.00	3.00
Mean roughness height, Rz, flank (μm)	[RZH]	2.50	2.50
Mean roughness height, Rz, root (μm)	[RZF]	20.00	20.00

Gear reference profile 1 :

Reference profile 1.25 / 0.38 / 1.0 ISO 53:1998 Profil A

Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)	
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[rhoaP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[alfprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[alfKP]	0.000	

not topping

Gear reference profile 2 :

Reference profile 1.25 / 0.38 / 1.0 ISO 53:1998 Profil A

Dedendum coefficient	[hfP*]	1.250	
Root radius factor	[rhofP*]	0.380 (rhofPmax*=0.472)	
Addendum coefficient	[haP*]	1.000	
Tip radius factor	[rhoaP*]	0.000	
Protuberance height coefficient	[hprP*]	0.000	
Protuberance angle	[alfprP]	0.000	
Tip form height coefficient	[hFaP*]	0.000	
Ramp angle	[alfKP]	0.000	

not topping

Summary of reference profile gears:

Dedendum reference profile	[hfP*]	1.250	1.250
Tooth root radius Refer. profile	[rofP*]	0.380	0.380
Addendum Reference profile	[haP*]	1.000	1.000
Protuberance height coefficient	[hprP*]	0.000	0.000
Protuberance angle (°)	[alfprP]	0.000	0.000
Tip form height coefficient	[hFaP*]	0.000	0.000
Ramp angle (°)	[alfKP]	0.000	0.000

Type of profile modification: none (only running-in)

Tip relief (μm)	[Ca]	2.0	2.0
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Lubrication type

Oil bath lubrication

Type of oil

Oil: ISO-VG 220

Lubricant base

Mineral-oil base

Kinem. viscosity oil at 40 °C (mm ² /s)	[nu40]	220.00	
Kinem. viscosity oil at 100 °C (mm ² /s)	[nu100]	17.50	
Specific density at 15 °C (kg/dm ³)	[roOil]	0.895	
Oil temperature (°C)	[TS]	70.000	

----- GEAR 1 ----- GEAR 2 --

Overall transmission ratio	[itot]	-1.500	
----------------------------	--------	--------	--

Gear ratio	[u]	1.500		
Transverse module (mm)	[mt]	1.812		
Pressure angle at pitch circle (°)	[alft]	20.647		
Working transverse pressure angle (°)	[alfwt]	28.344		
	[alfwt.e/i]	28.365 /	28.324	
Working pressure angle at normal section (°)	[alfwn]	27.420		
Helix angle at operating pitch circle (°)	[betaw]	15.902		
Base helix angle (°)	[betab]	14.076		
Reference centre distance (mm)	[ad]	86.057		
Sum of profile shift coefficients	[Summexi]	3.6918		
Profile shift coefficient	[x]	1.6915	2.0003	
Tooth thickness (Arc) (module) (module)	[sn*]	2.8021	3.0269	
Tip alteration (mm)	[k*mn]	-1.018	-1.018	
Reference diameter (mm)	[d]	68.846	103.269	
Base diameter (mm)	[db]	64.424	96.636	
Tip diameter (mm)	[da]	76.230	111.734	
(mm)	[da.e/i]	76.230 /	76.220	111.734 / 111.724
Tip diameter allowances (mm)	[Ada.e/i]	0.000 /	-0.010	0.000 / -0.010
Tip form diameter (mm)	[dFa]	76.230	111.734	
(mm)	[dFa.e/i]	76.230 /	76.220	111.734 / 111.724
Active tip diameter (mm)	[dNa]	76.230	111.734	
Active tip diameter (mm)	[dNa.e/i]	76.230 /	76.220	111.734 / 111.724
Operating pitch diameter (mm)	[dw]	73.200	109.800	
(mm)	[dw.e/i]	73.214 /	73.186	109.821 / 109.779
Root diameter (mm)	[df]	70.391	105.895	
Generating Profile shift coefficient	[xE.e/i]	1.6366/	1.6052	1.9453/ 1.9139
Manufactured root diameter with xE (mm)	[df.e/i]	70.199 /	70.089	105.702 / 105.593
Theoretical tip clearance (mm)	[c]	0.438	0.438	
Effective tip clearance (mm)	[c.e/i]	0.611 /	0.516	0.611 / 0.516
Active root diameter (mm)	[dNf]	71.406	107.083	
(mm)	[dNf.e/i]	71.446 /	71.374	107.123 / 107.051
Root form diameter (mm)	[dFf]	71.555	107.173	
(mm)	[dFf.e/i]	71.319 /	71.186	106.939 / 106.806
Reserve (dNf-dFf)/2 (mm)	[cF.e/i]	0.130 /	0.027	0.159 / 0.056
Addendum (mm)	[ha=mn*(haP*+x+k)]	3.692	4.233	
(mm)	[ha.e/i]	3.692 /	3.687	4.233 / 4.228
Dedendum (mm)	[hf=mn*(hfP*-x)]	-0.773	-1.313	
(mm)	[hf.e/i]	-0.676 /	-0.622	-1.217 / -1.162
Roll angle at dFa (°)	[xsi_dFa.e/i]	36.241 /	36.225	33.255 / 33.243
Roll angle to dNa (°)	[xsi_dNa.e/i]	36.241 /	36.225	33.255 / 33.243
Roll angle to dNf (°)	[xsi_dNf.e/i]	27.471 /	27.322	27.407 / 27.309
Roll angle at dFf (°)	[xsi_dFf.e/i]	27.209 /	26.932	27.153 / 26.969
Tooth height (mm)	[h]	2.919	2.919	
Virtual gear no. of teeth	[zn]	41.814	62.721	
Normal tooth thickness at tip circle (mm)	[san]	1.583	1.614	
(mm)	[san.e/i]	1.512 /	1.462	1.544 / 1.495
Normal-tooth thickness on tip form circle (mm)	[sFan]	1.583	1.614	
(mm)	[sFan.e/i]	1.512 /	1.462	1.544 / 1.495
Normal space width at root circle (mm)	[efn]	1.223	1.265	
(mm)	[efn.e/i]	1.210 /	1.204	1.252 / 1.245
Max. sliding velocity at tip (m/s)	[vga]	0.628	0.415	
Specific sliding at the tip	[zetaa]	0.245	0.176	
Specific sliding at the root	[zetaf]	-0.214	-0.325	
Mean specific sliding	[zetam]	0.218		
Sliding factor on tip	[Kga]	0.137	0.090	
Sliding factor on root	[Kgf]	-0.090	-0.137	

Pitch on reference circle (mm)	[pt]	5.692		
Base pitch (mm)	[pbt]	5.326		
Transverse pitch on contact-path (mm)	[pet]	5.326		
Lead height (mm)	[pz]	807.189		1210.784
Axial pitch (mm)	[px]	21.242		
Length of path of contact (mm)	[ga, e/i]	4.978 (	5.015 /	4.922)
Length T1-A, T2-A (mm)	[T1A, T2A]	15.397(	15.360/	15.444) 28.044(28.044/ 28.034)
Length T1-B (mm)	[T1B, T2B]	15.397(	15.360/	15.444) 28.044(28.044/ 28.034)
Length T1-C (mm)	[T1C, T2C]	17.377(	17.362/	17.391) 26.065(26.043/ 26.087)
Length T1-D (mm)	[T1D, T2D]	20.375(	20.375/	20.366) 23.066(23.030/ 23.113)
Length T1-E (mm)	[T1E, T2E]	20.375(	20.375/	20.366) 23.066(23.030/ 23.113)
Length T1-T2 (mm)	[T1T2]	43.442 (	43.405 /	43.478)
Diameter of single contact point B (mm)	[d-B]	71.406(	71.374/	71.446) 111.734(111.734/ 111.724)
Diameter of single contact point D (mm)	[d-D]	76.230(	76.230/	76.220) 107.083(107.051/ 107.123)
Addendum contact ratio	[eps]	0.563(	0.566/	0.558) 0.372(0.376/ 0.366)
Minimal length of contact line (mm)	[Lmin]	20.848		
Transverse contact ratio	[eps_a]	0.935		
Transverse contact ratio with allowances	[eps_a.e/m/i]	0.942 /	0.933 /	0.924
Overlap ratio	[eps_b]	1.083		
Total contact ratio	[eps_g]	2.017		
Total contact ratio with allowances	[eps_g.e/m/i]	2.024 /	2.016 /	2.007

2. FACTORS OF GENERAL INFLUENCE

		----- GEAR 1 ----- GEAR 2 --
Nominal circum. force at pitch circle (N)	[Ft]	13879.7
Axial force (N)	[Fa]	3719.1
Radial force (N)	[Fr]	5230.0
Normal force (N)	[Fnorm]	15291.5
Nominal circumferential force per mm (N/mm)	[w]	603.47
Only as information: Forces at operating pitch circle:		
Nominal circumferential force (N)	[Ftw]	13054.1
Axial force (N)	[Faw]	3719.1
Radial force (N)	[Frw]	7042.0
Circumferential speed reference circle (m/s)	[v]	4.33
Circumferential speed operating pitch circle (m/s)	[v(dw)]	4.60
Running-in value (μm)	[yp]	0.5
Running-in value (μm)	[yf]	0.5
Correction coefficient	[CM]	0.800
Gear body coefficient	[CR]	1.000
Basic rack factor	[CBS]	0.975
Material coefficient	[E/Est]	1.000
Singular tooth stiffness (N/mm/μm)	[c']	14.934
Meshing stiffness (N/mm/μm)	[cgalf]	14.934
Meshing stiffness (N/mm/μm)	[cgbet]	12.694
Reduced mass (kg/mm)	[mRed]	0.01462
Resonance speed (min-1)	[nE1]	8031
Resonance ratio (-)	[N]	0.149
Subcritical range		
Running-in value (μm)	[ya]	0.5
Bearing distance l of pinion shaft (mm)	[l]	46.000
Distance s of pinion shaft (mm)	[s]	4.600
Outside diameter of pinion shaft (mm)	[dsh]	23.000
Load in accordance with Figure 13, ISO 6336-1:2006	[-]	4

0:a), 1:b), 2:c), 3:d), 4:e)

Coefficient K' according to Figure 13, ISO 6336-1:2006	[K']	-1.00	
Without support effect			
Tooth trace deviation (active) (μm)	[Fby]	3.61	
from deformation of shaft (μm)	[fsh*B1]	6.43	
(fsh (μm) = 6.43, B1= 1.00, fHb5 (μm) = 6.00)			
Tooth without tooth trace modification			
Position of Contact pattern: favorable			
from production tolerances (μm)	[fma*B2]	12.02	
(B2= 1.00)			
Tooth trace deviation, theoretical (μm)	[Fbx]	4.25	
Running-in value (μm)	[yb]	0.64	
Dynamic factor			
	[KV]	1.043	
Face load factor - flank			
	[KHb]	1.029	
- Tooth root	[KFb]	1.025	
- Scuffing	[Kbb]	1.029	
Transverse load factor - flank			
	[KHα]	1.000	
- Tooth root	[KFα]	1.000	
- Scuffing	[KBα]	1.000	
Helical load factor scuffing			
	[Kbg]	1.046	
Number of load cycles (in mio.)			
	[NL]	1440.000	960.000

3. TOOTH ROOT STRENGTH

Calculation of Tooth form coefficients according method: B

		----- GEAR 1 -----	GEAR 2 --
Calculated with manufacturing profile shift	[xE.e]	1.6366	1.9453
Tooth form factor	[YF]	1.28	1.32
Stress correction factor	[YS]	1.86	1.72
Working angle (°)	[alfFen]	30.37	28.54
Bending moment arm (mm)	[hF]	2.28	2.24
Tooth thickness at root (mm)	[sFn]	4.15	4.07
Tooth root radius (mm)	[roF]	1.10	1.31
(hF* = 1.300/ 1.277 sFn* = 2.370/ 2.326 roF* = 0.629/ 0.749)			
(den (mm) = 80.558/ 118.226 dsFn(mm) = 70.936/ 106.524 alfsFn(°) = 30.00/ 30.00 qs = 1.884/ 1.552)			
Helix angle factor			
	[Ybet]	0.875	
Deep tooth factor			
	[YDT]	1.000	
Gear rim factor			
	[YB]	1.00	1.00
Effective facewidth (mm)			
	[beff]	23.00	23.00
Nominal stress at tooth root (N/mm²)			
	[sigF0]	714.51	685.76
Tooth root stress (N/mm²)			
	[sigF]	955.46	917.02
Permissible bending stress at root of Test-gear			
Notch sensitivity factor			
	[YdreIT]	0.994	0.990
Surface factor			
	[YRrelT]	0.957	0.957
size factor (Tooth root)			
	[YX]	1.000	1.000
Finite life factor			
	[YNT]	0.884	0.891
	[YdreIT*YRrelT*YX*YNT]	0.840	0.844
Alternating bending factor (mean stress influence coefficient)			
	[YM]	1.000	1.000

Stress correction factor	[Yst]	2.00	
Yst*sigFlim (N/mm ²)	[sigFE]	860.00	860.00
Permissible tooth root stress (N/mm ²)	[sigFP=sigFG/SFmin]	535.26	537.64
Limit strength tooth root (N/mm ²)	[sigFG]	722.60	725.82
Required safety	[SFmin]	1.35	1.35
Safety for tooth root stress	[SF=sigFG/sigF]	0.76	0.79
Transmittable power (kW)	[kWRating]	33.63	35.20

4. SAFETY AGAINST PITTING (TOOTH FLANK)

		----- GEAR 1 -----	GEAR 2 --
Zone factor	[ZH]		2.027
Elasticity factor ($\sqrt{N/mm^2}$)	[ZE]		189.812
Contact ratio factor	[Zeps]		1.011
Helix angle factor	[Zbet]		1.017
Effective facewidth (mm)	[beff]		23.00
Nominal contact stress (N/mm ²)	[sigH0]		1512.16
Contact stress at operating pitch circle (N/mm ²)	[sigHw]		1751.91
Single tooth contact factor	[ZB,ZD]	1.00	1.00
Contact stress (N/mm ²)	[sigHB, sigHD]	1751.91	1751.91
Lubrication coefficient at NL	[ZL]	1.020	1.020
Speed coefficient at NL	[ZV]	0.979	0.979
Roughness coefficient at NL	[ZR]	1.016	1.016
Material pairing coefficient at NL	[ZW]	1.000	1.000
Finite life factor	[ZNT]	0.902	0.913
	[ZL * ZV * ZR * ZNT]	0.915	0.926
Limited pitting is permitted:	No		
Size factor (flank)	[ZX]	1.000	1.000
Permissible contact stress (N/mm ²)	[sigHP=sigHG/SHmin]	1407.59	1425.20
Pitting stress limit (N/mm ²)	[sigHG]	1372.40	1389.57
Required safety	[SHmin]	0.97	0.97
Safety factor for contact stress at operating pitch circle			
	[SHw]	0.78	0.79
Safety for stress at single tooth contact	[SHBD=sigHG/sigHBD]	0.78	0.79
(Safety regarding transmittable torque)	[(SHBD)^2]	0.61	0.63
Transmittable power (kW)	[kWRating]	38.76	39.73

4b. MICROPITTING ACCORDING TO ISO/TR 15144-1:2014

Calculation did not run. (Lubricant: Load stage micropitting test is unknown.)

5. SCUFFING LOAD CAPACITY

Calculation method according to ISO TR 13989:2000

Lubrication coefficient (for lubrication type)	[XS]	1.000	
Scuffing test and load stage	[FZGtest]	FZG - Test A / 8.3 / 90 (ISO 14635 - 1)	12
Multiple meshing factor	[Xmp]	1.000	
Relative structure coefficient (Scuffing)	[XWrelT]	1.000	
Thermal contact factor (N/mm/s ^{0.5} /K)	[BM]	13.780	13.780

Relevant tip relief (μm)	[Ca]	2.00	2.00
Optimal tip relief (μm)	[Ceff]	50.51	
Ca taken as optimal in the calculation (0=no, 1=yes)		0	0
Effective facewidth (mm)	[beff]	23.000	
Applicable circumferential force/facewidth (N/mm)	[wBt]	809.994	
Kbg = 1.046, wBt*Kbg = 846.867			
Angle factor (ε1:0.563, ε2:0.372)	[Xalfbet]	1.089	

Flash temperature-criteria

Lubricant factor	[XL]	0.830	
Tooth mass temperature (°C) (theMi = theoil + XS*0.47*Xmp*theflm)	[theMi]	77.14	
Average flash temperature (°C)	[theflm]	15.19	
Scuffing temperature (°C)	[theS]	348.80	
Coordinate gamma (point of highest temp.) [Gamma.A]=-0.114 [Gamma.E]=0.173	[Gamma]	0.173	
Highest contact temp. (°C)	[theB]	111.41	
Flash factor (°K*N ^{0.75} *s ^{0.5} *m ^{-0.5} mm)	[XM]	50.058	
Approach factor	[XJ]	1.000	
Load sharing factor	[XGam]	1.000	
Dynamic viscosity (mPa*s)	[etaM]	41.90 (70.0 °C)	
Coefficient of friction	[mym]	0.061	
Required safety	[SBmin]	2.000	
Safety factor for scuffing (flash temperature)	[SB]	6.733	

Integral temperature-criteria

Lubricant factor	[XL]	1.000	
Tooth mass temperature (°C) (theMC = theoil + XS*0.70*theflaint)	[theMC]	80.57	
Mean flash temperature (°C)	[theflaint]	15.09	
Integral scuffing temperature (°C)	[theSint]	360.78	
Flash factor (°K*N ^{0.75} *s ^{0.5} *m ^{-0.5} mm)	[XM]	50.058	
Running-in factor (well run in)	[XE]	1.000	
Contact ratio factor	[Xeps]	0.432	
Dynamic viscosity (mPa*s)	[etaOil]	41.90 (70.0 °C)	
Mean coefficient of friction	[mym]	0.062	
Geometry factor	[XBE]	0.103	
Meshing factor	[XQ]	1.000	
Tip relief factor	[XCa]	1.053	
Integral tooth flank temperature (°C)	[theint]	103.21	
Required safety	[SSmin]	1.800	
Safety factor for scuffing (intg.-temp.)	[SSint]	3.496	
Safety referring to transmittable torque	[SSL]	8.757	

6. MEASUREMENTS FOR TOOTH THICKNESS

		----- Gear 1 -----	Gear 2 --
Tooth thickness deviation		DIN 3967 cd25	DIN 3967 cd25
Tooth thickness allowance (normal section) (mm)	[As.e/i]	-0.070 / -0.110	-0.070 / -0.110
Number of teeth spanned	[k]	7.000	10.000
Base tangent length (no backlash) (mm)	[Wk]	36.634	53.016
Actual base tangent length ('span') (mm)	[Wk.e/i]	36.568 / 36.530	52.950 / 52.913
(mm)	[ΔWk.e/i]	-0.066 / -0.103	-0.066 / -0.103
Diameter of measuring circle (mm)	[dMWk.m]	73.534	109.428

Theoretical diameter of ball/pin (mm)	[DM]	4.157	3.969
Effective diameter of ball/pin (mm)	[DMeff]	4.250	4.000
Radial single-ball measurement backlash free (mm)	[MrK]	40.656	58.115
Radial single-ball measurement (mm)	[MrK.e/i]	40.594 / 40.559	58.048 / 58.010
Diameter of measuring circle (mm)	[dMMr.m]	74.732	110.145
Diametral measurement over two balls without clearance (mm)	[MdK]	81.312	116.188
Diametral two ball measure (mm)	[MdK.e/i]	81.188 / 81.117	116.054 / 115.977
Diametral measurement over pins without clearance (mm)	[MdR]	81.312	116.230
Measurement over pins according to DIN 3960 (mm)	[MdR.e/i]	81.188 / 81.117	116.097 / 116.020
Measurement over 2 pins (free) according to AGMA 2002 (mm)	[dk2f.e/i]	0.000 / 0.000	116.050 / 115.974
Measurement over 2 pins (transverse) according to AGMA 2002 (mm)	[dk2t.e/i]	0.000 / 0.000	116.138 / 116.062
Measurement over 3 pins (axial) according to AGMA 2002 (mm)	[dk3A.e/i]	81.188 / 81.117	116.097 / 116.020
Chordal tooth thickness (no backlash) (mm)	[sc]	4.900	5.295
Actual chordal tooth thickness (mm)	[sc.e/i]	4.830 / 4.790	5.225 / 5.185
Reference chordal height from da.m (mm)	[ha]	3.771	4.293
Tooth thickness (Arc) (mm)	[sn]	4.904	5.297
(mm)	[sn.e/i]	4.834 / 4.794	5.227 / 5.187
Backlash free center distance (mm)	[aControl.e/i]	91.357 / 91.275	
Backlash free center distance, allowances (mm)	[jta]	-0.143 / -0.225	
dNf.i with aControl (mm)	[dNf0.i]	71.000	106.676
Reserve (dNf0.i-dFf.e)/2 (mm)	[cF0.i]	-0.160	-0.132
Tip clearance (mm)	[c0.i(aControl)]	0.308	0.308
Centre distance allowances (mm)	[Aa.e/i]	0.018 / -0.018	
Circumferential backlash from Aa (mm)	[jtw_Aa.e/i]	0.019 / -0.019	
Radial clearance (mm)	[jrw]	0.243 / 0.126	
Circumferential backlash (transverse section) (mm)	[jtw]	0.261 / 0.135	
Normal backlash (mm)	[jnw]	0.237 / 0.123	
Torsional angle at entry with fixed output:			
Entire torsional angle (°)	[j.tSys]	0.4087/0.2117	

7. GEAR ACCURACY

----- GEAR 1 ----- GEAR 2 --

According to ISO 1328-1:1995, ISO 1328-2:1997

Accuracy grade	[Q]	6	6
Single pitch deviation (μm)	[fptT]	7.50	7.50
Base circle pitch deviation (μm)	[fpbT]	7.00	7.00
Sector pitch deviation over k/8 pitches (μm)	[Fpk/8T]	12.00	13.00
Profile form deviation (μm)	[ffaT]	6.50	6.50
Profile slope deviation (μm)	[fHaT]	5.50	5.50
Total profile deviation (μm)	[FaT]	8.50	8.50
Helix form deviation (μm)	[ffbT]	8.50	8.50
Helix slope deviation (μm)	[fHbT]	8.50	8.50
Total helix deviation (μm)	[FbT]	12.00	12.00
Total cumulative pitch deviation (μm)	[FpT]	26.00	26.00
Runout (μm)	[FrT]	21.00	21.00
Single flank composite, total (μm)	[FisT]	39.00	39.00
Single flank composite, tooth-to-tooth (μm)	[fisT]	13.00	13.00
Radial composite, total (μm)	[FidT]	31.00	31.00

Radial composite, tooth-to-tooth (μm)	[fidT]	9.50	9.50
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Axis alignment tolerances (recommendation acc. to ISO TR 10064-3:1996, Quality 6)

Maximum value for deviation error of axis (μm)	[fSigbet]	12.00 (Fb= 12.00)
Maximum value for inclination error of axes (μm)	[fSigdel]	24.00

8. ADDITIONAL DATA

Maximal possible centre distance ($\text{eps}_a=1.0$)	[aMAX]	91.335	
Mass (kg)	[m]	0.760	1.675
Total mass (kg)	[m]	2.435	
Moment of inertia (system with reference to the drive): calculation without consideration of the exact tooth shape			
single gears ((da+df)/2...di) ($\text{kg}\cdot\text{m}^2$)	[TraeghMom]	0.0005107	0.002479
System ((da+df)/2...di) ($\text{kg}\cdot\text{m}^2$)	[TraeghMom]	0.001519	
Torsional stiffness at entry with driven force fixed:			
Torsional stiffness (MNm/rad)	[cr]	0.285	
Torsion when subjected to nominal torque ($^\circ$)	[delcr]	0.096	
Mean coeff. of friction (acc. Niemann)	[mum]	0.068	
Wear sliding coef. by Niemann	[zetw]	0.204	
Gear power loss (kW)	[PVZ]	0.281	
(Meshing efficiency (%))	[etaz]	99.532)	
Sound pressure level (according to Masuda)	[dB(A)]	77.5	

Indications for the manufacturing by wire cutting:

Deviation from theoretical tooth trace (μm)	[WireErr]	137.8	91.9
Permissible deviation (μm)	[Fb/2]	6.0	6.0

9. MODIFICATIONS AND TOOTH FORM DEFINITION

Data for the tooth form calculation :

Data not available.

10. SERVICE LIFE, DAMAGE

Required safety for tooth root	[SFmin]	1.35
Required safety for tooth flank	[SHmin]	0.97

Service life (calculated with required safeties):

System service life (h)	[Hatt]	0.728
-------------------------	--------	-------

Tooth root service life (h)	[HFatt]	0.7279	1.142
Tooth flank service life (h)	[HHatt]	143.3	215

Damage calculated on the basis of the required service life [H] (20000.0 h)

F1%	F2%	H1%	H2%
9999.99	9999.99	9999.99	9303.44

Damage calculated on basis of system service life [Hatt] (0.7 h)

F1%	F2%	H1%	H2%
100.00	63.72	0.51	0.34

Calculation of the factors required to define reliability $R(t)$ according to B. Bertsche with Weibull distribution:

$$R(t) = 100 * \exp(-((t^{*fac} - t_0)/(T - t_0))^b) \% ; t (h)$$

Gear		fac	b	t0	T	R(H)%
1	Tooth root	72000	1.7	5.059e+004	7.775e+004	0.00
1	Tooth flank	72000	1.3	9.301e+006	4.432e+007	0.00
2	Tooth root	48000	1.7	5.294e+004	8.135e+004	0.00
2	Tooth flank	48000	1.3	9.301e+006	4.432e+007	0.00

Reliability of the configuration for required service life (%) 0.00 (Bertsche)

REMARKS:

- Specifications with [e/i] imply: Maximum [e] and Minimal value [i] with consideration of all tolerances
Specifications with [m] imply: Mean value within tolerance
- For the backlash tolerance, the center distance tolerances and the tooth thickness deviation are taken into account. Shown is the maximal and the minimal backlash corresponding the largest resp. the smallest allowances
The calculation is done for the operating pitch circle.
- Calculation of Zbet according Corrigendum 1 ISO 6336-2:2008 with $Z_{bet} = 1/(\cos(\beta))^0.5$
- Details of calculation method:
cg according to method B
KV according to method B
KHb, KFb according method C
fma following equation (64), fsh following (57/58), Fbx following (52/53/57)
KH_a, KF_a according to method B
- The logarithmically interpolated value taken from the values for the fatigue strength and the static strength, based on the number of load cycles, is used for coefficients ZL, ZV, ZR, ZW, ZX, YdreIT, YRreIT and YX..

End of Report

lines: 533

Dynamic Analysis Figures

C.1 C14

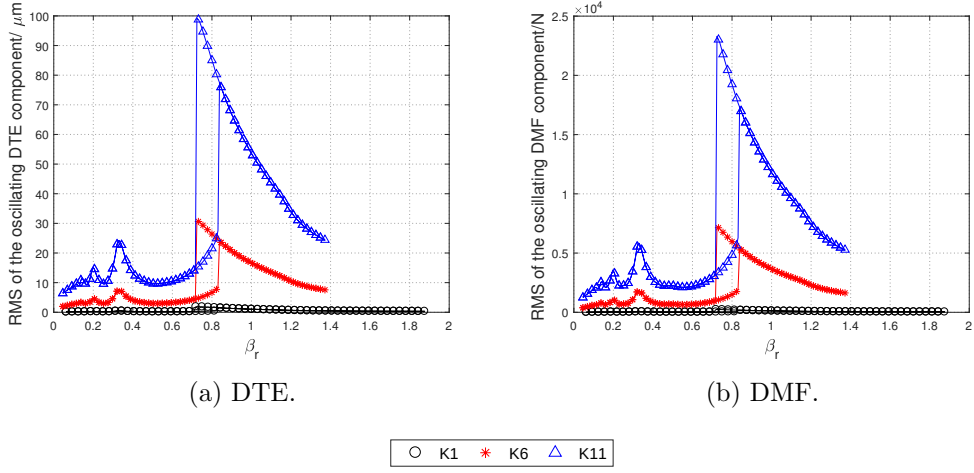


Figure C.1: RMS of the oscillating component of dynamic parameters for all three torques for the C14.

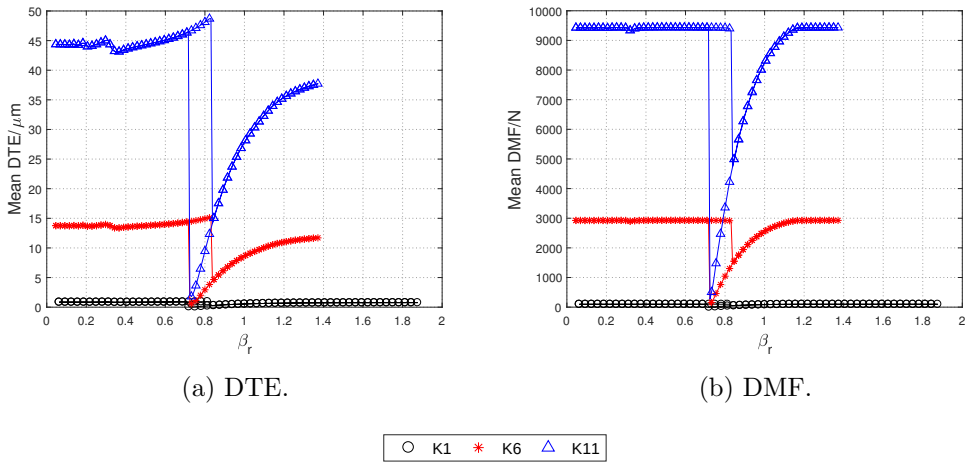


Figure C.2: Mean dynamic parameters for all three torques for the C14.

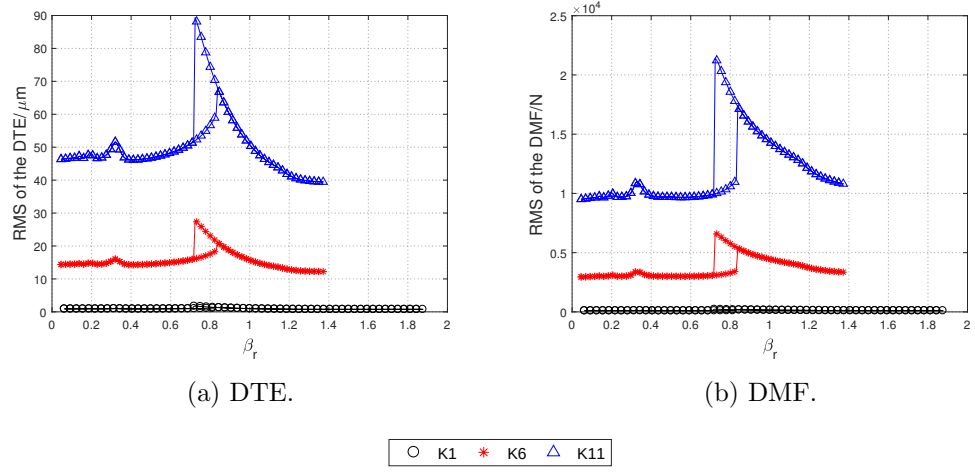


Figure C.3: RMS of dynamic parameters for all three torques for the C14.

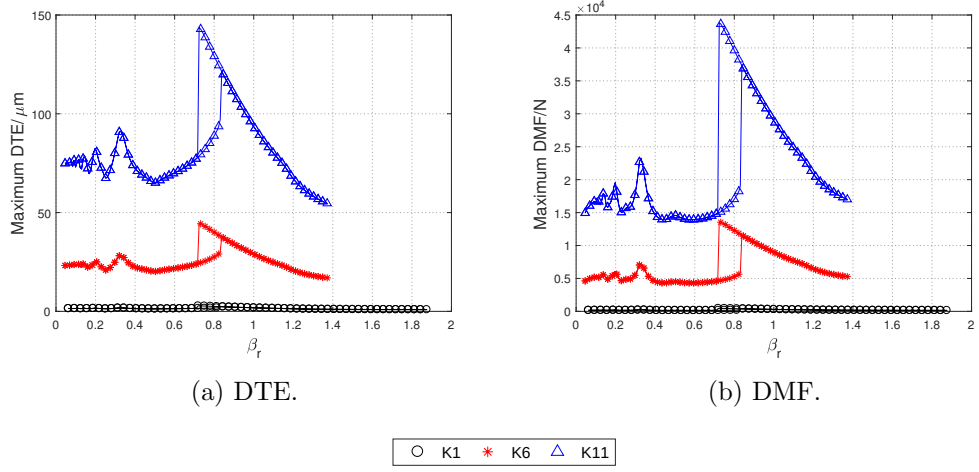


Figure C.4: Maximum dynamic parameters for all three torques for the C14

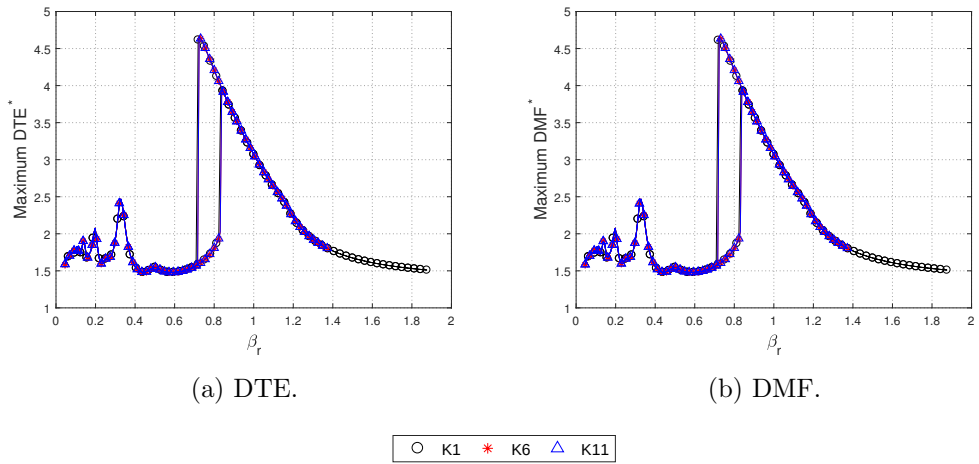


Figure C.5: Maximum normalized dynamic parameters for all three torques for the C14.

C.2 H501

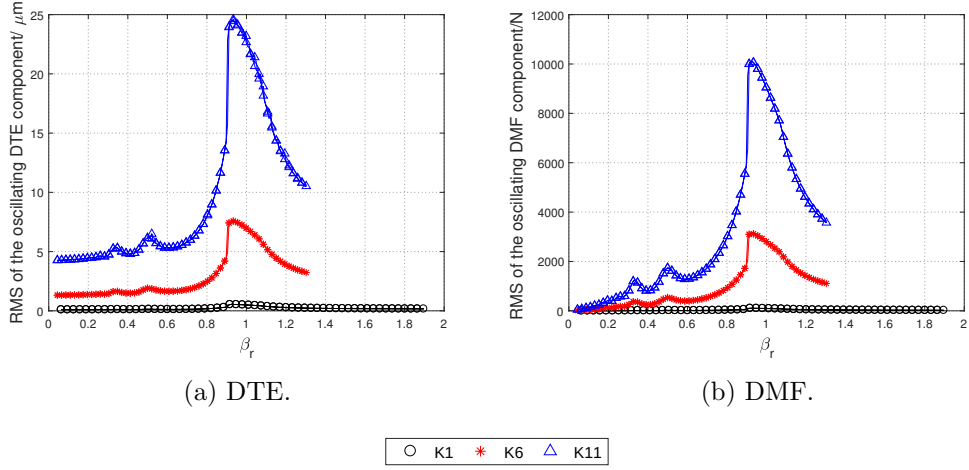


Figure C.6: RMS of the oscillating component of dynamic parameters for all three torques for the H501.

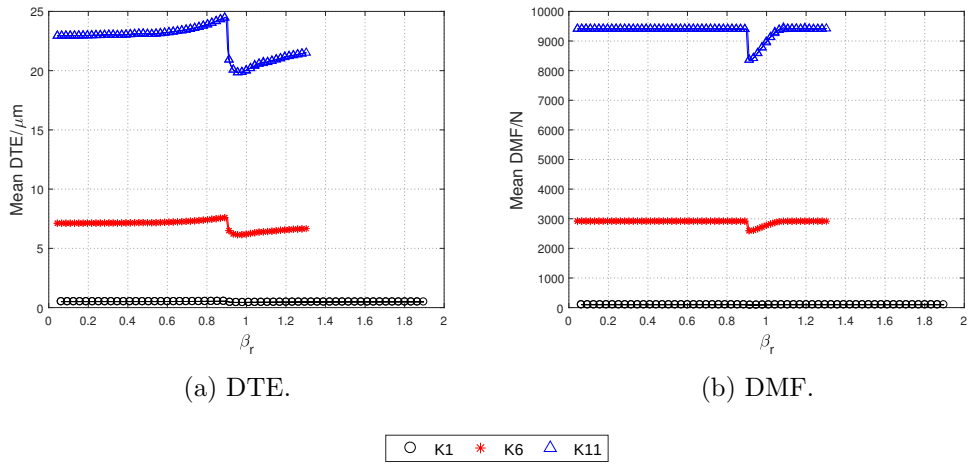


Figure C.7: Mean dynamic parameters for all three torques for the H501.

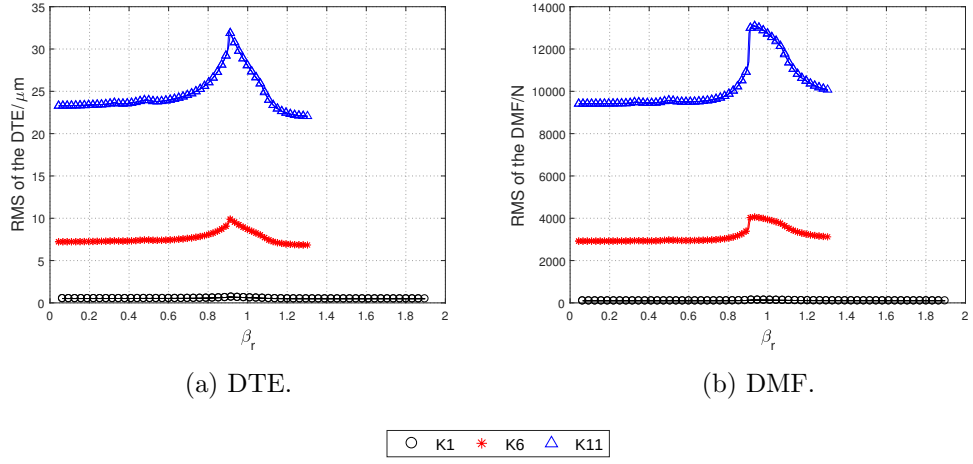


Figure C.8: RMS of dynamic parameters for all three torques for the H501.

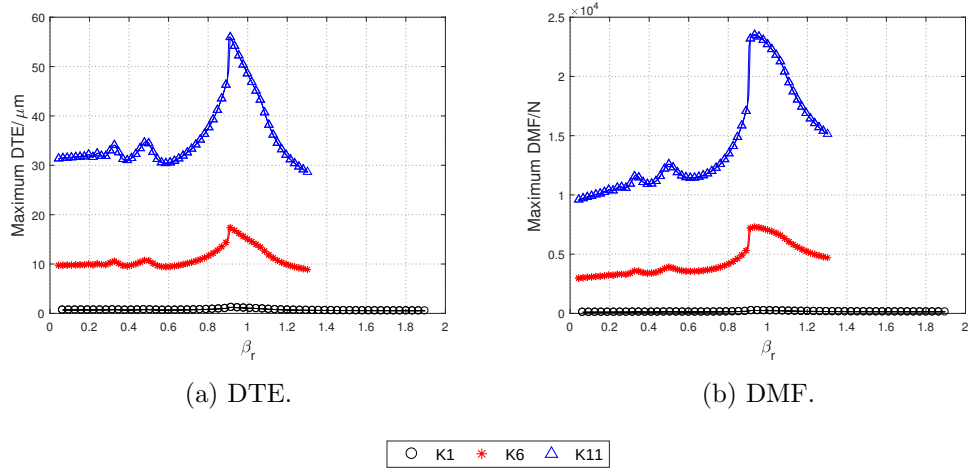


Figure C.9: Maximum dynamic parameters for all three torques for the H501.

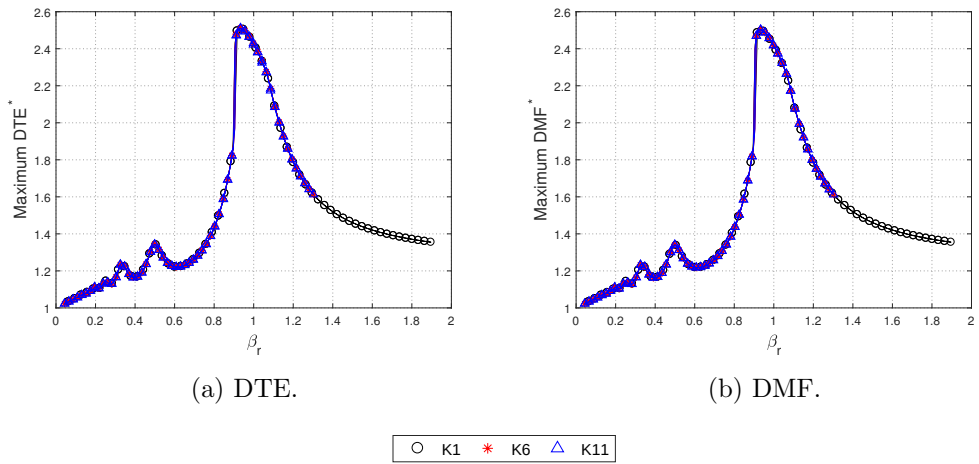


Figure C.10: Maximum normalized dynamic parameters for all three torques for the H501.

C.3 H951

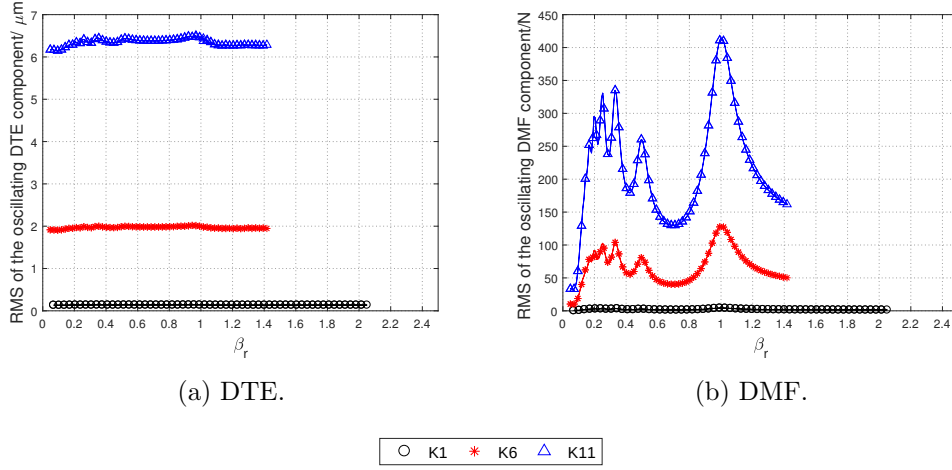


Figure C.11: RMS of the oscillating component of dynamic parameters for all three torques for the H951.

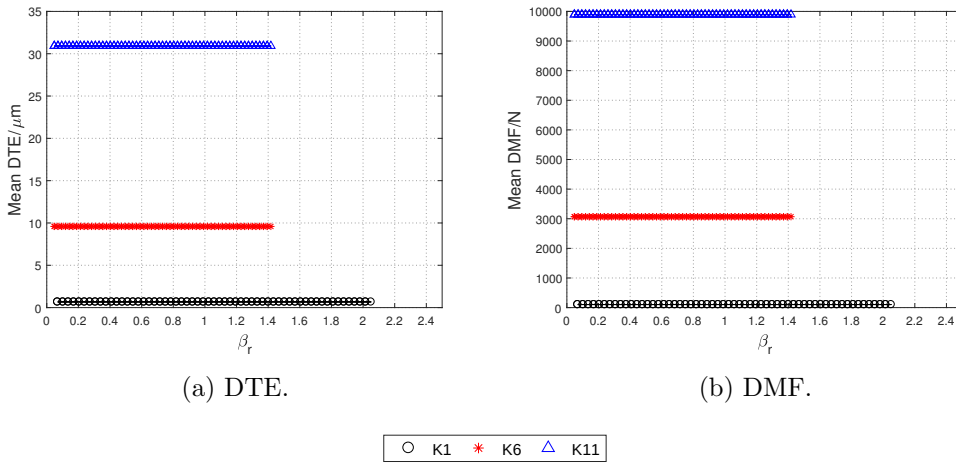


Figure C.12: Mean dynamic parameters for all three torques for the H951.

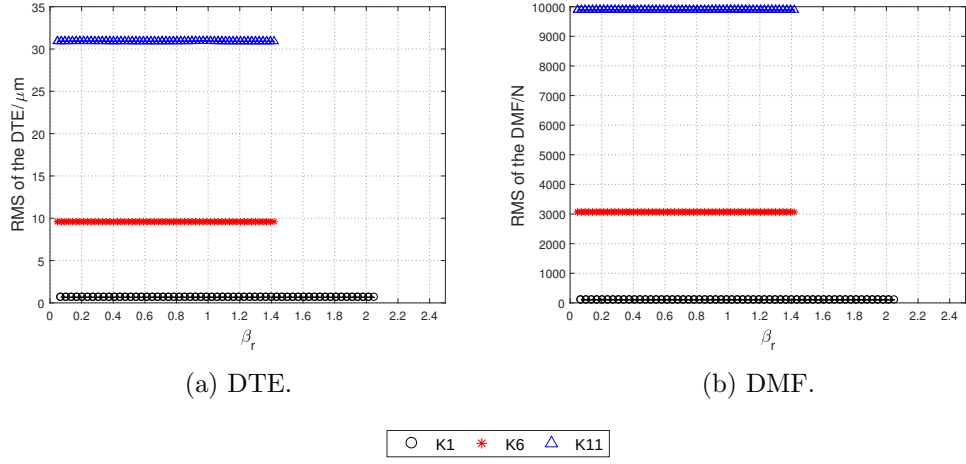


Figure C.13: RMS of dynamic parameters for all three torques for the H951.

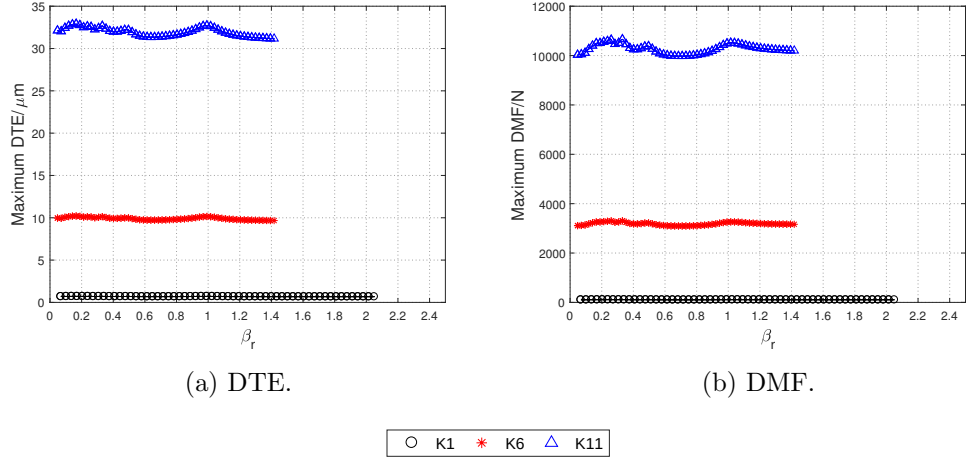


Figure C.14: Maximum dynamic parameters for all three torques for the H951.

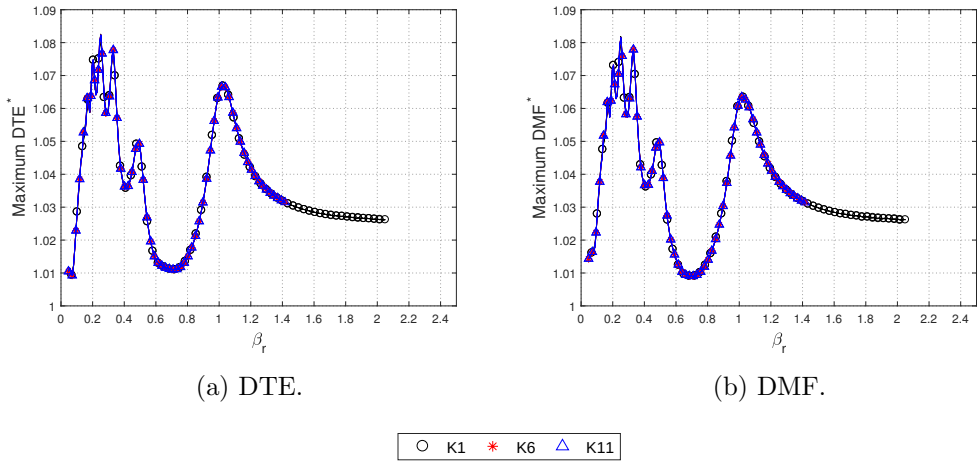


Figure C.15: Maximum normalized dynamic parameters for all three torques for the H951.

C.4 CMS55-E

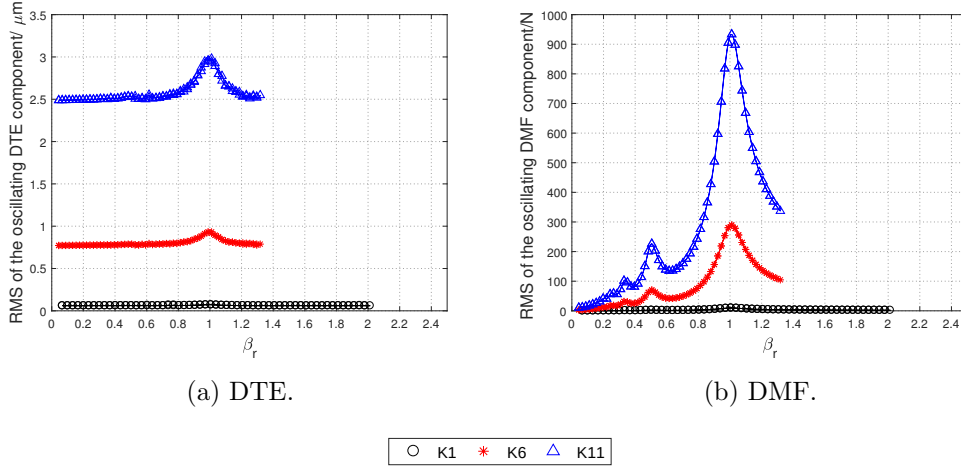


Figure C.16: RMS of the oscillating component of dynamic parameters for all three torques for the CMS55-E.

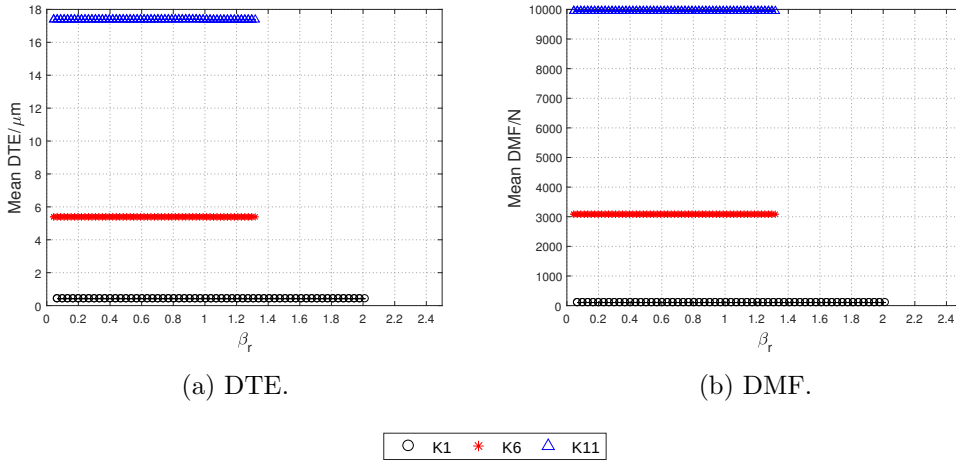


Figure C.17: Mean dynamic parameters for all three torques for the CMS55-E.

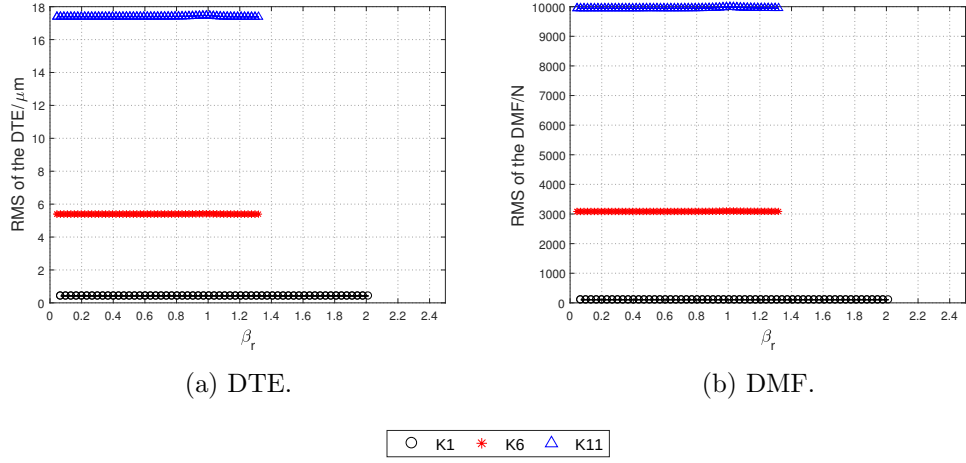


Figure C.18: RMS of dynamic parameters for all three torques for the CMS55-E.

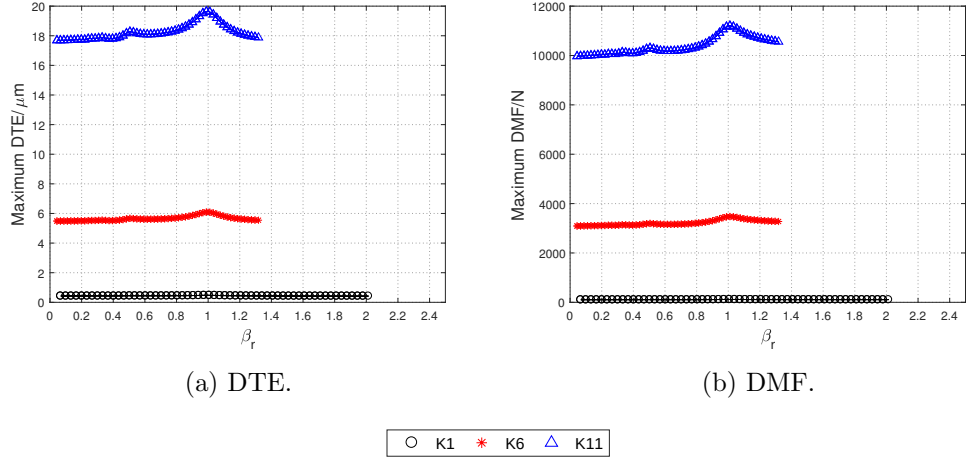


Figure C.19: Maximum dynamic parameters for all three torques for the CMS55-E.

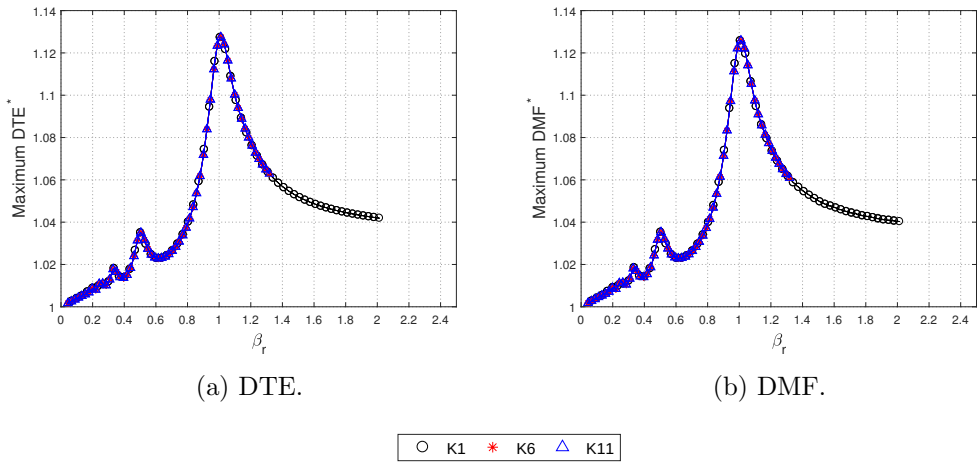


Figure C.20: Maximum normalized dynamic parameters for all three torques for the CMS55-E.

C.5 CMS55-R

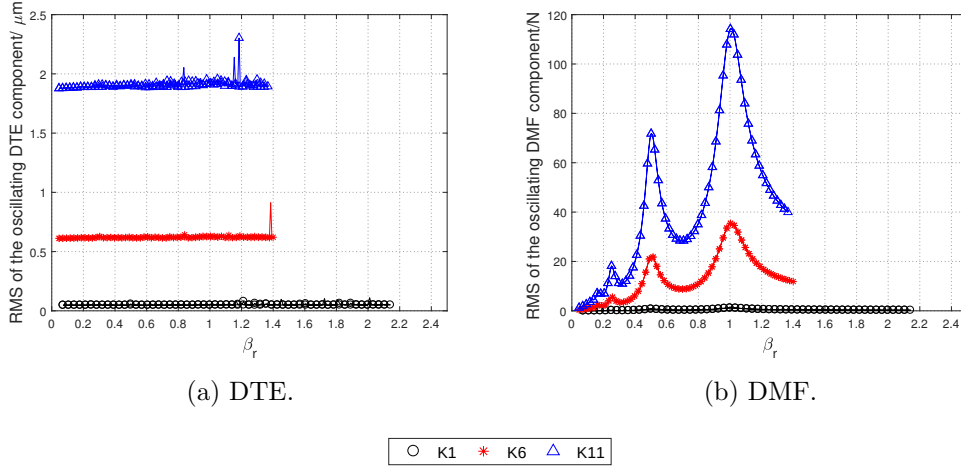


Figure C.21: RMS of the oscillating component of dynamic parameters for all three torques for the CMS55-R.

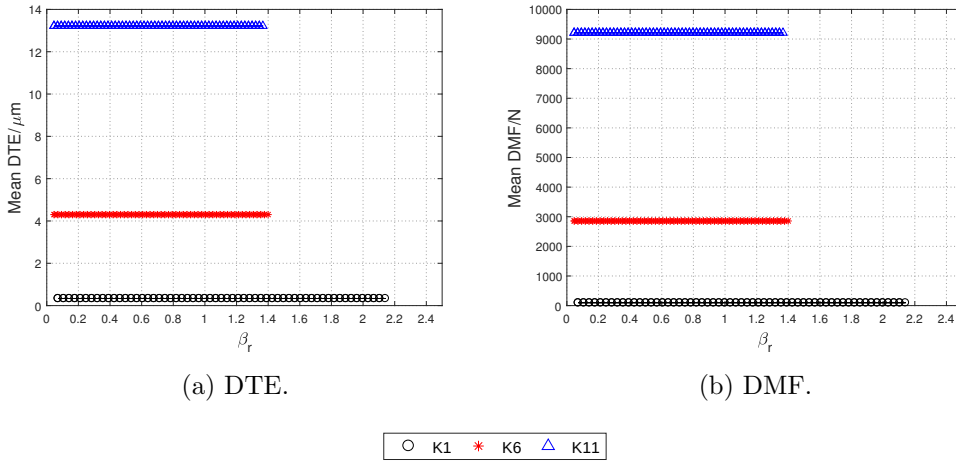


Figure C.22: Mean dynamic parameters for all three torques for the CMS55-R.

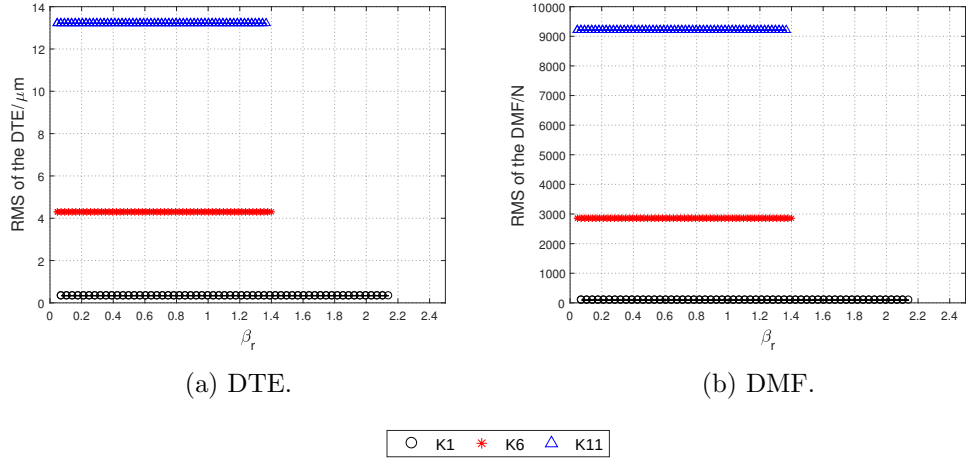


Figure C.23: RMS of dynamic parameters for all three torques for the CMS55-R.

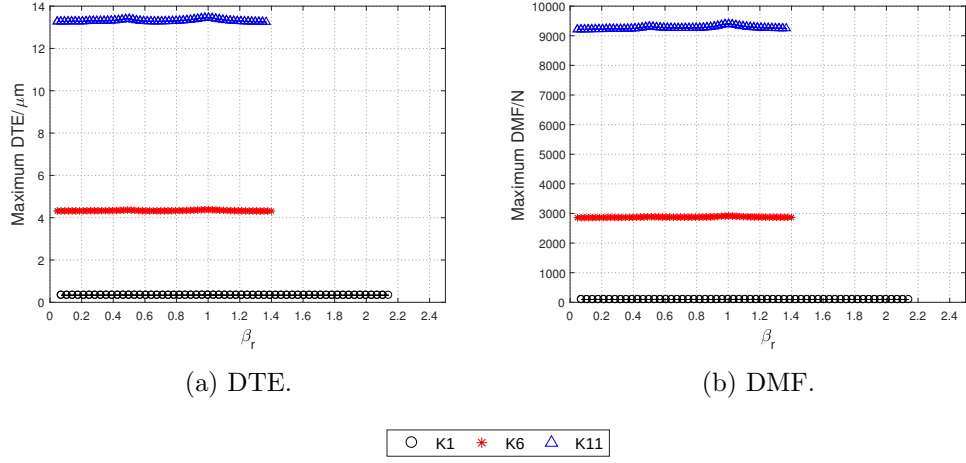


Figure C.24: Maximum dynamic parameters for all three torques for the CMS55-R.

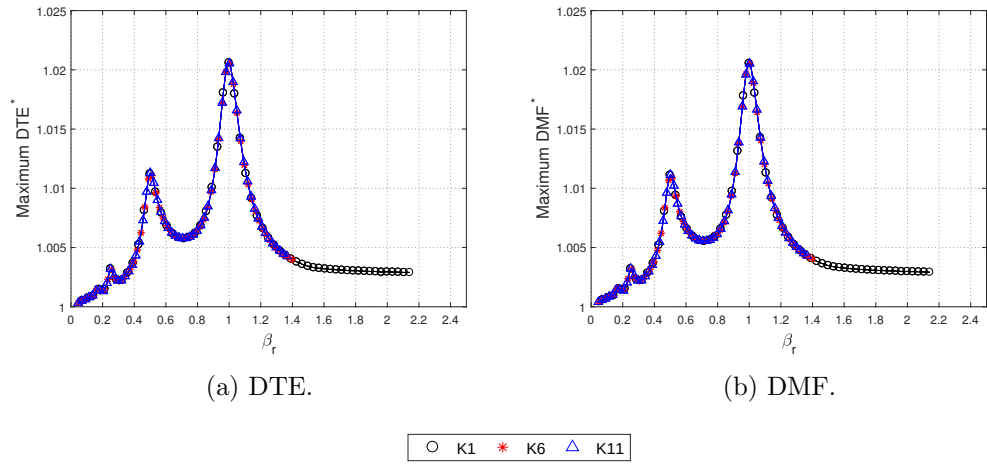


Figure C.25: Maximum normalized dynamic parameters for all three torques for the CMS55-R.

C.6 CMS92-LE

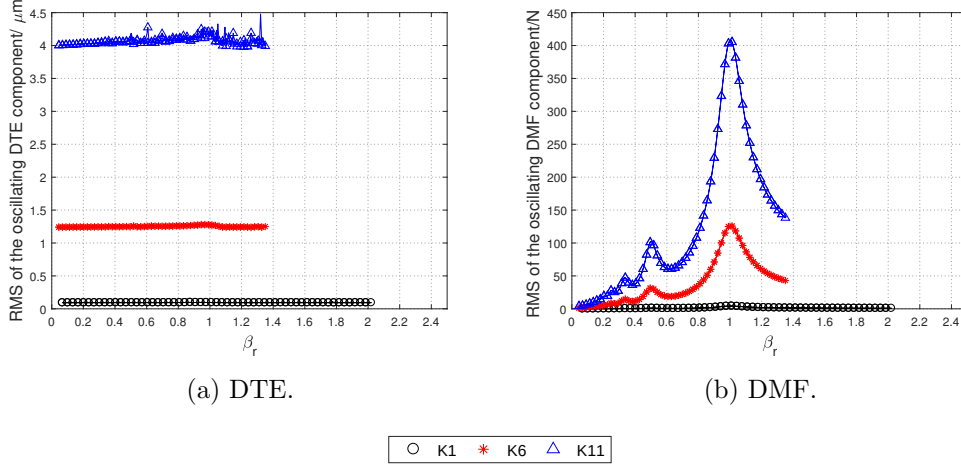


Figure C.26: RMS of the oscillating component of dynamic parameters for all three torques for the CMS92-LE.

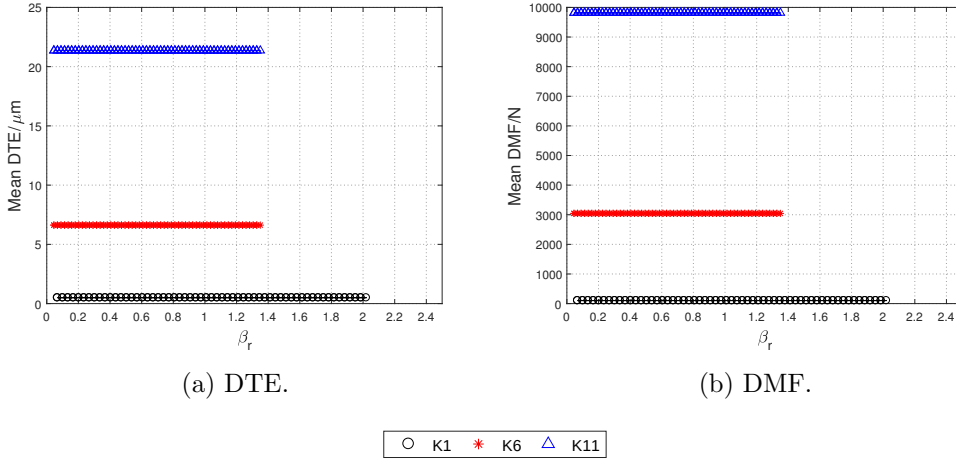


Figure C.27: Mean dynamic parameters for all three torques for the CMS92-LE.

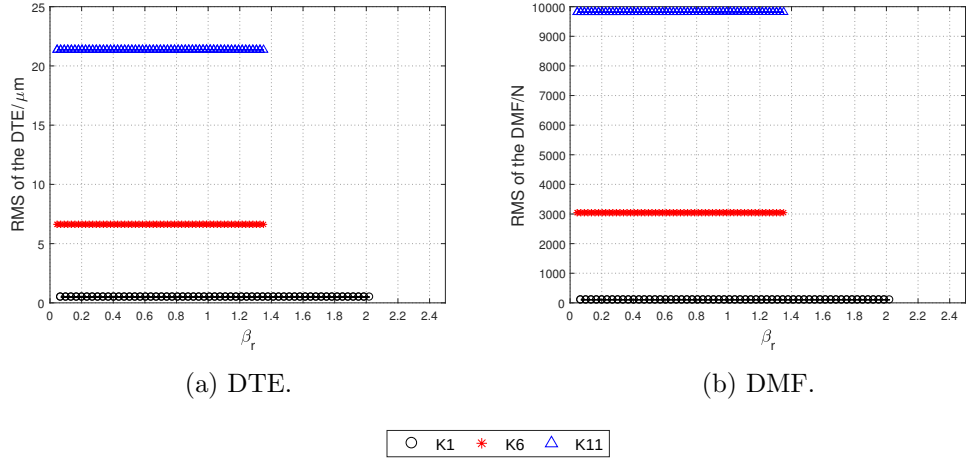


Figure C.28: RMS of dynamic parameters for all three torques for the CMS92-LE.

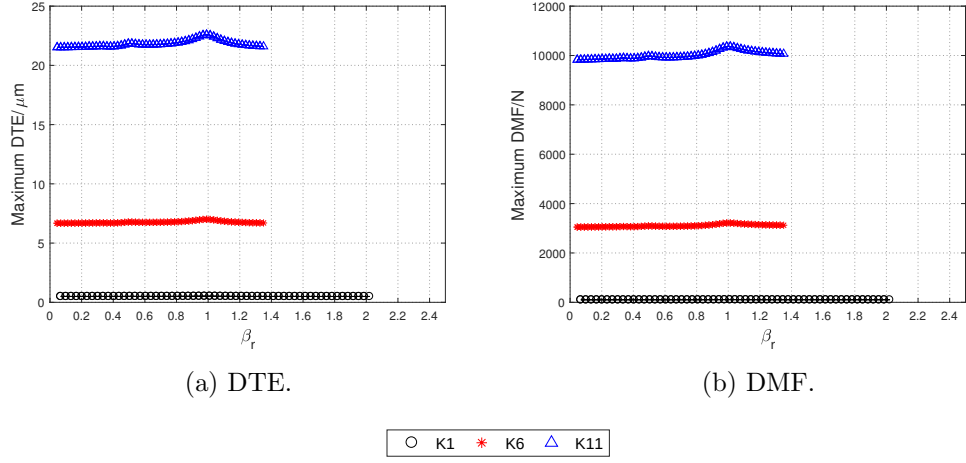


Figure C.29: Maximum dynamic parameters for all three torques for the CMS92-LE.

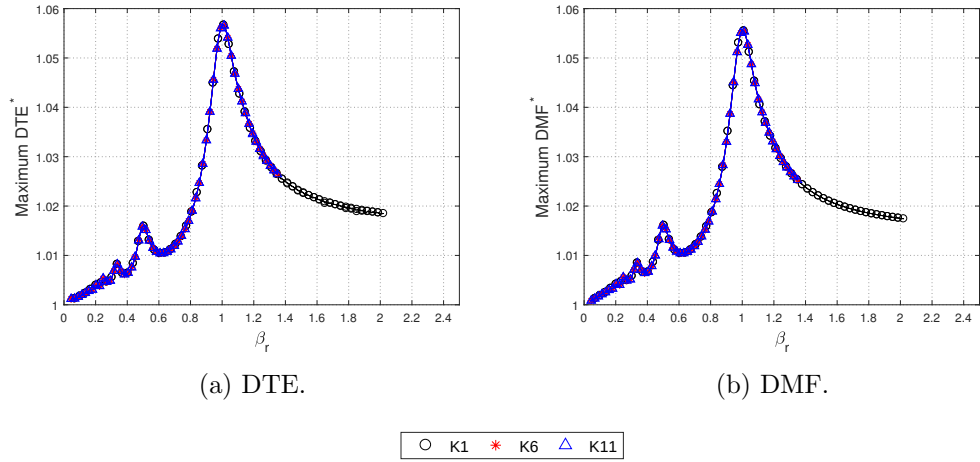


Figure C.30: Maximum normalized dynamic parameters for all three torques for the CMS92-LE.

C.7 CMS50-E

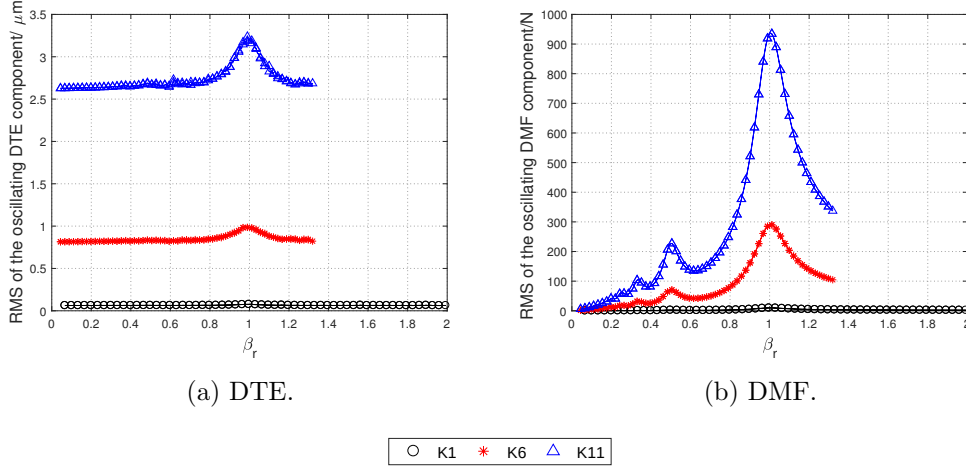


Figure C.31: RMS of the oscillating component of dynamic parameters for all three torques for the CMS50-E.

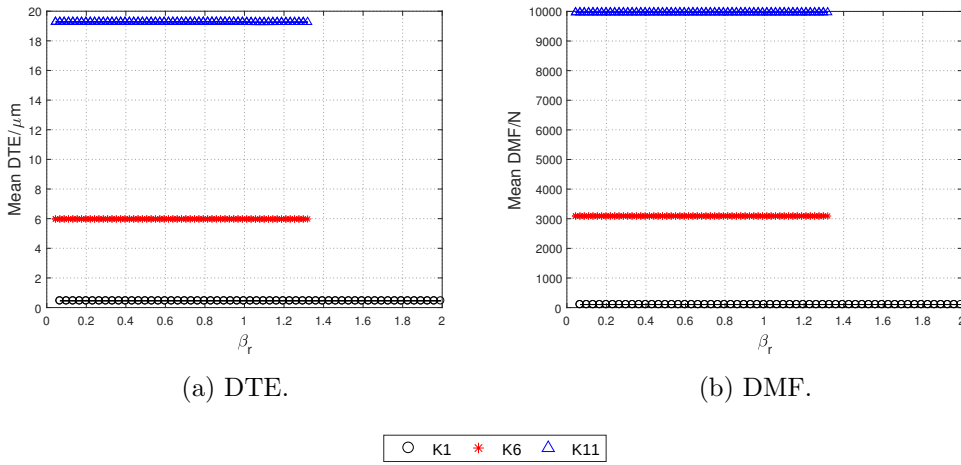


Figure C.32: Mean dynamic parameters for all three torques for the CMS50-E.

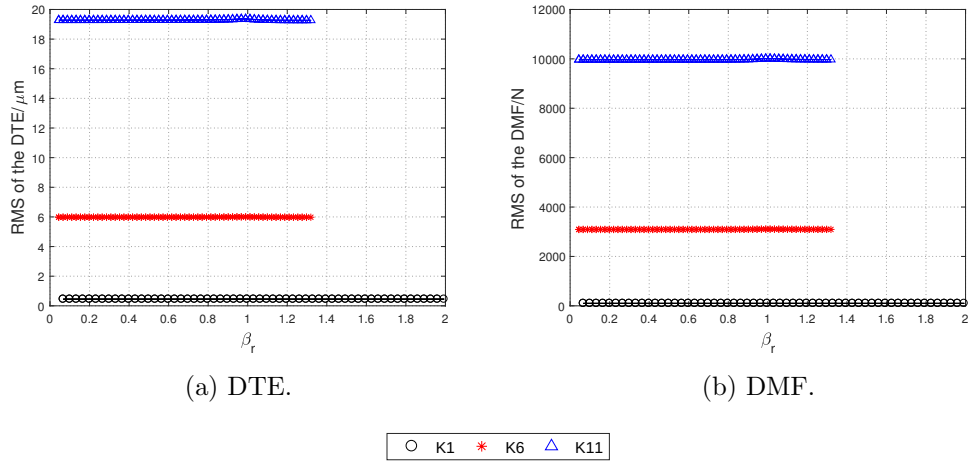


Figure C.33: RMS of dynamic parameters for all three torques for the CMS50-E.

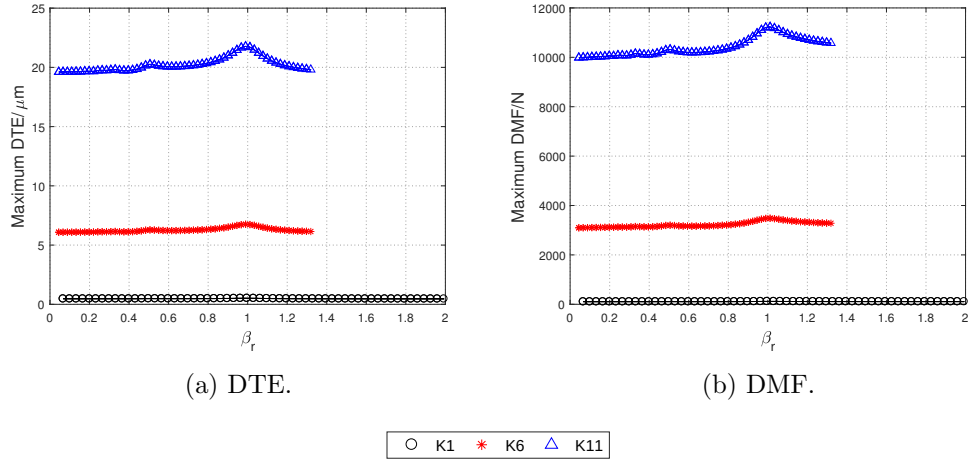


Figure C.34: Maximum dynamic parameters for all three torques for the CMS50-E.

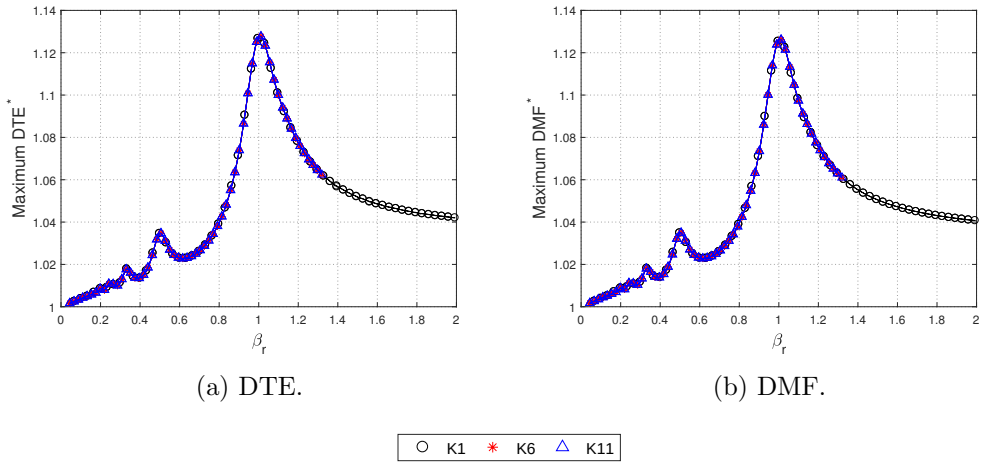


Figure C.35: Maximum normalized dynamic parameters for all three torques for the CMS50-E.

C.8 CMS60-R

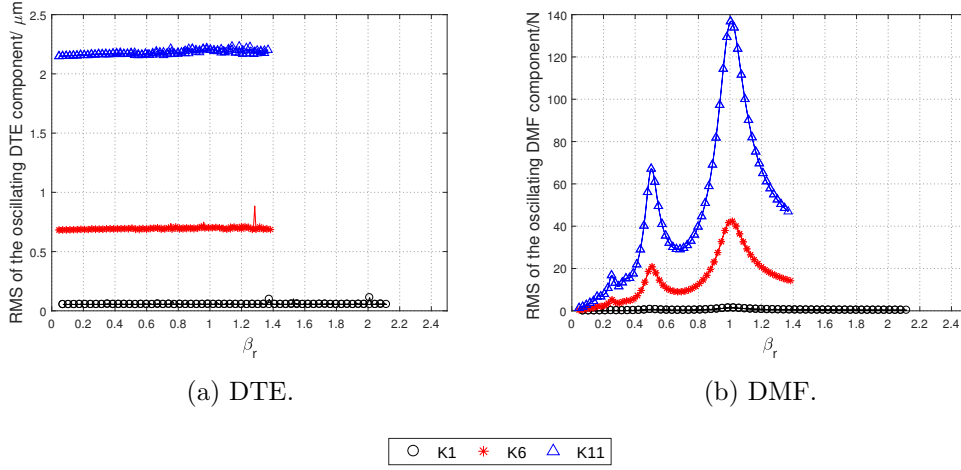


Figure C.36: RMS of the oscillating component of dynamic parameters for all three torques for the CMS60-R.

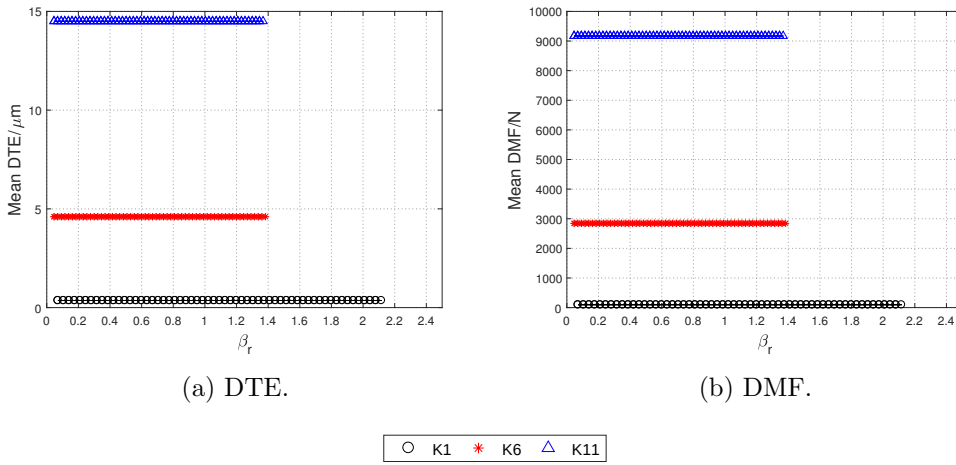


Figure C.37: Mean dynamic parameters for all three torques for the CMS60-R.

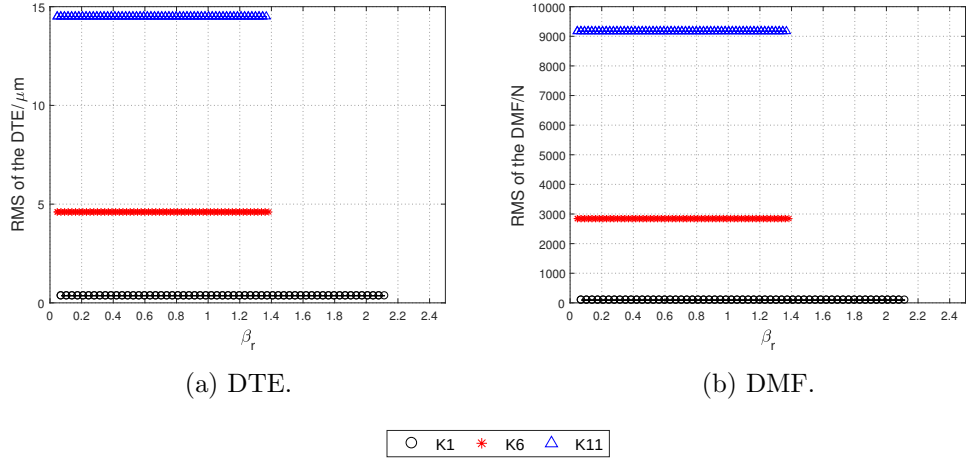


Figure C.38: RMS of dynamic parameters for all three torques for the CMS60-R.

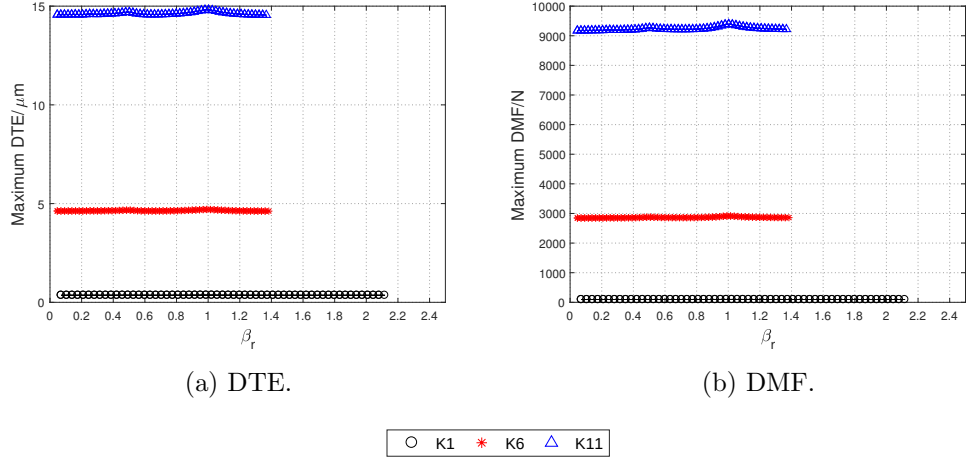


Figure C.39: Maximum dynamic parameters for all three torques for the CMS60-R.

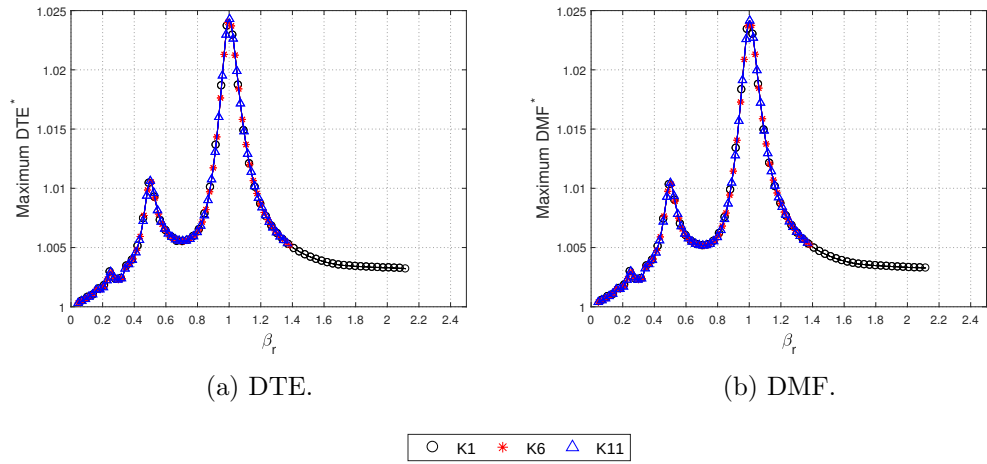


Figure C.40: Maximum normalized dynamic parameters for all three torques for the CMS60-R.