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CONTACT ANALYSIS FOR SPUR GEARS

CANANAU, S[orin]

Abstract: *This paper presents a development of a numerical procedure for the contact study of spur gears under load. The base of the study is represented by parametric description geometry of both gears and the kinematic contact. The approximation of elastic deformations is made using the finite element method. Using this method is also possible to simulate the operation of gear drives. The engagement of operating surfaces at successive points of the theoretical profile enables to simulate the conjugate action. Numerical results are presented and discussed for some important cases of spur gears contact.*
Key words: *Gear, Contact, Stiffness*

1. INTRODUCTION

One of the important problems of the functioning gears is the meshing contact. Among the inherent difficulties in analyzing a meshing gear contact involve, by one side, the complexity of profile and boundary shape, the coupling of solid body rotation and the evaluation of rolling contact. By other side, the task involves complicated contact models which are not generally available or simply treated in standard commercial numerical analysis codes. One of the important problems is the mesh stiffness along the lines of contact. For each tooth any potential point of contact is credited with a direct flexibility only such that, under load conditions, there is no deflection at any other point of the tooth, (Ajmi & Velex, 2005).

In this paper we are going to purpose a model for spur gear contact. The hypothesis of the model includes the theoretical meshing gear of the involutes external spur gears, the theory of elasticity for deformable solid, the contact along a contact line of the tooth width. Also, a relevant importance is accorded to the calculus of the stiffness. Many researchers are taking into account the face load factor of load distribution along tooth width (Chaari et al., 2005)

We are going also to use the FEM analysis to find the behaviour of elastic bodies such as teeth structure in the gears systems. We are going to use also the FEM model to simulate the manufacturing errors and their effects on the gear mesh stiffness.

Also, we are going to introduce an increasing load as a torque up to the limit of the fracture initiation at the tooth root. In that fallows friction forces are neglected and contact lines are considered parallel to shaft axis

2. GEOMETRICAL AND MATHEMATICAL MODEL

For a gear tooth the modelling principle is to represent the structural elasticity viewed from any line of contact on the flank by a foundation with position varying characteristics to simulate the evolutions of the contact lines during the meshing process. So we use for geometrical purposes a 2D model of spur gear contact (Pimsarn & Kazerounian, 2002). The geometrical model of conjugate action is:

a. The geometry of the gear drive and gear driving is obtained using mathematical formulations for a real case of manufacturing, including trochoid form at the tooth foot region.
b. The 2D model includes the rim geometry with defined ratio parameters of the rim thickness/tooth height more than 2 to 1.
c. The number of the teeth taking into account is 3 to 4, conjugate action, in order to simulate the real process of single and double pair teeth in contact.

Table 1 gives the geometry and mechanical properties of the studied gears.

Concerning the mathematical model of the contact meshing gears we consider that the “contact line “ between the two teeth involved in the meshing process is a segment with non fixed end points, A and B. Also we consider the method of the influence coefficients and we suppose that the load w(x)dx applied in the point P(x) will give an elastic deformation in any point along the contact line AB. In this case, for example in the point P(u) this load will produce a deformation (Schmidt ,1973):

$$df(u) = \alpha(u,x)w(x)dx \tag{1}$$

where α(u,x) is the influence coefficient of the load applied in the point P(x) for the elastic deformation in the point P(u). The model is described in the Fig. 1

In this case the global contact elastic deformation f(u) in the point P(u) is a function of the load distribution along contact line:

$$f(u) = \int_{x_A}^{x_B} \alpha(u,x)w(x)dx \tag{2}$$

This model of load distribution will permit to find the real and continuous distribution along the width of the tooth. The shape of load distribution along the tooth will be smooth using spline transformations.

Geometry /Mechanical properties	Units	Data
Number of teeth, z_1/z_2	-	17/31
Center distance, a	mm	120
Pressure angle, α_n	deg	20
Module, m	mm	5
Diametral pitch, p_n	mm	15.7080
Working face width, b	mm	40
Young modulus, E	N/mm^2	2.07×10^5
Poissons’s ratio, ν	-	0.33

Tab. 1. Materials properties and geometry for gear set

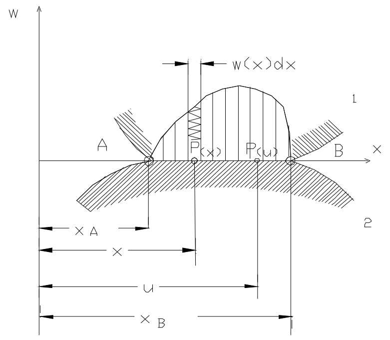

Fig. 1. Load distribution along contact line

with condition of the transmitted load:

$$F_{tot} = \int_{x_A}^{x_B} w(x)dx \tag{3}$$

3. FINITE ELEMENT MODEL

The 2D geometrical model is now associated with finite element description. In terms of global design variables are defined both the structure of drive and driven gear sectors. Mesh generation for each geometrical subdomain takes place following various stages. The relation between conjugate action and finite element structure is established by using predefined points of conjugate contact on each active involute profile of teeth pair in contact, Fig.2.Concerning the 3D load distribution we are going to introduce a global model, which take into account the information of the planar model described in the Fig.2. Such model is widely used and accepted in literatures (Sirichai, 1999). Finite element contact between pinion and wheel tooth pairs is also taking into account, but the hertzian deformations in the contact zone are found to be small relative to the bending deflections. In order to find the singular stiffness of one tooth of the pinion, we will introduce a torque or a distributed force (load) which simulates the action of the meshing tooth of the wheel. Using the equivalent force, this was applied to the tooth flank normal to the involute profile and along the line of action at the appropriate nodes. The representation of the wheel sector is shown in Fig. 3.

4. RESULTS AND DISCUSSIONS

First of all we have to find the dependencies of tooth displacement under load for the location of contact position on the contact line. This dependency is given by real process of contact including single and double pair of teeth contact. In the Fig.4 there is a result concerning the behaviour of elastic tooth under load taking into account a 2D model and a double pairs teeth contact

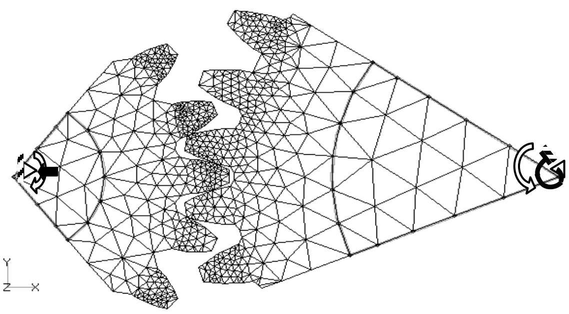

Fig. 2. Finite Element Model and conjugate action

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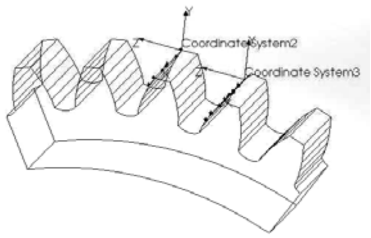

Fig. 3. The 3D CAD model of the sector wheel

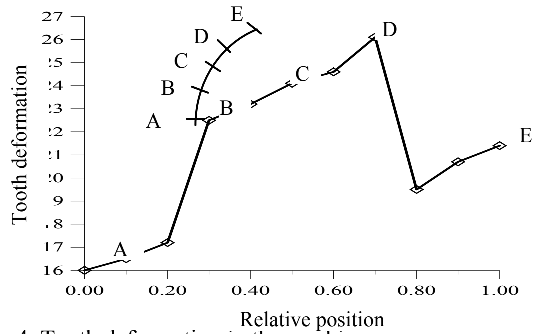

Fig. 4. Tooth deformation in the meshing process

In the same process we take into account the 3D model and the elastic deformation along the contact line, for each line parallel to gear rotation axis. For this analysis we estimate the elastic deformation at ten points along the contact line and for eight lines of contact from the the point A to the point E along the meshing line. The 3 D result for a gear set contact and for a specific line, without misalignement between gears axes is shown in Fig. 5. We can easily onserve the “end effect” regarding the load distribution along contact line.

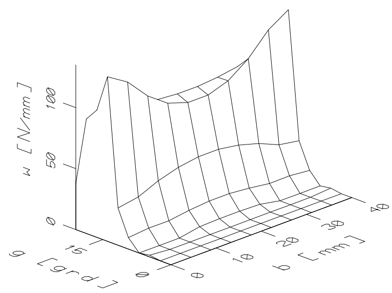

Fig. 5. Load distribution along contact line for gears without misalignment

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