Bevel Gear pattern optimisation

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Abstract

Usually, a finite element model allows giving an idea of the housing and gear shaft deflexion. These deflexions values are taken into account in order to simulate the relative displacements between the pinion and the wheel and to give the consequence in terms of tooth contact pattern.

In fact, this method is not precise enough to analyse the strength in the root tooth area and/ or the Hertz pressure distribution. So, a further step in the bevel gear development has been achieved by taking into account the effect of the deflexions due to the rim and the web of the gear area. These parameters are particularly significant in helicopter gears behaviour where the reduction of weight leads to slim and consequently flexible gear design.

The different web configurations used in the gearbox design have been analysed and a parametric study has been performed.

The method is based on the mutual use of specific bevel gear development software and CATIA C.A.D. Software. The bevel gear software has been developed with the expertise of INSA (Institut National des Sciences Appliquées) laboratory. It provides all the information needed to generate the teeth (flanks, root, profile, ….), while the CATIA software is used to generate the 3D model of the teeth and to perform the automatic merger between the web and the teeth. In the end, the complete 3D model of the bevel gear shaft is therefore obtained and several loading cases are analysed.

The meshing and the calculation are realised respectively with CATIA MESH software and ELFINI software.

The results are introduced in a new SPIRO software module in order to simulate the contact pattern, the hertz pressure distribution and the tooth root strength. This software has been validated by a bevel gear transmission error test.

Introduction

Actually, the optimisation of the tooth contact pattern depends on the correct housing and gear deflexion knowledge. Theses deflexion are obtained thanks to a FEM model. These deflexion are introduce in an EC SPIRO software to obtain :

- The model the geometry of the tooth
- The model the inner cone shaft in order to take into account the stiffness of the root tooth
- The optimisation of the setting machine in function of the topography desired.

On an another side, We also have to know the contribution of the flexion of the tooth as well as the local deformations due to the pressure of contact.
So, the rim, the web and the shaft stiffness are taken into account. It is the reason why this software has been modified.

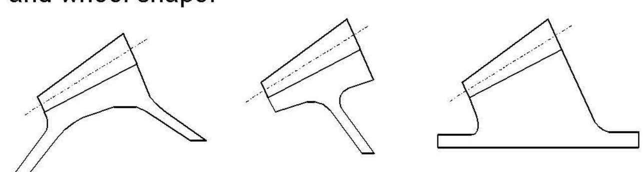
The first step of the software development was to parameterise the more usefully the pinion and wheel design.

Parameter setting of the pinion and the wheel

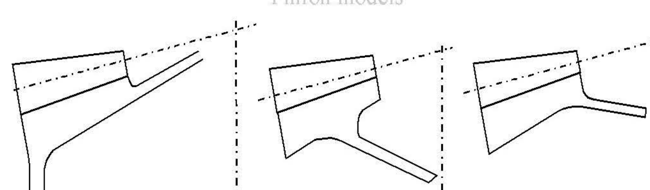
According the tooth shape and location, it is possible to establish formulas which allow us to model the rim, the web and the link between them. These formulas have been established from the EC designer knowledge :

- rim and web thickness,
- number of webs
- web tilt angle
- web limits between the webs and the shaft.

The following drawings show different kinds of pinion and wheel shape.

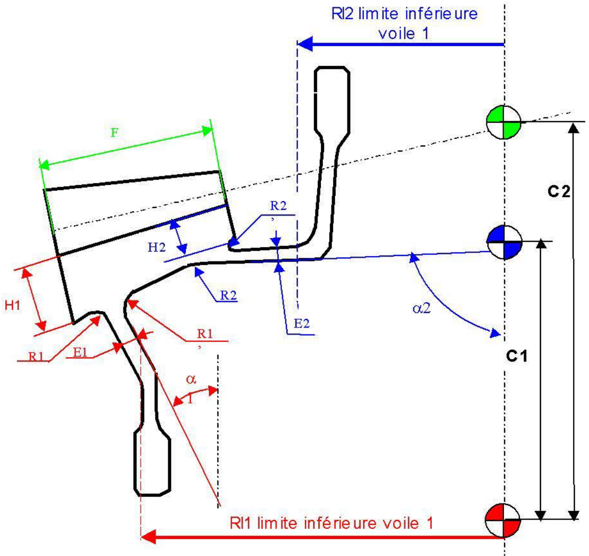


Pinion models



Wheel models

The following scheme gives an idea of this parameter setting.

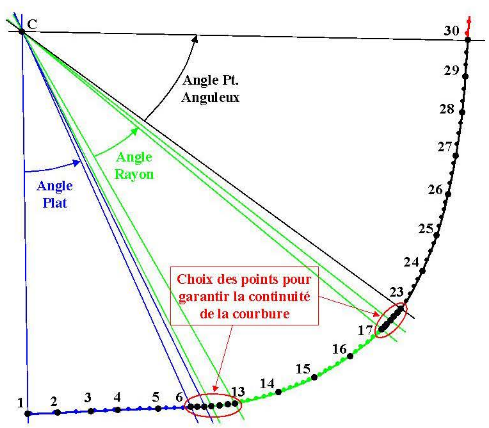


Parameters

Automatic CATIA Software models generation :

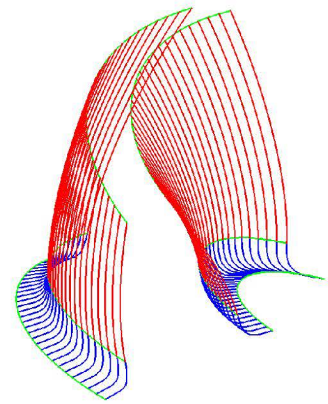
It is possible to create CATIA models from bevel gear surface topographic files. So, the SPIRO output file format has been modified in order to perform an automatic tooth 3D modelling.

We have to optimise the number of points on the root and on the flank of the tooth. The curvature continuity has to be also respected.

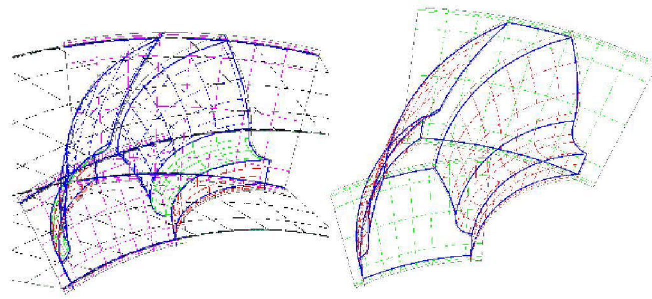


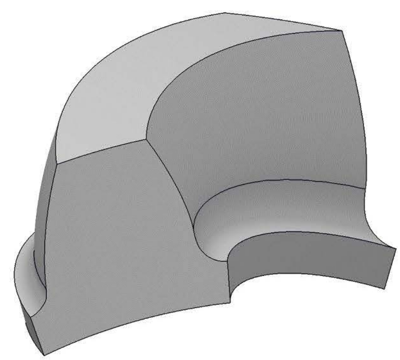
Root tooth curvature

After, the main profile lines are built.

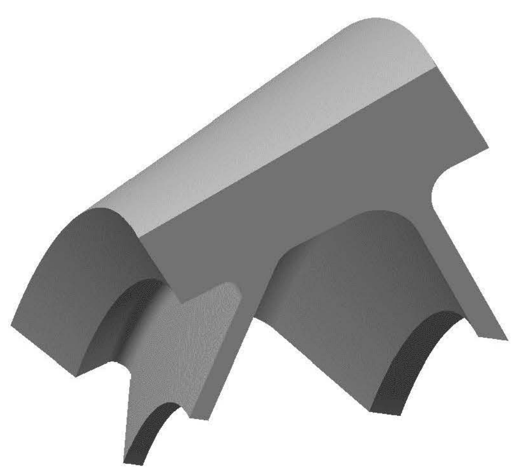


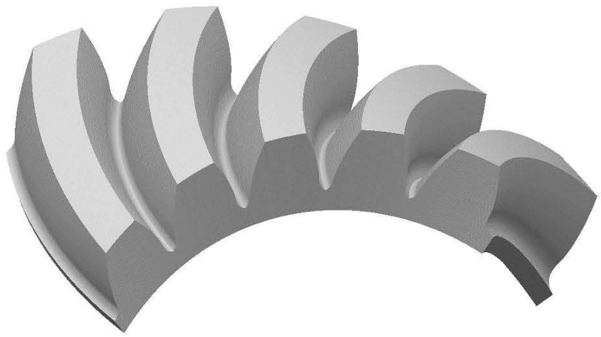
Thanks to special CATIA function, we create a 3D model.





At the end we have to merge the tooth 3D model with the rim-web 3D model and duplicate this model to obtain a five teeth 3D model.

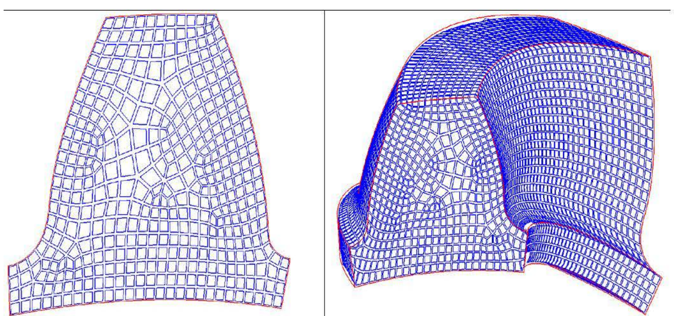




5 teeth model

Finite Element Model :

We have to optimise the number and the location of the meshing nodes.
A surface meshing has to be created on inner diameter or on the outer diameter. So, the meshing is propagated to the opposite side.



Maillage automatique du petit bout — *Propagation du maillage surfacique*

A section in this abstract gives more information concerning the chosen meshing.
So, thanks to this FEM, models and procedures have to be developed in order to determine the Hertz pressure and the tooth root stress distributions.

Pressure contact model :

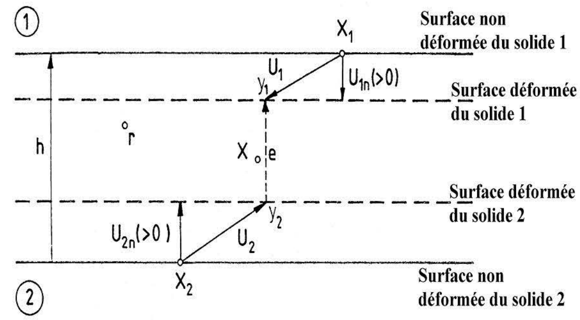
The pressure contact model has to take into account the friction and the sliding phenomenon.

Contact pressure analysis:

So, we have to know at the beginning if under load (with deflexion displacement U1 and U2) what are the pinion surface points which are in contact with the wheel surface points.

The distance between the booth teeth without deflexion is "h".

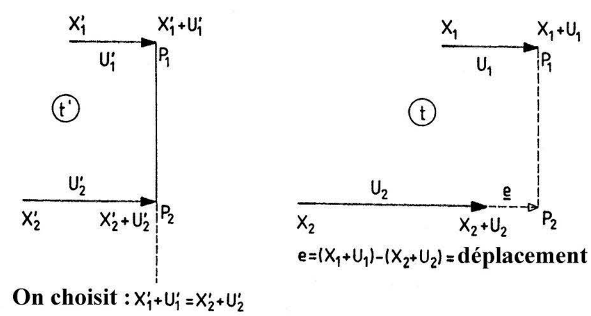
According the e=h-U1-U2 value, we can say if the contact exists or no. In the case where e=0, so a contact pressure P is applied.



Pressure contact model

Sliding analysis:

We have also to know at each time, the pinion and wheel relative position in the deformed state.

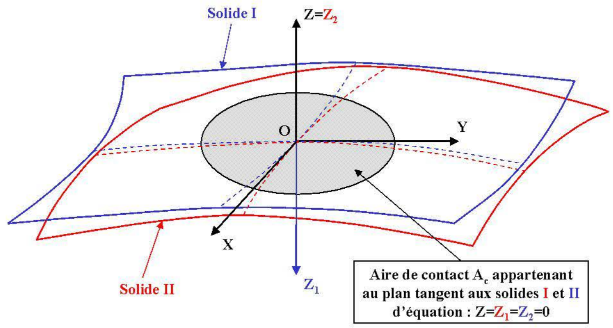


Sliding model

Friction analysis :

The friction is the phenomenon generating a force which opposes to the movement of sliding a solid on the other one. This force is called force of friction. Generally, a given compensatory force is necessary to make slide two solids the one on the other one. So in many examples, the force of friction is constant. Therefore, we make the hypothesis that the force of cutting is limited by the limit force g which depends on the normal force Fz, the amplitude of the speed of gliding V and the other parameters.

Contact Area Model :



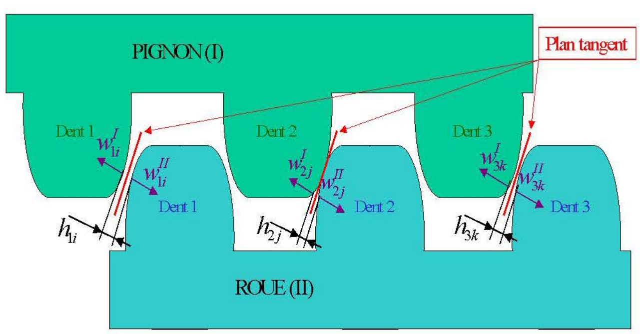
Potential contact area

The calculation of the movements in the potential contact, with the laws of the linear elasticity and with the law of Hooke, uses the combined elastic constants calculated from the elastic constants of both solids.

- The stiffness combined elastic constant G_c .
- The Poisson stiffness combined elastic constant ν_c .
- The difference parameter de K :

<u>Split load distribution:</u>

Now, let us consider the case of the meshing between the pinion and the wheel. In that case, several teeth can be in contact simultaneously as represented on the following figure.



Complete meshing

All the teeth are in contact according to the same normal direction. This normal corresponds to the normal for the tangent plan of the running driving tooth, calculated by the kinematics module of the new SPIRO software for every kinematics position.

Then, for teeth participating potentially in the potential contact, we look if there is a point of the pinion and wheel active side where the normal is identical to that of the tangent plan of the running tooth.

All the couples of teeth Pinion / wheel having a point on the active side participate potentially in the load sharing.

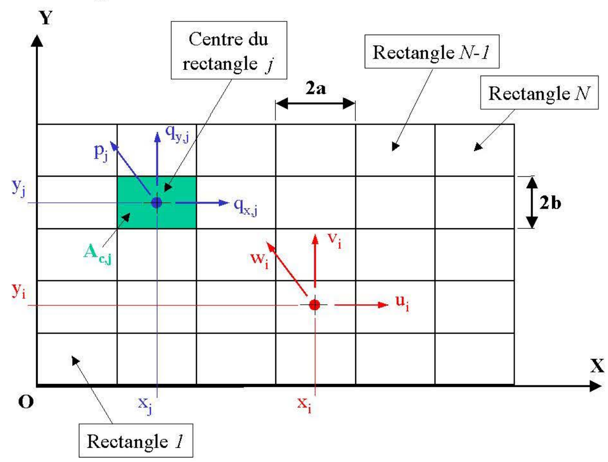
We consider that the gears have the same mechanical characteristic. So, the normal model is uncoupled by the tangential model. The dimension of the contact area is only determined by the normal model. The tangential model supplies us the distribution between the zones of adhesion and sliding within the area of contact. We can study both models of load sharing separately.

The influence contact coefficient calculation :

It is really interesting to know the influence of a load applied on one point of the surface on the others points.

It is the reason why the influence coefficients have to de determined.

The tangent plan where is situated the potential area of contact is divides in N rectangles of constant size (2a*2b) on which the distribution of normal pressure and tangential is considered constant.



According to the tangential load ($q_{x,j}$ and $q_{y,i}$) and the pressure contact P_j, we can obtain the relative displacement (ui, vi and wi) for all the others points inside the potential area of contact.

The following matrix gives these influence contact coefficients.

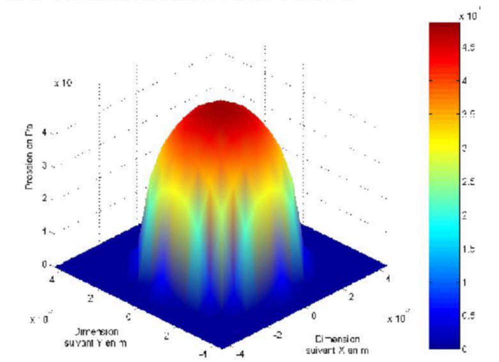
$$\begin{Bmatrix} u_i \\ v_i \\ w_i \end{Bmatrix} = \begin{bmatrix} [A_1] & [B] & 0 \\ [B] & [A_2] & 0 \\ 0 & 0 & [C] \end{bmatrix} \cdot \begin{Bmatrix} q_{x,j} \\ q_{y,j} \\ p_j \end{Bmatrix}$$

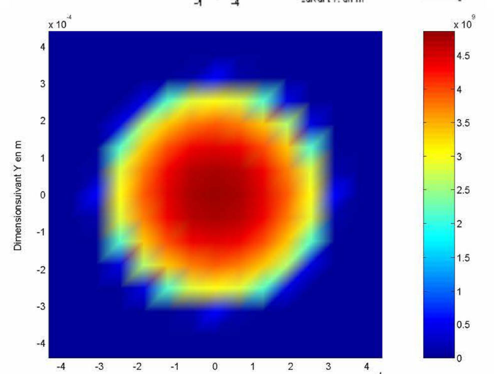
Influence coefficient matrix

Contact model validation :

To perform the validation of the algorithm in the case of a pure normal load, we model Hertzian contacts and we compare the results obtained numerically with the results stemming from the theory of Hertz as well as the results of the works of Hamrock and Brewe.
We made this validation for the contact between two pressing balls and two pressing barrels.

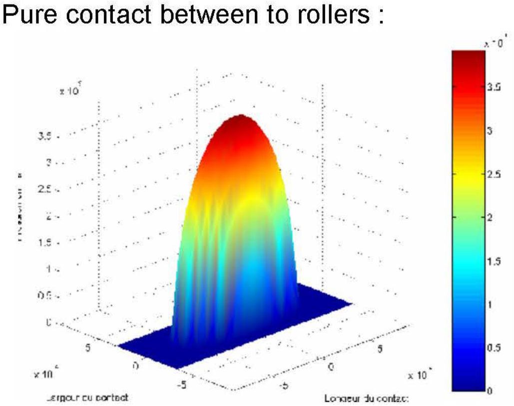
- Pure contact between to balls :

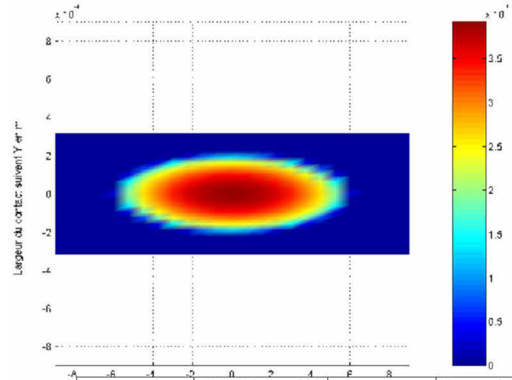




	Résultats avec la théorie de Hertz	Résultats avec les formules de Hamrock et Brewe	Résultats numériques	Erreur relative par rapport à la théorie
Rayon de l'aire de contact a en mm	0.3142	a=b=0.3142	0.3142	0 %
Ecrasement global α en mm	$2.1721 \cdot 10^{-2}$	$2.1721 \cdot 10^{-2}$	$2.1726 \cdot 10^{-2}$	0.023 %
Pression maximale p_{max} en MPa	$4.836 \cdot 10^{3}$	$4.836 \cdot 10^{3}$	$4.855 \cdot 10^{3}$	0.39 %

- Pure contact between to rollers :





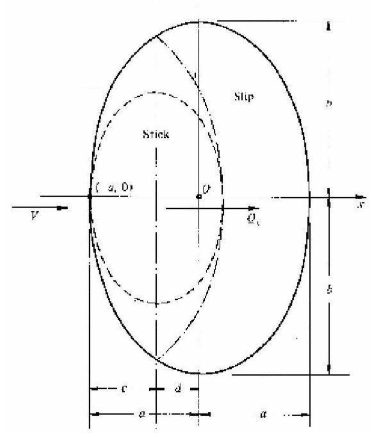
	Résultats avec la théorie de Hertz	Résultats avec les formules de Hamrock et Brewe	Résultats numériques	Erreur relative par rapport à la théorie	
Demi grand axe a en mm	0.6011	0.6243	0.5994	0.28 %	3.99 %
Demi petit axe b en mm	0.2041	0.2225	0.1907	6.57 %	14.29 %
Ecrasement global α en mm	$1.8298 \cdot 10^{-2}$	$1.2610 \cdot 10^{-2}$	$1.8219 \cdot 10^{-2}$	0.43 %	49.81 %
Pression maximale p_{max} en MPa	$3.732 \cdot 10^{3}$	$3.437 \cdot 10^{3}$	$3.916 \cdot 10^{3}$	4.93 %	13.94 %

- Tangential contact between to rollers :

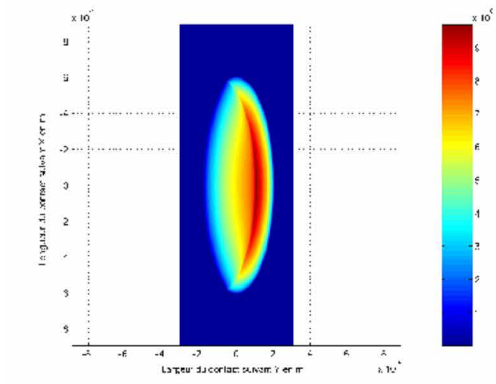
In the case of a tangential contact in movement, the limit conditions for the calculation from the sliding and the resolution of the problem is identical.

As regards this type of contact tangential, the works supply representations of the zone of adhesion and sliding but there is no formula approached as previously. The works of Ollerton and Hatreds, and Johnson allowed to put in evidence that the zone of adhesion has a shape of lemon or ellipses.

Furthermore, one of the sides of the zone of adhesion coincides with the entry of the contact and the other side corresponds to the limit with the zone of sliding. Having validated numerically the results previously, we are going to verify the coherence of the tangential pressure distribution and the shape of the area of adhesion with regard to the conclusion of the works of Ollerton, Hatreds and Johnson.



Theory

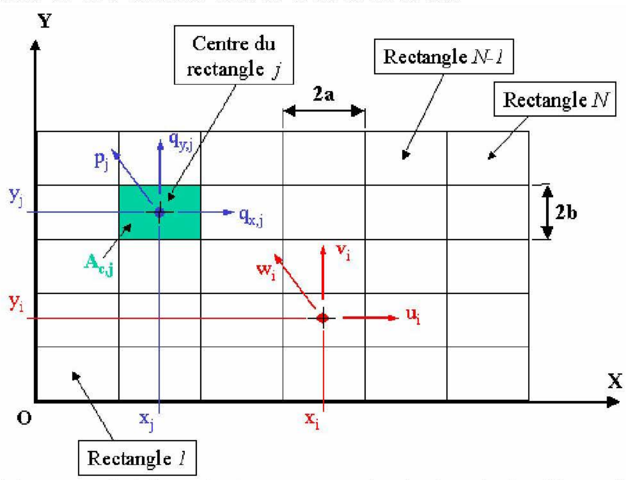


Results

Bending model :

After the pressure influence contact coefficients, we have to calculate the bending influence contact coefficients.
For every kinematics position, the bending coefficients of influences are the movements induced by the unit load of a potential contact point on all the others points of potential contact.

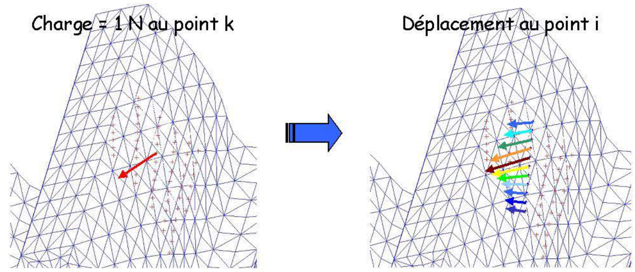
The following Figure shows the normal movements due to the flexion under a normal load.



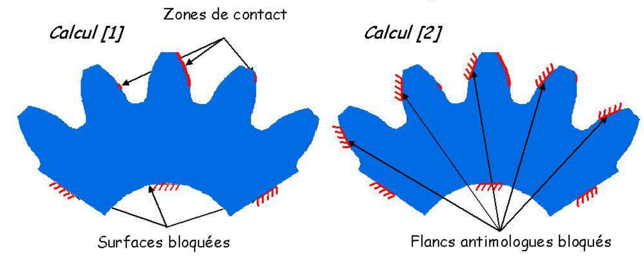
Every point has to be successively loaded with unit load to obtain all the coefficients of influence of flexion:

- Those due to a normal load
- Those due to a tangential load.

We shall detail only the calculation in the case of the normal coefficients of influences of flexion.



Several limit conditions can be analysed :

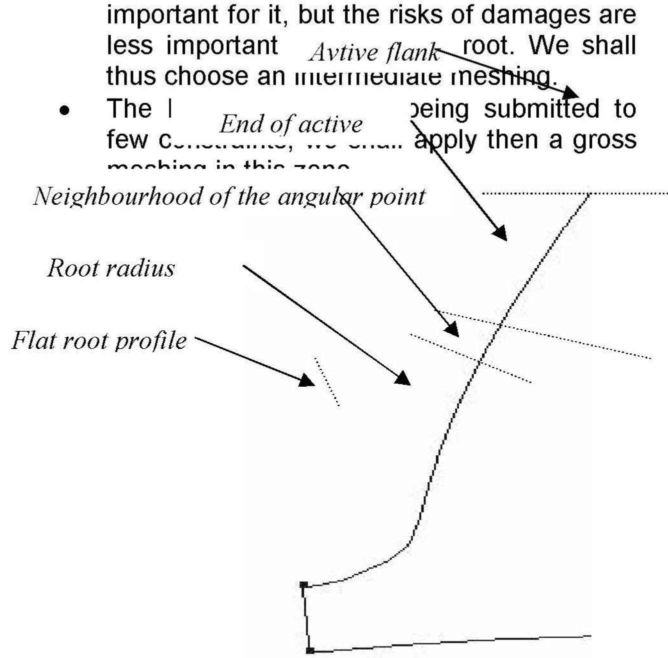


limit conditions

Automatic Meshing algorithm :

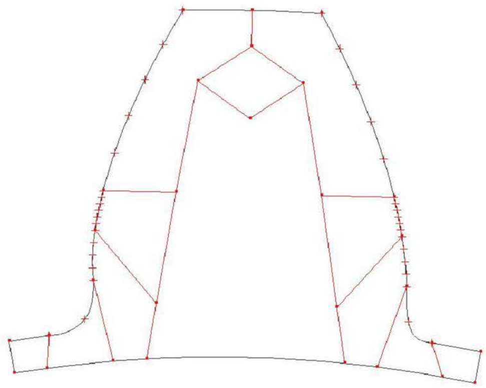
From the meshing generated by CATIA, we modify it by using macro-elements.
It is possible to define zones of meshing, to optimise the calculations.

- In the part geometry, the tooth root is a zone of likely cutting: we shall privilege it a fine meshing.
- The active side of the tooth is the zone of load of the gear where the loads are important for it, but the risks of damages are less important [illegible] root. We shall thus choose an intermediate meshing.
- The l[illegible] being submitted to few c[illegible] apply then a gross meshing in this zone.



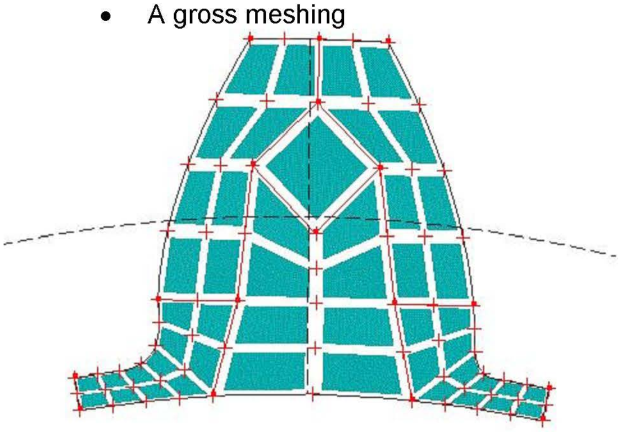
A lot of points are needed in the both neighbourhood of the angular point and root radius.

We saw in the previous paragraphs a breakdown of a profile of tooth according to mechanical, geometrical and technical considerations finished elements. We can then distinguish a maximum of 12 zones being the object of a particular, but dependent meshing some of the others to insure the connection:

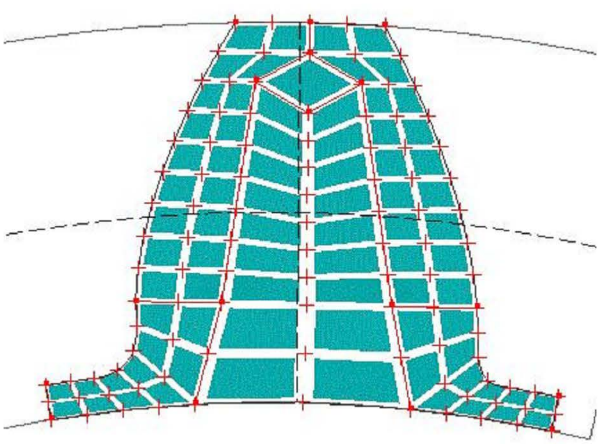


Based on this method, we can create two types of models according to the needs :

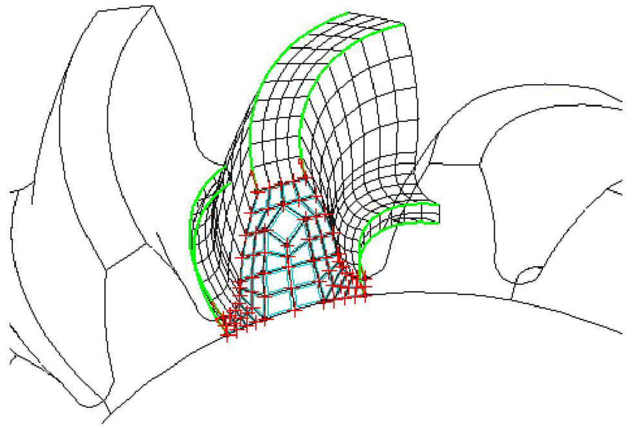
- A gross meshing



- A precise model



At the end we propagate the surface meshing in order to obtain a volumic meshing.



Meshing propagation

Application of this method on a known bevel gear :

A Eurocopter Gear Box bevel gear has been chosen to implement the method :

SPIRO - DONNEES DIMENSION DU PIGNON
335II+81 PO00 PI00 RO00 RI00

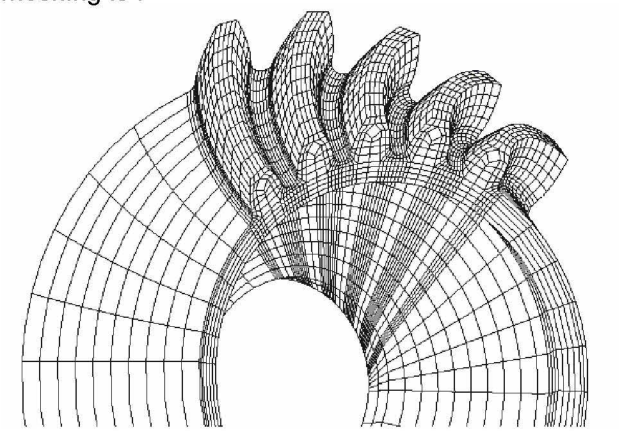
NOMBRE DE DENTS (N)		22
LARGEUR DE DENTURE (E1)	(MM)	61.
DIAMETRE PRIMITIF (DIA1)	(MM)	192.24
SAILLIE (C1)	(MM)	10.43
CREUX (D1)	(MM)	7.75
ANGLE PRIMITIF (APRIM)		27.18
ANGLE DE TETE (ATETE)		30.40
ANGLE DE PIED (APIED)		25.18
UNITE DES ANGLES (AG)	(DM/DD)	DM
SENS DE LA SPIRALE (HAND)	(RH/LH)	RH

SPIRO - DONNEES DIMENSION DE LA ROUE
335II+81 PO00 PI00 RO00 RI00

NOMBRE DE DENTS (N)		45
LARGEUR DE DENTURE (E1)	(MM)	61.
DIAMETRE PRIMITIF (DIA1)	(MM)	393.22
SAILLIE (C1)	(MM)	5.01
CREUX (D1)	(MM)	13.17
ANGLE PRIMITIF (APRIM)		69.42
ANGLE DE TETE (ATETE)		71.42
ANGLE DE PIED (APIED)		66.2
UNITE DES ANGLES (AG)	(DM/DD)	DM
SENS DE LA SPIRALE (HAND)	(RH/LH)	LH

DONNEES SPIRO - DONNEES GENERALE A L'ETUDE 335II+81 P000 PI00 R000 RI00	
ANGLE DE PRESSION (EN DEGRES)	18
ANGLE DE SPIRALE EXT. (EN DEGRES)	30
JEU NORMAL (EN MM)	0.3
NOMBRE DE COLONNES DU MAILLAGE NIO	30
NOMBRE DE LIGNES DU MAILLAGE NIG	20
TYPE D ENGRENAGE (CONIQ OU DROIT)	CONIQ
SENS DE ROT. PIECE MENANTE (CW/CCW)	CCW
VIT.DE ROTATION PIECE MENANTE (TR/MN)	4888
ANGLE ENTRE LES DEUX AXES (EN DEGRES)	97.11
ROTATION DE LA PIECE MENANTE (0)	0
NBRE DE POSITION D'ETU. SUCESS (20)	20
PUISSANCE TRANSMISE (EN KW)	2600

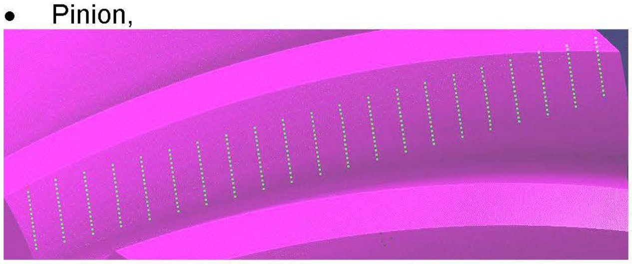
The result in terms of 3D model and volumic meshing is :



We perform also the procedure for the wheel.

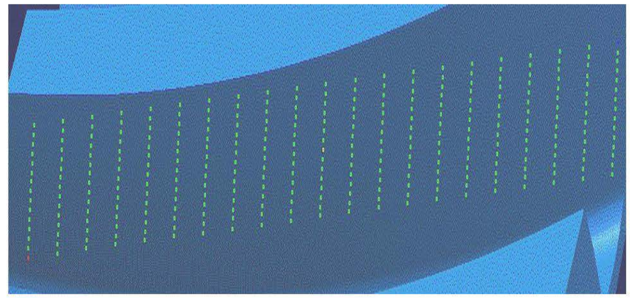
The tooth surface are used to analyse the relative location between the :

- Pinion,



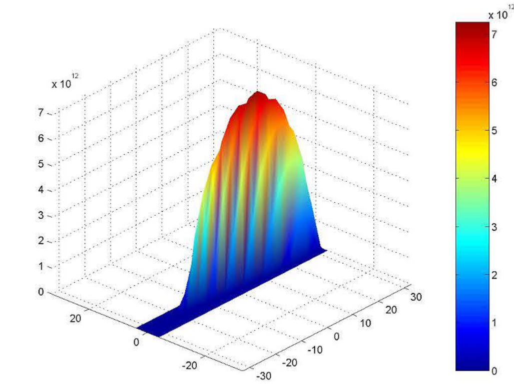
Pinion topography

- And the wheel.

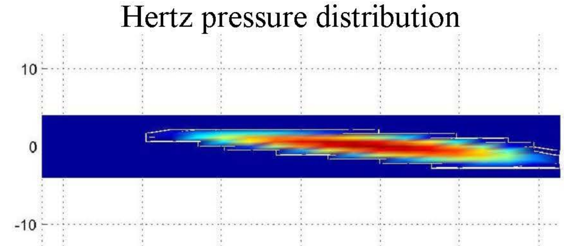


Wheel topography

The contact analysis between the 2th and 3rd teeth gives :



Hertz pressure distribution



Pattern

This pattern is the representation of the contact due to a torque value (Ti). Usually, on a test rig where the maximum torque is applied progressively, we can see the total envelope of the entire contact patterns corresponding to the different torque values.

One another improvement should be to represent the superposition of the all patterns.

Software validation :

Validation criteria chose :

Now, the software has to be calibrate. We have to know if this method gives us a good model of a real bevel gear meshing.

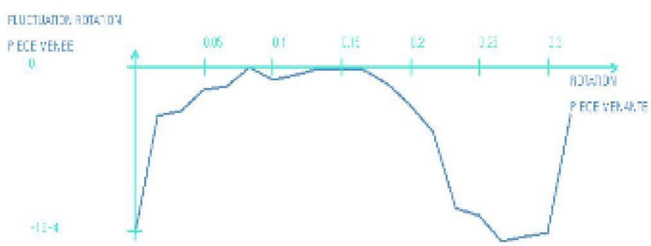
In our point of view, the only global information given by the EC software which allow us to be sure that this model is close of the reality is the "mesh transmission error".

Effectively, the transmission error is influenced by :

- the gears bending deflection due to the tooth and web stiffness..
- the Hertz pressure local deflexion
- the manufacturing default like the pitch error,
- the deflection under load (axial movement, offset and angle axis modification).

Finally, all the improvements introduced in the new SPIRO software have a direct effect on the "mesh transmission error" curve. So, if the curve given by the software and this one obtained during the test are the same, we can consider that the calibration is good.

So, the gearbox bevel gear stage has been measured in terms of topography, pitch error.
All these data have been introduced in the new SPIRO software. As result, we obtain an error transmission diagram.



Transmission error SPIRO curve

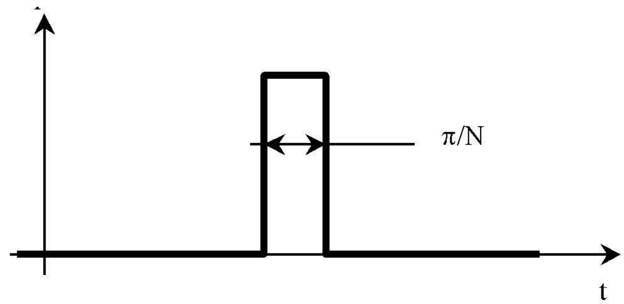
On the other side, a test has been performed on an EC tail gearbox test rig with the previous bevel gear stage. At each instant, the shift between the pinion and the wheel is determined.

<u>Transmission Error Test :</u>

Principle :

The principle of release of a synchronised acquisition is simple and bases on the localisation of a revolution top delivered by the optical converters. This synchronisation requires nevertheless the relatively precise location of the optical converter on the gear.

Most of the optical converters deliver a signal called revolution top which is to the low level (0 volts) except during the passage of a particular line of the converter. The duration of this signal is about half a period of the signal generated by the optical converter.



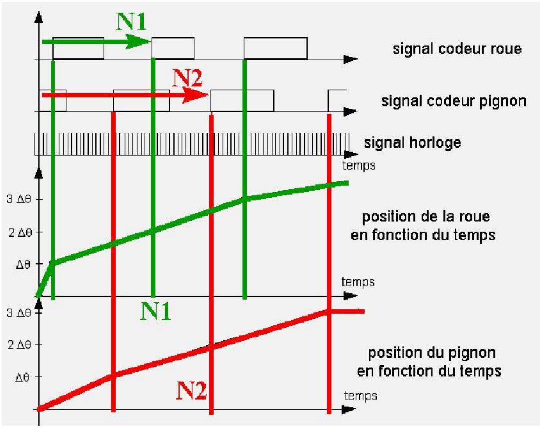
Top tour signal

The synchronisation is made in two steps:

- During the passage of the revolution top of the wheel, we put a logical door to a high state (or low) which allows to specify that in the next passage of the top tour of the pinion, we will begin the acquisition.
- The release of the acquisition is validated when the first top tour of the converter pinion appears. This entry is validated by the fact that the signal of the logical door was validated.

It is possible to measure the error of transmission by means of optical converters having a weak resolution that is a weak number of lines by revolution. The principle of this measure is based on the counting of the number of impulses delivered by a clock with very high frequency between two rising fronts of the signal delivered by the optical converter. This counting has to be simultaneously made on both ways (pinion & wheel) and on same clock reference.

This allows to obtain a robust temporal reference on both ways and to measure the simultaneous difference between both ways. It is possible then to reconstruct the law of evolution of the positions of the wheel and the pinion according to time and it in a rate given by the number of lines on each converter.

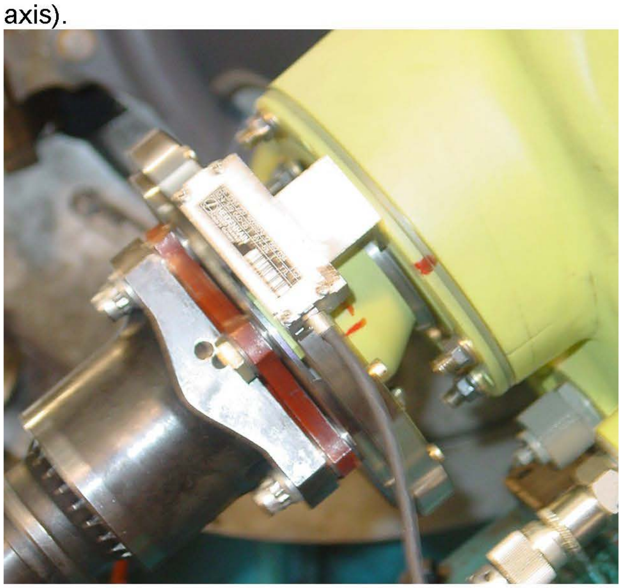


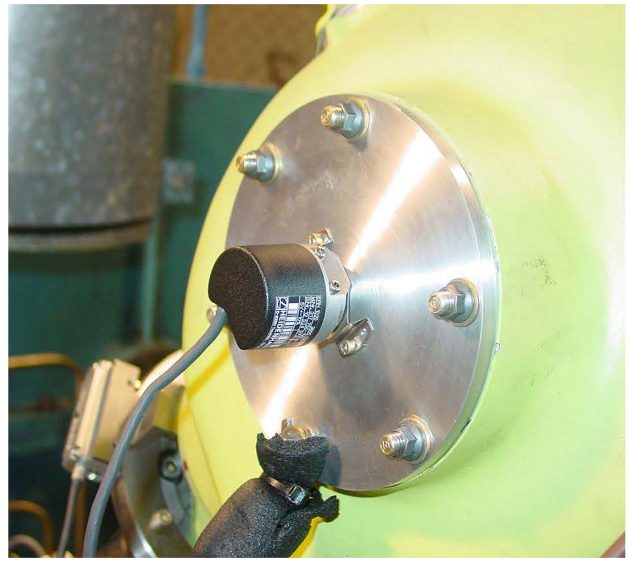
Counting principle

The test rig :

The test rig is an Eurocopter test rig.

The following photos show the optical converters installation (on the pinion axis and on the wheel axis).





Some marks are realised on the pinion external part, the shaft external area and on the housing in order to allow the initial resetting.

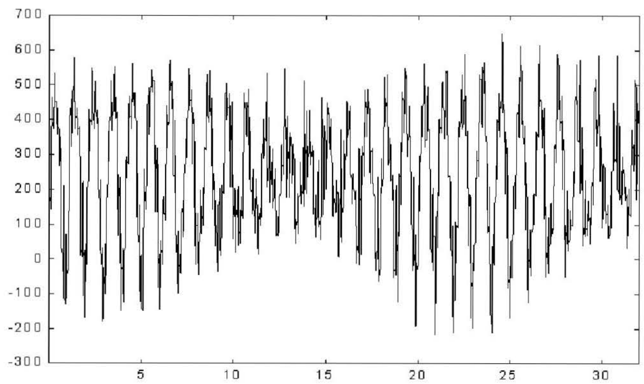
Signal treatment :

The treatment of the signal allows to reconstruct the curve of the error of transmission. Two methods can be used :

- The angular method,
- The synchrony method.

The following figure shows the both transmission error obtained. There are low and high frequency constituents corresponding to two eccentricities on the gears or on the converters, and the passages of teeth, with higher frequency.

This measure is given in arc second of revolutions compared to the high number of pinion revolution



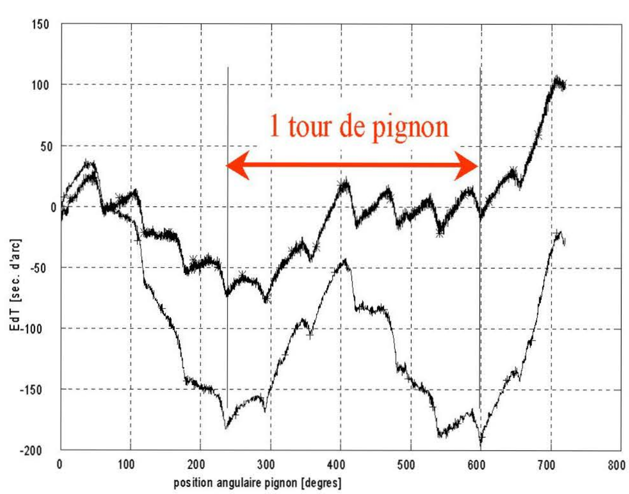
Test rig signal

Principle of the initial resetting :

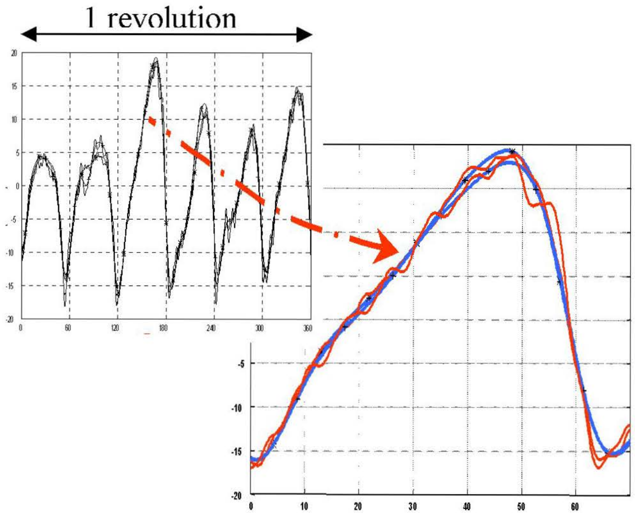
The Error of Transmission so measured is known to a constant near, because the measure is a relative measure of both angular positions of both shafts. On the other hand, this constant will be always identical because the positions of the beginning of acquisition will always be the same if converters are not moved between two experiments.

In other words, the zero of this measure does not indicate necessarily that there is not an error of transmission, but a transmission error which will always be the same.

Another disturbance must be also eliminated, it is about the constituent low frequency representing the effects of the eccentricity. This constituent is eliminated by means of a filter pass-height allowing to reject the constituents low frequency. With regard to a raw signal such as that presented previously, it is possible to eliminate these constituents of eccentricity.



Before treatment



After treatment

This Transmission Error integrates the errors of profile due to the processes of manufacture & assembly and the pitch error due to the deflexion under load and/or manufacture process.

The pitch error taking into account will allow to delay or to advance the meshing entrance of teeth to contact on the transmission error curves.

It's better to keep the complete signal without pitch error and deflexion treatment to compare it with that given by the software to be sure to validate in its entirety the brought improvements.

<u>Conclusion :</u>

This new software integrates the complete shape of the gears. It allows to take into account of the deflexion under load and the manufacture defaults.
A first test has shown that the results on the test rig were close to the software results.
Now, this checking has to be performed on others bevel gear stages.