Current Journal of Applied Science and Technology

Volume 42, Issue 47, Page 100-120, 2023; Article no.CJAST.110354
ISSN: 2457-1024
(Past name: British Journal of Applied Science & Technology, Past ISSN: 2231-0843, NLM ID: 101664541)

Mean Wear Approach for Modeling and Predicting Wear for Gears in Plastics Materials and their Composites

Y. N. Agbetossou [a*], A. Afio [a], D. Koffi [b], S. Tiem [c++] and K. A. Kassegne [c++]

[a] Laboratoire de Structures et de Mécanique des Matériaux (LaS2M), Département de Génie Mécanique, Ecole Polytechnique de Lomé, Université de Lomé, 01 B.P. 1515 Lomé 1, Togo.
[b] Institut d'Innovation en Écomatériaux, Écoproduits et Écoénergies (I^2E^3) de l'Université du Québec à Trois-Rivières, CP 500, Trois-Rivières, Québec, G9A 5H7, Canada.
[c] (LaS2M), Département de Génie Mécanique, Ecole Polytechnique de Lomé, Université de Lomé, 01 B.P. 1515 Lomé 1, Togo.

Authors' contributions

This work was carried out in collaboration among all authors. All authors read and approved the final manuscript.

Article Information

DOI: 10.9734/CJAST/2023/v42i474321

Open Peer Review History:
This journal follows the Advanced Open Peer Review policy. Identity of the Reviewers, Editor(s) and additional Reviewers, peer review comments, different versions of the manuscript, comments of the editors, etc are available here: https://www.sdiarticle5.com/review-history/ 110354

Original Research Article

Received: 11/10/2023
Accepted: 15/12/2023
Published: 21/12/2023

ABSTRACT

It is currently recognized by the scientific and industrial world that gears made of plastic materials and their composites have numerous advantages (light weight and inertia reduction, no lubrication or initial lubrication, low friction coefficient, shock and vibration absorbing, good load distribution, low costing manufacturing, etc.) and they will continue to beneficially replace metal gears in a good number of applications in all areas; above all, today the family of plastic materials and their composites is expanding with the development of new eco-plastics and their natural fiber

++ *Professeur Titulaire;*
**Corresponding author: E-mail: niconay@yahoo.fr;*

composites as an alternative for sustainable development. However, the challenge remains to continue research in the field of these plastic gears and their composites in order to overcome the problems that still hamper their use.
The literature reveals that wear constitutes one of the failure modes of gears and in particular it remains the most frequent cause of damage in gears made of plastic materials and their composites. According to the results of experimental work carried out on the wear behavior of plastic gears and their composites, the wear prediction models developed for their metallic counterparts are not applicable to them.
The main objective of this present work is to study the wear behavior of gear teeth made of plastic materials and their composites in order to develop a model of its prediction.
In this paper, a mean wear approach is used to develop a model based on Archard's law for the prediction of wear in gears made of plastic materials and their composites. The model is built on experimental works observations and depends on the pair of materials and the operating conditions of the mesh, as well as the parameters which are determined once and for all from the initial experimental results. The model also takes into account the very significant thermal effect on the wear of plastic gears.
The results from a simulation carried out, using MATLAB software for the pair of HDPE30B materials (HDPE polyethylene composite with 30% birch wood fiber) running under dry conditions, are presented and analyzed. The results are consistent with those of our experimental work and are mainly validated with a relative error below 15% by the latter.
The models developed can already provide solutions to needs on an industrial scale.

Keywords: Plastic and composite gears; wear; prediction; simulation; MATLAB.

1. INTRODUCTION

It is currently recognized by the scientific and industrial world that gears made of plastic materials and their composites have numerous advantages (light weight and inertia reduction, no lubrication or initial lubrication, low friction coefficient, shock and vibration absorbing, good load distribution, low costing, etc.) and they will continue to beneficially replace metal gears in a good number of applications in all areas [1,2,3]; above all, today the family of plastic materials and their composites is expanding with the development of new eco-plastics and their natural fiber composites as an alternative for sustainable development [4]. However, the challenge remains to continue research in the field of these plastic gears and their composites in order to overcome the problems that still hamper their use.

In gear transmissions, there is a relative slip between the two profiles in contact due to the very kinematics of the gear transmission mechanism. This sliding associated with the load and speed causes friction, intense heating and wear of the profiles in contact and impacts the performance of the gears. Wear constitutes one of the failure modes of gears and in particular it remains the most frequent cause of damage in gears made of plastic materials and their composites [5,6].

Different authors have studied and developed wear prediction law models for metallic gears, notably those of Flodin A [7], Onishchenko V [8], Wojnarowski J, Onishchenko V [9] and Wu S Cheng HS [10].

There is also numerous experimental work on plastic-to-metal meshing [12]. On the other hand, there is only a little experimental work on profile wear in plastic gears, notably the work of Mao et al. [12] and that of Kukureka et al. [11]. Recently, new experimental work has been carried out on the wear of teeth of gears made of plastic materials and their composites, namely the work of Singh et al. [13] and that of Herba et al. [14] carried out at I²E³ as part of our own research work.

As for the wear law model for predicting the wear of gears made of plastic materials and their composites, it is only recently that Agbetossou et al. [15] developed an empirical law model. This model predicts well the worn shape of the teeth consistent with that observed by the work of Düzcükoğlu H [16], but its application requires the additional determination of a correction factor Kc for each pair of meshing materials. In addition, recent work by Singh et al. [13] and Herba et al. [14] showed that for plastic gears and their composites, the wear rate decreases when the operating speed increases. Particularly, the work of Herba et al. 14] who studied the wear of a few points of the tooth profile in the same operating condition, obtained the worn shape identical to that obtained by Mao et al. [12] and finds that the wear rates for these different points oscillate closely around their average value.

Our analysis of all these experimental results led us to the following observations:

- there are random phenomena of non-uniform material removal in the wear process of polymeric materials, but their effects converge into an average value in terms of total loss of volume or mass;
- there is mild (or medium) wear and severe wear which are delimited by a critical load with the geometric parameters of meshing combined, above which we switch to severe wear, Mao et al. [12];
- for materials whose internal cohesion forces are high, in the case of mild or medium wear, the worn shape of the tooth profile is consistent with that observed by the work of Düzcükoğlu H [16];
- for any type of material in the case of severe wear and for materials with low internal cohesion effort even in the case of mild or medium wear, all points of the tooth profile have a wear depth practically identical whose values oscillate around the average of their values, Herba et al. [21];
- the same model can be adaptable to both types of wear (mild and severe), with different coefficients, determined for each case in its load zone.

These observations lead us to develop a new model based on the mean wear approach for predicting the profile wear of gear teeth made of plastic materials and their composites. This work includes firstly, a theoretical study of the modeling of the wear law, then a numerical simulation is carried out using MATLAB software. The results from the simulation are presented and analyzed.

2. METHORDS

2.1 Theoretical Study of the Modeling of the Wear Law

2.2.1 Choice of the wear law model for gears made of polymeric materials

2.2.2.1 The shape of the worn profile of gears made of polymeric materials

The literature reports numerous works relating to experimental studies of the wear of both metallic and plastic gears. This work shows that two types of tests are often carried out: the simulated meshing test using double discs and the real meshing test on a gear test bench [7,12,16,11,17]. Tests on a gear test bench make it possible to obtain the actual worn profile of the tooth. The wear rate in terms of profile depth loss per meshing cycle clearly reflects the shape of the worn profile and is what is widely used in this type of wear test. Two methods are used to measure the wear rate: real-time measurement by the machine and measurement after shutdown, cooling and cleaning. This second method gives greater precision.

For metal gears, we observe that the worn profile respects the analytic law, of maximum wear at the top and at the root of the tooth, and practically zero wear at the pitch point [18]. For gears made of polymeric materials, the wear is not zero at the original point and the worn profile appears as shown in the work of Düzcükoğlu H [16] in Fig. 1. The shapes of the worn profile shown by the work of Mao et al. [12] Fig. 2, show two cases: that of less severe operating conditions (Fig. 2 a) which is identical to Düzcükoğlu and that of severe operating conditions which practically gives uniform wear of all points of the worn part of the profile of the tooth (Fig. 2 b).

Fig. 3 shows a schematic diagram of the evolution of tooth profile change due to wear (a) and the method of measuring tooth profile wear of plastic gears (b).

2.2.2.2 Choice of the wear law model for gears made of polymeric materials

Conventionally the phenomenon of wear is quantified by the volume or mass of material removed. But for the case of gears, knowing the depth worn per cycle is more useful. This is why, from the initial formulation of Archard's law, global or local expressions of the law of usury have been used by different authors. Flodin and Anderson [19], developed this model, to calculate the wear depth (h) at each point on the surface of spur and helical teeth in the form:

$$\frac{dh}{ds} = k.P \Rightarrow h = k.\int_0^s P.ds = k.\int_0^t P.v_s.dt$$

With:

k, the wear coefficient (m²/N) (k=$\frac{K}{H}$).
P: Contact pressure (N/m²)
s: Sliding distance (m)
v_s: Sliding speed (m/s)
t: Sliding time (s)

It appears from our bibliographic study that the wear model of Flodin and Anderson [19] derived from Archard's law is more appropriate for gears, due to the fact that it makes it possible to calculate the depth of wear of each point on the contact profile and that the results of the simulations also agree with the results of the experimental tests.

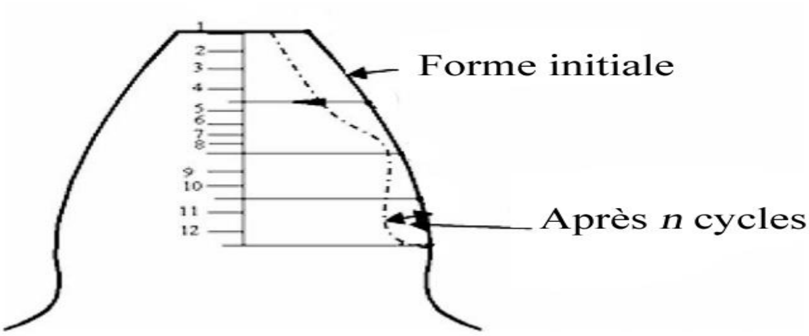

Fig. 1. Change in tooth shape with wear [11].

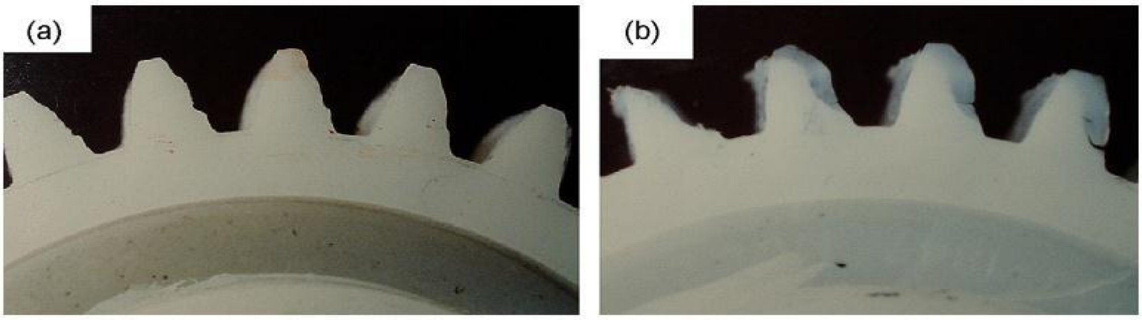

Fig. 2. Acetal gear wear forms under loads of (a) 7 Nm and (b) 10 Nm [9]

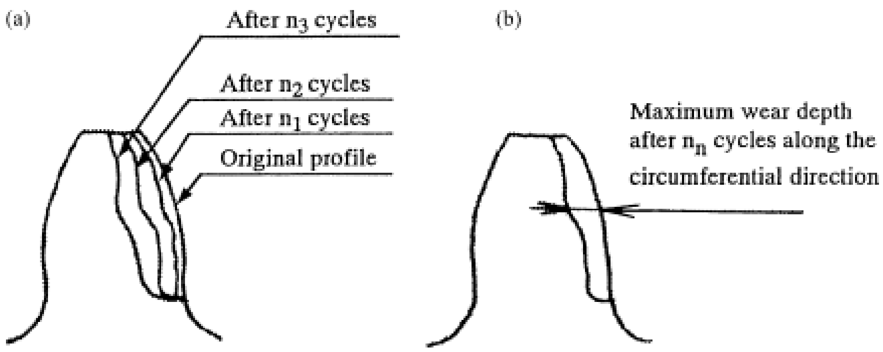

Fig. 3. Schematic drawings of tooth profile change (a) and measurement of wear depth (b) [17]

We will therefore stay in line with the most common models in the literature (in particular those of Flodin and Anderson [18,19] by also using an Archard law.

We will develop a new wear model still based on Archard's law by replacing the wear coefficient k by the complacency $1/E^{\lambda}$ to take into account the very sensitive effect of elastoviscoplastic materials to temperature. In the presence of lubricant, we will use the equivalent E which will be determined using rheological models.

In order to be able to make comparisons, we will also adapt the Flodin model to plastic gears by determining the sliding distance differently with a new approach [20] which gives a non-zero sliding distance at the pitch point.

2.2 Theoretical Analysis of the Wear Law Modeling

Starting from our hypothesis that the complacency 1/E affected by an exponent, can replace the wear coefficient k in the Archard wear model adapted to gears, and according to our observations drawn from the analysis of the experimental results, we formulate our models as follows:

2.2.1 Adaptation of the Flodin and Anderson model to gears made of polymeric materials

For its adaptation to gears made of polymeric materials, we formulate it as follows:

$$h_i = k_{my0} P_i S_i \left(\frac{P_{my}}{P_{my0}}\right)^{cphk} \left(\frac{\omega_0}{\omega}\right)^{cvk} \quad (1a)$$

$$h_{my} = \frac{\sum_{i=1}^{n} h_i}{n} \quad (1b)$$

$$k_{my0} = \frac{\sum_{i=1}^{n} k_i}{n} \quad (1c)$$

With:

h_i: Wear depth per cycle (m) of the considered point of the profile
h_{my}: The average value of h_i
k_i: The wear coefficient (m²/N) of the considered point of the profile without taking into account the effect of temperature on the Young's modulus E
k_{my0}: The average value of k_i
P_i: Contact pressure (N/m²) of the considered point of the profile taking into account the effect of temperature on the Young's modulus E
S_i: Sliding distance (m) for the considered point of the profile
n: The number of profile points chosen for the calculation
P_{my}: The average of the contact pressure of the calculation points, without taking into account the effect of temperature on the Young's modulus E
P_{my0}: The average of the contact pressure of the calculation points for the experimental conditions used to determine the model coefficients, without taking into account the effect of temperature on the Young's modulus E
ω_0 : Rotation speed (in radians) for the experimental conditions used to determine the model coefficients
ω : Rotation speed (in radians) for the operation considered
cphk: Influence coefficient of the effect of the operating contact pressure to be determined using experimental results
cvk: Coefficient of influence of the operating rotation speed to be determined using experimental results
$\left(\frac{P_{my}}{P_{my0}}\right)^{cphk}$ Accounts for the increase in the wear rate with that of the load
$\left(\frac{\omega_0}{\omega}\right)^{cvk}$ Accounts for the decrease in the wear rate with increasing speed.

2.2.2 Study of a new model based on Archard's law for predicting the wear of gears made of polymeric materials

We formulate our new model as follows:

$$h_i = \frac{P_i S_i}{E_0^{\lambda}} \left(\frac{P_{my}}{P_{my0}}\right)^{cph} \left(\frac{\omega_0}{\omega}\right)^{cv} \quad (2a)$$

$$h_{my} = \frac{\sum_{i=1}^{n} h_i}{n} \quad (2b)$$

With:

E_0: The equivalent modulus of elasticity (N/mm²) of the pair of meshing materials without taking into account the effect of temperature
h_i: Depth of wear per cycle (m) of the considered point of the profile
h_{my}: The average value of h_i
λ: Exponential wear coefficient (non-dimensional) to be determined experimentally

P_i: Contact pressure (N/m^2) of the considered point of the profile taking into account the effect of temperature on the Young's modulus E
S_i: Sliding distance (m) for the considered point of the profile
n: The number of profile points chosen for the calculation
P_m: The average of the contact pressure of the calculation points, without taking into account the effect of temperature on the Young's modulus E
P_{my0}: The average of the contact pressure of the calculation points for the experimental conditions used to determine the model coefficients, without taking into account the effect of temperature on the Young's modulus E
ω_0 : Rotation speed (in radians) for the experimental conditions used to determine the model coefficients
ω: Rotation speed (in radians) for the operation considered
cph: Influence coefficient of the effect of the operating contact pressure to be determined using experimental results
cv: Coefficient of influence of the operating rotation speed to be determined using the experimental results

$\left(\frac{P_{my}}{P_{my0}}\right)^{cph}$ Accounts for the increase in the wear rate with that of the load

$\left(\frac{\omega_0}{\omega}\right)^{cv}$ Accounts for the decrease in the wear rate with increasing speed.

2.3 Procedure for Determining the different Parameters of the Model

2.3.1 Study of the determination of the exponential wear coefficient λ

The coefficient λ is determined by experimental method. The experimental method consists of obtaining the wear depths (h_i) of a few points of the profile by the wear test on the gear test bench. Then, we determine the values of the contact pressures (P_i) and the sliding distance (S_i) of these points of the profile by calculation and then we deduce the value of the coefficient λ using the average wear value. λ will then be unique for each pair of meshing materials in its zone of validity.

2.3.2 Study of the determination of the coefficient ki

There are two methods for determining the wear coefficient (k) which are the experimental method and the statistical regression method.

The experimental method consists of obtaining, with the meshing parameters of the experimental test, the wear depth (h_i) of a few points of the profile by the wear test on the gear test bench. Then, we determine the values of the contact pressures (P_i) and the sliding distance (S_i) of these points of the profile by calculation and then we deduce the values of the wear coefficient (k_i) of each of the points on the profile. From these values of (k_i) determined experimentally, we can then determine the value of (k_i) for any point of the profile for this pair of meshing materials.

The statistical regression method consists of carrying out a statistical analysis of the different parameters which affect the wear coefficient to develop an empirical formula based on them. Thus Janakiraman et al. [20] carried out static analyzes of the parametric effects of load, speed, lubrication and roughness of contact surfaces on wear and proposed an approximate formula for the coefficient wear (k) as follows:

$$k = \frac{3.981\times10^{29}}{E} \times L^{1.219} \times G^{-7.377} \times S^{1.589} \quad (3)$$

$$L = \frac{W_0}{ER} \quad (4)$$

$$G = \alpha_k E \quad (5)$$

$$S = \frac{R_\alpha^c}{R} \quad (6)$$

$$\frac{1}{E} = \frac{1}{2}\left[\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_1^2}{E_2}\right] \quad (7)$$

$$\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2} \quad (8)$$

$$R_\alpha^c = \sqrt{R_{\alpha1}^2 + R_{\alpha2}^2} \quad (9)$$

With:

L: the dimensionless load.
G: the dimensionless pressure-viscosity coefficient.
S: dimensionless amplitude of roughness.
E_1, E_2, E respectively the modulus of elasticity of the material of the pinion, the wheel and the equivalent modulus of elasticity (N/mm^2).

v_1 and v_2 are the Poisson's ratios of the pinion and wheel materials.
W_0: the normal load per unit width (N/mm).
R_1 and R_2 the radii at the point of contact (mm).
$R_{\alpha 1}$ and $R_{\alpha 2}$ are the roughness of the two surfaces in contact (mm).
α_k: the pressure-viscosity coefficient with α_k =1 for meshing without lubrication.

For the method of statistical regressions in the case of gears made of polymeric materials, the question is to know which roughness values (before or after wear) should be used in the calculations. For this reason, we used a hybrid experimental-analytical method, where after having experimentally determined the wear rate for selected points of the profile, we use the formula of statistical regressions to deduce the values of the wear coefficients k_i from these points interactively by varying the value of the roughness (becoming aware of the roughness indices) so that the calculated wear rate is equal to the experimental wear rate.

2.3.3 Study of the determination of the sliding distance S_i

The sliding distance will be determined by the new approach based on an analytical-numerical method for calculating the sliding distance of the points of the tooth profile during meshing, which we have previously developed in Agbetossou et al. [19]. This method gives a non-zero sliding distance at the pitch point.

2.3.4 Study of the determination of the contact pressure P_i

In the numerous works which have been carried out on the wear of different types of gears, the determination of the contact pressure (p) is made either by the Hertzian contact method or by the finite element method. As for the value of (P), many authors have used a static (p) and only a few authors have used a dynamic (p). Zhang J and Xianzeng L [21] reported in their literature review a number of authors who used one or other of the aforementioned methods.

As part of our study, we will use a static contact pressure determined by the Hertzian method. So after taking into account the load distribution factor taken up by a tooth in the case of plastic meshing [22,23], we obtain the maximum value of the Hertzian contact pressure by the formula:

$$P_{imax} = \frac{2W_i}{\pi b_i l} \tag{10}$$

W_i is the load transmitted along the line of action to the main tooth meshing at position Spn(i);
b_i is the half Hertz contact width;
l is the width of the tooth.

During sliding, the value of the contact pressure varies from zero at the moment of entry into contact, then increases to the maximum value at the theoretical contact position of the point considered, and then decreases to the zero value at the moment of exit from the contact zone.

We have: $$P_i = P_{imax}\sqrt{1 - \left(\frac{x}{b_i}\right)^2} \tag{11}$$

With x which varies from $-b_i$ to $+b_i$.
As the value of the half width of Hertz is very small, we have, in the very close vicinity of $-b_i$ and b_i, the values of P_i which are practically equal to the value of P_{imax} (P_i>0.99P_{imax}). If we also consider the impact effect of coming into contact, we can then work with P_{imax} on the entire $2b_i$ contact zone. This has been the practice by several authors who have dealt with the subject of gear wear, and as part of our study, we will also work with the contact pressure of Hertz P_{imax}.

2.4 Simulation

2.4.1 Data and characteristic curves used for the simulation

For this simulation, we exploit the experimental results of the wear test on a gear test bench (Tables 2 to 4) for the HDPE30B composite [14] carried out at I^2E^3-UQTR as part of our work on new HDPE/Birch composite materials (HDPE, HDPE30B, HDPE40B). Our UQTR wear tests give us the wear rates of 5 points on the tooth profile, namely the points of the head, pitch, foot, a point between head and pitch, finally a point between pitch and foot. For the experimental method, given that the admissible torque for a 40% .m fiber gear rolling at a given speed of 500 rpm is 12.5 N.m [24], we chose to lower the margin of torques to be applied by half of the admissible torque to retain four values (2Nm, 4Nm, 5Nm and 6 Nm) in order to be able to play on three rotation speeds (500 rpm, 1000 rpm and 1500 rpm). The duration of each test is four hours [11]. The ambient temperature in the room where the tests take place varies between 20 and 24°C. The geometric and operational data are summarized in Table 1.

We use the curve of variation of the modulus E as a function of the temperature of the new HDPE material with 30% birch with a frequency of 20Hz (Fig. 4). This curve was obtained experimentally by the DMA method [24].

We notice that the DMA curve of HDPE30B (30B3 20Hz), has a linear variation in the temperature range from 25°C to 120°C. This curve can be modeled by the line passing through the points A (40°C, 3800 MPa) and B(100°C, 1260 MPa) whose equation is:
$E = -42.33T_f+5493.2$ (12) T_f en °C.

2.4.2 Method for determining equilibrium and instantaneous temperatures

Research in the field of thermal study of plastic meshing has been refined over the years to such an extent that thermal models commonly used today have taken into account all the parameters that affect meshing such as those of the geometry, the material and those of the operating conditions. This being said, these models give satisfactory prediction results which are in agreement with the experimental results. In this present work we have chosen the numerical method of calculating temperatures from Koffi et al. [22] and translated the original FORTRAN program into MATLAB.

2.4.3 The simulation procedure

From the geometric and operational data used to conduct the experimental tests, we determine the distribution of the equilibrium temperature in the tooth and the evolution of the instantaneous temperature on the profile as a function of the normalized positions using the program for calculating temperatures. The calculated instantaneous temperatures make it possible to determine the values of the elastic modulus as a function of the temperature and the normalized position, using the equation of the DMA curve.

Table 1. Geometric and operational data for the wear test at I^2E^3 on testing bench [14]

Module	**2.54mm**
Number of tooth	30 (for the two gears)
Pressure angle	20°
Tooth wide	6.5 mm
Torque	2 ; 4 ; 5 et 6Nm
Rotation speed	500 ; 1000 and 1500rpm
Contact ratio	1.785
Material 1	HDPE + (0% ; 30% and 40% Birch)
Material 2	HDPE + (0% ; 30% and 40% Birch)

Table 2. Wear rate per cycle in µm for the torque 2Nm, HDPE30B [14]

Torque	**Normalized position**	**500 rpm**		**1000 rpm**		**1500 rpm**	
		Driven	**Driver**	**Driven**	**Driver**	**Driven**	**Driver**
2Nm	1	0.00031573	0.00020126	0.0001678	9.9659E-05	0.00016846	0.00016409
	2	0.0003199	0.00031506	0.00017996	0.00011027	0.00017393	0.00017975
	3	0.00029877	0.00028182	0.00021989	0.00015115	0.00018574	0.00015918
	4	0.00013642	0.00032192	0.00016164	9.9758E-05	0.00014734	0.00015491
	5	0.00014004	0.0001075	0.00013678	6.6349E-05	0.00015551	0.00010768
Average		0.00024217	0.00024551	0.00017322	0.00010544	0.0001662	0.00015312

Table 3. Wear rate per cycle in µm for the torque 5Nm, HDPE30B [21]

Torque	**Normalized position**	**500 rpm**		**1000 rpm**		**1500 rpm**	
		Driven	**Driver**	**Driven**	**Driver**	**Driven**	**Driver**
5Nm	1	0.00143705	0.00102396	0.00077712	0.00065465	0.00053274	0.00045146
	2	0.00143896	0.00095448	0.00078732	0.00065592	0.0005364	0.00046097
	3	0.00144179	0.00095907	0.00081588	0.00062154	0.00056486	0.00045815
	4	0.00136326	0.000953	0.00072317	0.00052139	0.00051148	0.00044854
	5	0.00137816	0.00084596	0.00070492	0.00049939	0.00050075	0.00043217
Average		0.00141185	0.00094729	0.00076168	0.00059058	0.00052925	0.00045026

Table 4. Wear rate per cycle in µm for the torque 6Nm, HDPE30B [21]

Torque	Normalized position	500 rpm		1000 rpm		1500 rpm	
		Driven	Driver	Driven	Driver	Driven	Driver
6Nm	1	0.00159412	0.0011404	0.00113039	0.00071187	0.00080619	0.00065905
	2	0.00162785	0.00110403	0.00118893	0.00076035	0.00081739	0.00064587
	3	0.00162645	0.00115759	0.00118823	0.00078713	0.00081783	0.00066135
	4	0.00164313	0.00103491	0.00111323	0.00072162	0.00079414	0.0006146
	5	0.00148358	0.0009439	0.00111679	0.00072195	0.00077009	0.00061198
Average		0.00159503	0.00107616	0.00114751	0.00074058	0.00080113	0.00063857

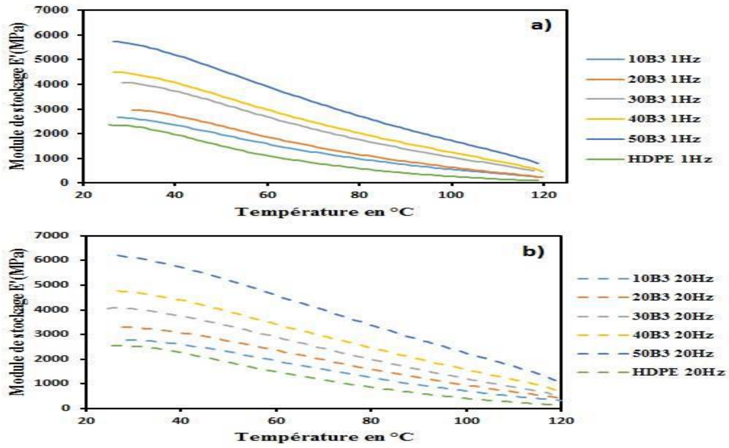

Fig. 4. Storage modulus E' with different wood fiber (birch) rates for frequencies a) 1Hz and b) 20Hz as a function of temperature

The normal pressure P_i and the sliding distance Si are also calculated for each normalized position. The exponential wear coefficient λ is determined with the experimental average wear value and is assumed to be constant for all normalized positions.

The simulation flowchart is summarized in Fig. 5.

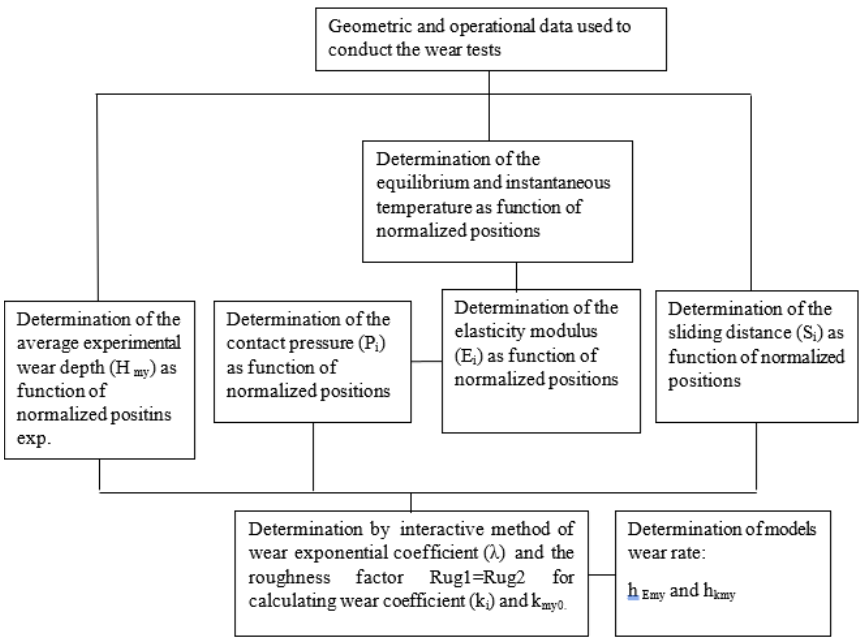

Fig 5. Flowchart for the simulation

After all parameters have been calculated, the values are fed into the model formulas to simulate wear across the entire tooth profile.

3. RESULTS AND DISCUSSION

In this section, we present the main results that come from our simulations and their analyses.

3.1 Determination of Parameters and Results Obtained

For each pair of materials (HDPE30B/HDPE30B) the roughness index for model k and the exponential wear coefficient λ for model E are determined with the average experimental wear values for the 5Nm torque and the rotation speed 1000 rpm, then another torque (6Nm or 2Nm) is used to determine the cph and cphk coefficients and finally, another speed (500 or 1000 rpm) is used to determine the cv and cvk coefficients. The determination is made interactively (adjustment method) on MATLAB software. Once the model parameters are determined from an experimental value, the model is able to predict the wear for that pair of meshing materials within its range of validity. The zone of validity can be the entire zone of mild or severe wear, or split into two or three intervals for greater accuracy of the model. The zone of validity is delimited by a margin of value of the contact pressure obtained for an entire geometric and operational operating condition combined.

For the pair of HDPE30B materials, the parameters determined using the experimental results (Tables 2 to 4) are those in Tables 5 and 6 for the driven gear. The parameters in Table 6 give greater accuracy of the models because the validity zone is split in two. The parameters for the driving gear are those in Table 7.

Table 5. The values of the parameters for the whole unique zone: HDPE30B, driven

Model k :	Model E :
Rug_1= 0.001887	λ= 2.8865
Rug_2= 0.001887	cph= 1.95
cphk= 1.95	cv= 0.55
cvk= 0.55	
k_{my0}=8.9322e-11	

For the thermal behavior in operation, Fig. 6 shows the evolution according to the normalized positions, of the equilibrium and instantaneous temperatures on the profiles of the teeth of the driving gear and the driven gear. These results are consistent with those obtained by Koffi et al. [23].

Table 6. The values of the parameters for the zone split in two: HDPE30B, driven

Model k : 5Nm	Model E : 5Nm	Model k : 2Nm	Model E : 2Nm
Rug_1= 0.001887	λ= 2.8865	Rug_1= 0.003075	λ= 2.9255
Rug_2= 0.001887	cph= 1.95	Rug_2= 0.003075	cph= -1.3
cphk= 1.95	cv= 0.55	cphk= -1.3	cv= 0.25
cvk= 0.55		cvk= 0.25	
k_{my0}=8.9322e-11		k_{my0}=6.5301e-11	

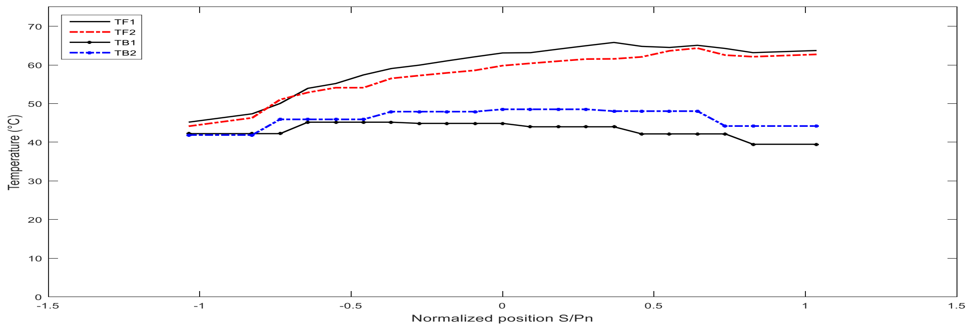

Fig. 6. Evolution according to the normalized position of the equilibrium temperatures TB1, TB2 and the instantaneous temperatures TF1, TF2 for the driver and the driven gears: HDPE30B/HDPE30B ; C = 5Nm (Phmoy =34.33 MPa), ω = 1000rpm

Table 7. The values of the parameters for the zone split in two: HDPE30B, driver

Model k : 5Nm	Model E : 5Nm	Model k : 2Nm	Model E : 2Nm
Rug_1= 0.001561	λ= 2.9240	Rug_1= 0.002296	λ= 2.9835
Rug_2= 0.001561	cph= 1.35	Rug_2= 0.002296	cph= 0.25
cphk= 1.35	cv= 0.35	cphk= 0.25	cv= 0.45
cvk= 0.35		cvk= 0.45	
k_{my0}=6.6889e-11		k_{my0}=4.1050e-11	

Fig. 7 shows the distribution of the load (torque and Hertz pressure) according to the normalized positions which is specific to the meshing of plastics and their composites Koffi et al. [23]. Likewise, we see the variation of the sliding distance following the normalized positions which presents a non-zero sliding distance at the pitch point Agbetossou et al. [15]. This figure finally shows the effect of temperature on the equivalent Young's modulus following the normalized positions, as well as the variation in the wear coefficient k.

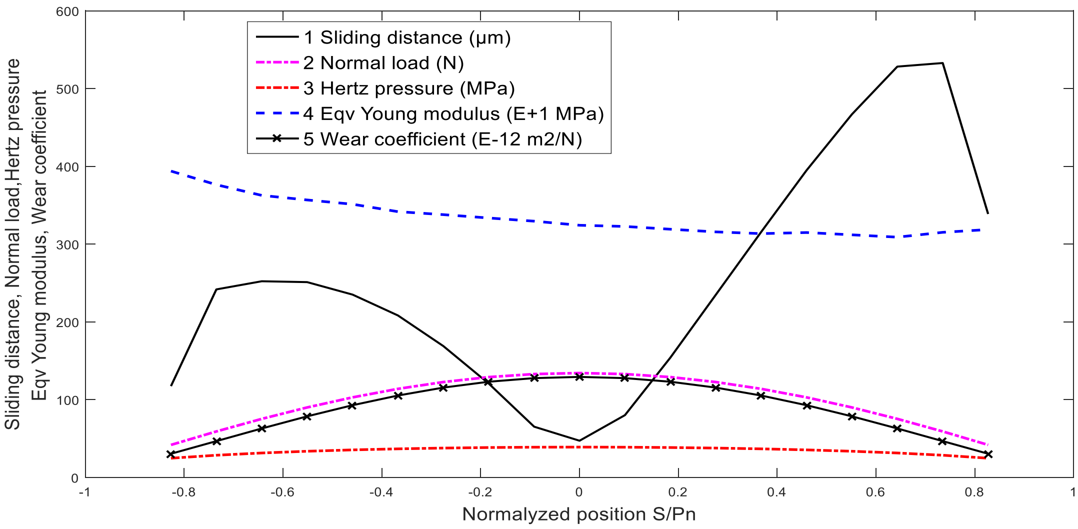

Fig. 7. Evolution according to the normalized position of the sliding distance, the normal load, the Hertz pressure, the equivalent Young's modulus, and the wear coefficient of the driver gear: HDPE30B/HDPE30B; C = 5Nm, ω = 1000rpm ; exponential wear coefficient λ = 2.8865

3.2 Results on the Driven Gear

Table 8 presents the average values (experimental and models) of the wear rate on the driven gear. These results show that the two models predict practically the same values. Likewise, we notice that the values of the wear rate predicted by the models have the same trends as those experimental (the wear rate increases when the torque increases, but decreases when the speed increases).

To assess the difference between the experimental values and the model values of the wear rate, we calculated the relative errors.

Table 9 presents the relative errors of the two models for a torque of 5Nm and a speed of 500 rpm for the driven wheel. For the average values, we have a relative error of 11.73% for model k and a relative error of 11.76% for model E. For individual values, the greatest difference is observed at the pitch point for the two models with 14.10% for model k and 14.13% for model E.

Fig. 8 presents the results graphically for a torque of 5 Nm and a speed of 500 rpm. Curve 1 (wear h2k) is the theoretical trend of the variation of the wear rate on the driven wheel following the normalized positions of the points of the profile, according to the adapted Flodin model (model k). Curve 2 (wear h2E) is the theoretical trend of the variation in the wear rate on the driven wheel following the normalized positions of the profile points, according to the new model (model E). We notice on the graph that these two curves practically coincide. Curve 3 (H2kmy wear), which is a straight line, represents the average of the values of curve 1 and it is this which predicts uniform wear by the adapted Flodin model. Likewise, curve 4 (H2Emy wear), which is a straight line, represents the average of the values of curve 2 and it is this which predicts uniform wear by the new model. Logically, curves 3 and 4 are practically identical. Curve 5 (H2myexp) represents the experimental average value of the wear rate on the driven wheel and curve 6 (h2exp), represents the experimental values of the wear rate for individual points on the driven wheel. For the torque of 5 Nm and the speed of 500 rpm, we notice that the experimental values are all above the average value predicted by the models, but with a difference less than 14.13% relative error.

For 5 Nm torque and 1000 rpm speed, the results are similar to 5 Nm torque and 500 rpm speed. The relative error calculation is presented in Table 10 and the graphical presentation of the results is made in Fig. 9. For the average values, we have a relative error of 3.65% for model k and a relative error of 3.62% for model E. For individual values, the largest difference is observed at the root point for both models with 10.83% for model k and 10.80% for model E.

For the torque of 5 Nm and the speed of 1000 rpm, we notice in Fig. 9 that the experimental values oscillate around the average value predicted by the models, with a difference less than 10.83% relative error.

For 5 Nm torque and 1500 rpm speed, the results are similar to 5 Nm torque and 500 rpm speed. The relative error calculation is presented in Table 11 and the graphical presentation of the results is made in Fig. 10. For the average values, we have a relative error of 7.26% for model k and a relative error of 7.35% for model E. For individual values, the largest difference is observed at the root point for both models with 12.36% for model k and 12.34% for model E.

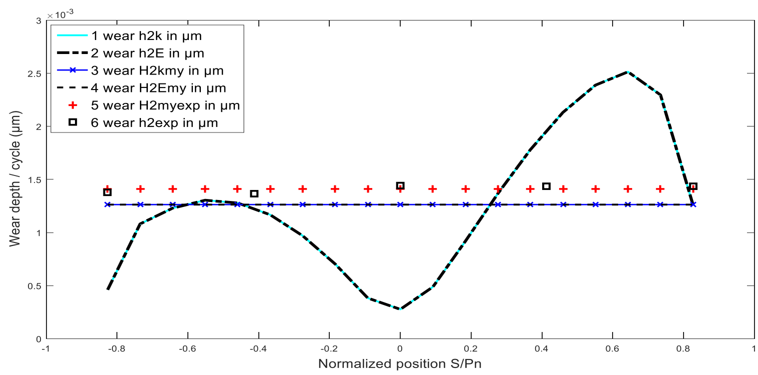

Fig. 8. Evolution of the wear rate experimental and models according to the s/pn positions, driven gear wheel, torque 5Nm, speed 500 rpm

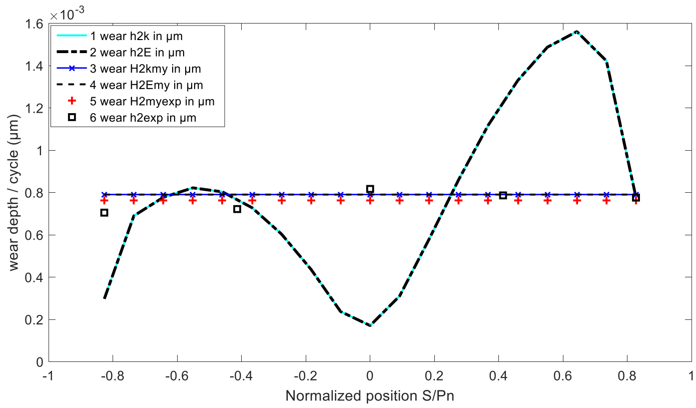

Fig. 9. Evolution of the wear rate experimental and models according to the s/pn positions, driven gear wheel, torque 5Nm, speed 1000 rpm

Table 8. Average wear rate on driven gear, HDPE30B (μm/cycle)

Torque	Experiment			Model k			Model E		
	500rpm	1000rpm	1500rpm	500rpm	1000rpm	1500rpm	500rpm	1000rpm	1500rpm
2 Nm	2.42E-04	1.73E-04	1.66E-04	2.28E-04	1.82E-04	1.57E-04	2.28E-04	1.82E-04	1.57E-04
5 Nm	1.41E-03	7.62E-04	5.29E-04	1.26E-03	7.91E-04	5.71E-04	1.26E-03	7.90E-04	5.71E-04
6 Nm	1.60E-03	1.15E-03	8.01E-04	1.63E-03	9.97E-04	6.93E-04	1.63E-03	9.97E-04	6.93E-04

Table 9. Relative errors for the two models, torque 5Nm, speed 500 rpm, driven gear

Torque	Position	Wear in μm/cycle for driven gear, speed 500 rpm				
		Experiment	**Model k**	**Er %**	**Model E**	**Er %**
5 Nm	1	0.00143705	0.00126363	13.75%	0.00126326	13.76%
	2	0.00143896	0.00126363	13.88%	0.00126326	13.91%
	3	0.00144179	0.00126363	14.10%	0.00126326	14.13%
	4	0.00136326	0.00126363	7.88%	0.00126326	7.92%
	5	0.00137816	0.00126363	9.06%	0.00126326	9.10%
	Average	0.00141185	0.00126363	11.73%	0.00126326	11.76%

Table 10. Relative errors for the two models, torque 5Nm, speed 1000 rpm, driven gear

Torque	Position	Wear in μm/cycle for driven gear, speed 1000 rpm				
		Experiment	**Model k**	**Er %**	**Model E**	**Er %**
5Nm	1	0.00077712	0.00079051	1.69%	0.00079028	1.67%
	2	0.00078732	0.00079051	0.40%	0.00079028	0.37%
	3	0.00081588	0.00079051	3.21%	0.00079028	3.24%
	4	0.00072317	0.00079051	8.52%	0.00079028	8.49%
	5	0.00070492	0.00079051	10.83%	0.00079028	10.80%
	Average	0.00076168	0.00079051	3.65%	0.00079028	3.62%

Table 11. Relative errors for the two models, torque 5Nm, speed 1500 rpm, driven gear

Torque	Position	Wear in µm/cycle for driven gear, speed 1500 rpm				
		Experiment	**Model k**	**Er %**	**Model E**	**Er %**
5Nm	1	0.00053274	0.00057139	6.76%	0.00057122	6.74%
	2	0.0005364	0.00057139	6.12%	0.00057122	6.10%
	3	0.00056486	0.00057139	1.14%	0.00057122	1.11%
	4	0.00051148	0.00057139	10.48%	0.00057122	10.46%
	5	0.00050075	0.00057139	12.36%	0.00057122	12.34%
	Average	0.00052925	0.00057139	7.26%	0.00057122	7.35%

For the torque of 5 Nm and the speed of 1500 rpm, we notice in Fig. 10 that the experimental values are all below the average value predicted by the models, but with a difference less than 12.36% relative error.

For 6 Nm torque and 500 rpm speed, the results are similar to 5 Nm torque and 500 rpm speed. The relative error calculation is presented in Table 12 and the graphical presentation of the results is made in Fig. 11. For the average values, we have a relative error of 2.38% for model k and a relative error of 2.35% for model E. For individual values, the largest difference is observed at the root point for both models with 9.20% for model k and 9.18% for model E.

For the torque of 6 Nm and the speed of 500 rpm, we notice in Fig. 11 that the experimental values oscillate around the average value predicted by the models, with a difference of less than 9.20% relative error.

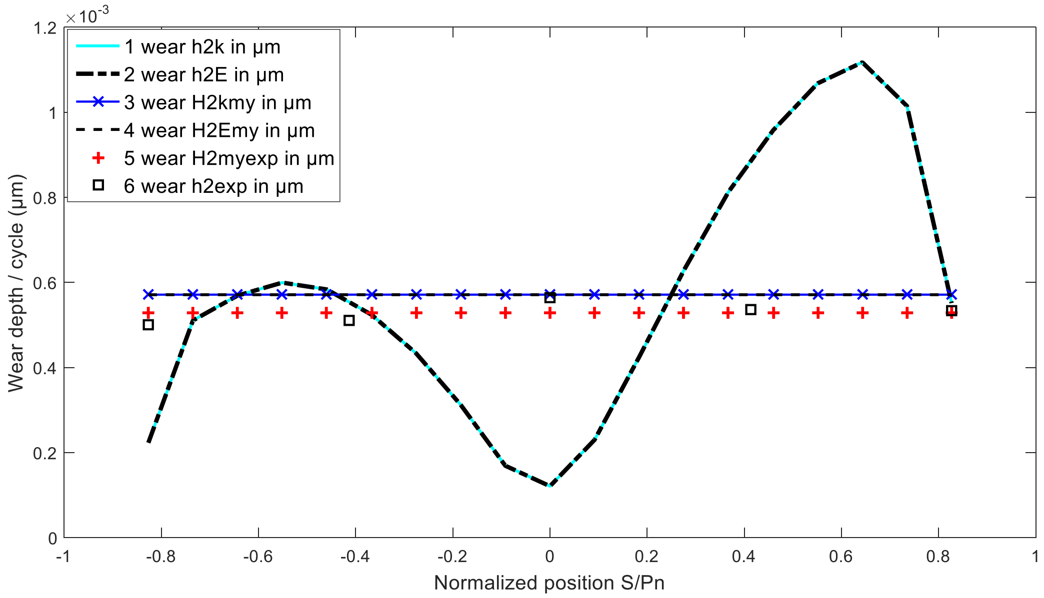

Fig. 10. Evolution of the wear rate experimental and models according to the s/pn positions, driven gear wheel, torque 5Nm, speed 1500 rpm

Table 12. Relative errors for the two models, torque 6Nm, speed 500 rpm, driven gear

Torque	Position	Wear in µm/cycle for driven gear, speed 500 rpm				
		Experiment	**Model k**	**Er %**	**Model E**	**Er %**
6 Nm	1	0.00159412	0.00163393	2.44%	0.00163346	2.39%
	2	0.00162785	0.00163393	0.37%	0.00163346	0.33%
	3	0.00162645	0.00163393	0.46%	0.00163346	0.43%
	4	0.00164313	0.00163393	0.56%	0.00163346	0.59%
	5	0.00148358	0.00163393	9.20%	0.00163346	9.18%
	Average	0.00159503	0.00163393	2.38%	0.00163346	2.35%

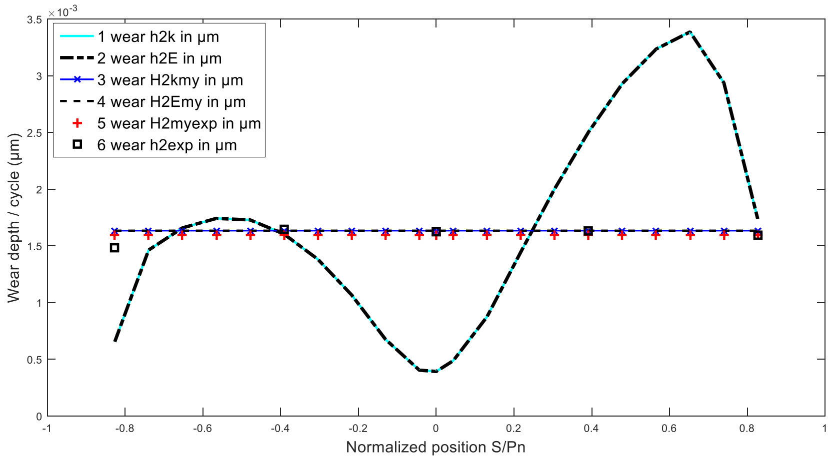

Fig. 11. Evolution of the wear rate experimental and models according to the s/pn positions, driven gear wheel, torque 6Nm, speed 500 rpm

For 2 Nm torque and 1000 rpm speed, the results are similar to 5 Nm torque and 500 rpm speed. The relative error calculation is presented in Table 13 and the graphical presentation of the results is made in Fig. 12. For the average values, we have a relative error of 5.02% for model k and a relative error of 5.05% for model E. For individual values, the greatest difference is observed at the point between the head and the pitch point for both models with 25.00% for model k and 25.02% for model E. For this operating condition (low torque and low speed), we notice that the individual values of the wear rates oscillate around the experimental average value, but are further away from it (Fig. 12) in comparison to the other operating conditions. In addition to the above, the wear rate values per cycle are very low (low torque) and this explains the relative errors up to 25%. For any operating condition, after a large number of cycles, the absolute errors will remain of the same order as per one cycle, then the relative errors will become very small.

For the torque of 2 Nm and the speed of 1000 rpm, we notice in Fig. 12 that the experimental values oscillate around the average value predicted by the models, but with a difference less than 25.02% relative error.

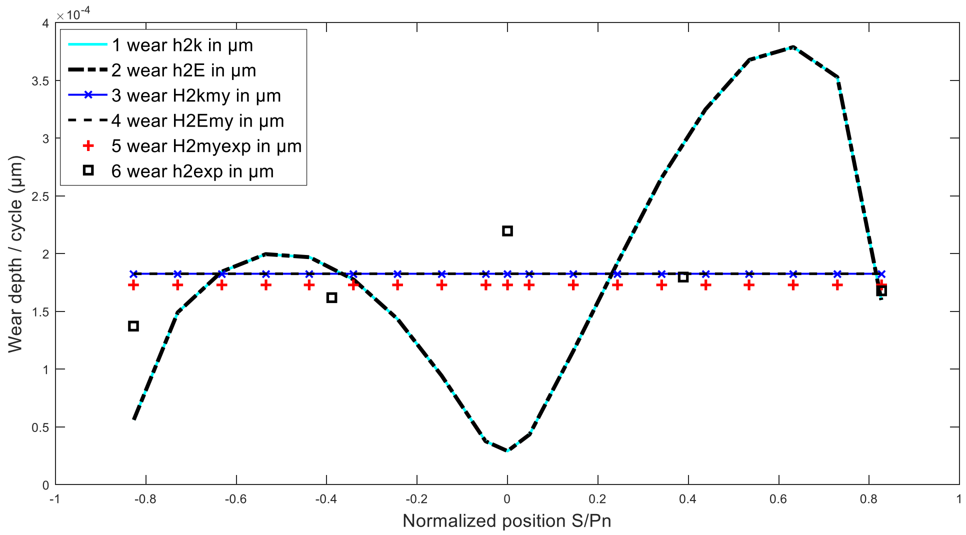

Fig. 12. Evolution of the wear rate experimental and models according to the s/pn positions, driven gear wheel, torque 2Nm, speed 1000 rpm

Table 13. Relative errors for the two models, torque 2Nm, speed 1000 rpm, driven gear

Torque	Position	Wear in µm/cycle for driven gear, speed 1000 rpm				
		Experiment	Model k	Er %	Model E	Er %
2 Nm	1	0.0001678	0.00018237	7.99%	0.00018243	8.02%
	2	0.00017996	0.00018237	1.32%	0.00018243	1.35%
	3	0.00021989	0.00018237	20.57%	0.00018243	20.53%
	4	0.00016164	0.00018237	11.37%	0.00018243	11.40%
	5	0.00013678	0.00018237	25.00%	0.00018243	25.02%
	Average	0.00017322	0.00018237	5.02%	0.00018243	5.05%

3.3 Results on the Driving Gear

The results on the driving wheel are similar to those on the driven wheel. Table 14 presents the relative errors of the two models for a torque of 5Nm and a speed of 500 rpm for the driving wheel. For the average values, we have a relative error of 15.84% for model k and a relative error of 15.78% for model E. For individual values, the largest difference is observed at the head point for both models with 25.21% for model k and 25.15% for model E.

Fig. 13 presents the results graphically for a torque of 5 Nm and a speed of 500 rpm for the driving wheel. Curve 1 (wear h1k) is the theoretical trend of the variation of the wear rate on the driving wheel following the standardized positions of the points of the profile, according to the adapted Flodin model (model k). Curve 2 (wear h1E) is the theoretical trend of the variation of the wear rate on the driving wheel following the standardized positions of the profile points, according to the new model (model E). We notice on the graph that these two curves practically coincide. Curve 3 (wear H1kmy), which is a straight line, represents the average of the values of curve 1 and it is this which predicts uniform wear by the adapted Flodin model. Likewise, curve 4 (H1Emy wear), which is a straight line, represents the average of the values of curve 2 and it is this which predicts uniform wear by the new model. Logically, curves 3 and 4 are practically identical. Curve 5 (H1myexp) represents the experimental average wear rate value on the driving wheel and curve 6 (h1exp) represents the experimental wear rate values for individual points on the driving wheel. For the torque of 5 Nm and the speed of 500 rpm, we note that the experimental values are all above the average value predicted by the models, but with a difference less than 25.21% relative error.

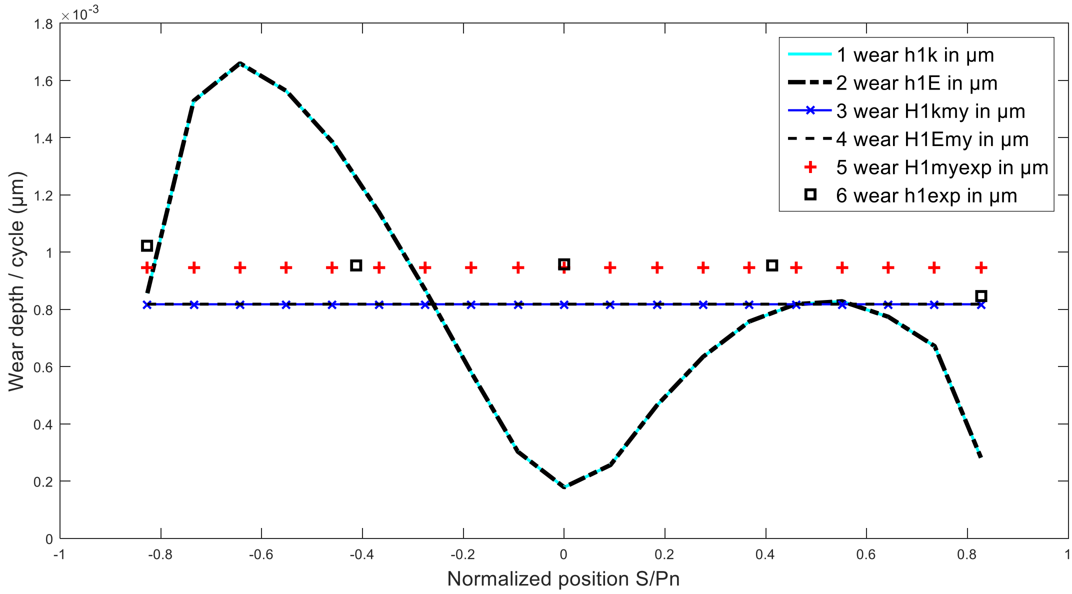

Fig. 13. Evolution of the wear rate experimental and models according to the s/pn positions; driver gear wheel; torque 5Nm; speed 500 rpm

Table 14. Relative errors for the two models, torque 5Nm, speed 500 rpm, driver gear

Torque	Position	Wear in µm/cycle for driver gear, speed 500 rpm				
		Experiment	Model k	Er %	Model E	Er %
5 Nm	1	0.00102396	0.00081779	25.21%	0.00081818	25.15%
	2	0.00095448	0.00081779	16.71%	0.00081818	16.66%
	3	0.00095907	0.00081779	17.28%	0.00081818	17.22%
	4	0.000953	0.00081779	16.53%	0.00081818	16.49%
	5	0.00084596	0.00081779	3.44%	0.00081818	3.40%
	Average	0.00094729	0.00081779	15.84%	0.00081818	15.78%

For 5 Nm torque and 1000 rpm speed, the results are similar to those of 5 Nm torque and 500 rpm speed of the driver wheel. The relative error calculation is presented in Table 15 and the graphical presentation of the results is made in Fig. 14. For the average values, we have a relative error of 0.06% for model k and a relative error of 0.02% for model E. For individual values, the largest difference is observed at the root point for both models with 15.39% for model k and 15.43% for model E.

For the torque of 5 Nm and the speed of 1000 rpm, we notice in Fig. 14 that the experimental values oscillate around the average value predicted by the models, with a difference less than 15.43% relative error.

For 5 Nm torque and 1500 rpm speed, the results are similar to those of 5 Nm torque and 500 rpm speed of the driver wheel. The relative error calculation is presented in Table 16 and the graphical presentation of the results is made in Figure 15. For the average values, we have a relative error of 3.21% for model k and a relative error of 3.26% for model E. For individual values, the largest difference is observed at the root point for both models with 7.10% for model k and 7.14% for model E.

For the torque of 5 Nm and the speed of 1500 rpm, we notice in figure 15 that the experimental values are all slightly below the average value predicted by the models, with a difference of less than 7.14% relative error.

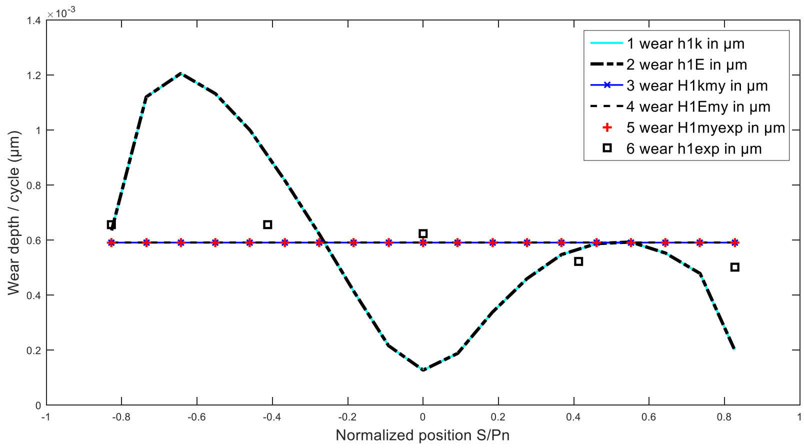

Fig. 14. Evolution of the wear rate experimental and models according to the s/pn positions; driver gear wheel; torque 5Nm; speed 1000 rpm

Table 15. Relative errors for the two models, torque 5Nm, speed 1000 rpm, driver gear

Torque	Position	Wear in µm/cycle for driver gear, speed 1000 rpm				
		Experiment	Model k	Er %	Model E	Er %
5 Nm	1	0.00065465	0.00059020	10.92%	0.00059048	10.88%
	2	0.00065592	0.00059020	11.14%	0.00059048	11.08%
	3	0.00062154	0.00059020	5.31%	0.00059048	5.26%
	4	0.00052139	0.00059020	11.66%	0.00059048	11.70%
	5	0.00049939	0.00059020	15.39%	0.00059048	15.43%
	Average	0.00059058	0.00059020	0.06%	0.00059048	0.02%

Table 16/ Relative errors for the two models, torque 5Nm, speed 1500 rpm, driver gear

Torque	Position	Wear in µm/cycle for driver gear, speed 1500 rpm				
		Experiment	Model k	Er %	Model E	Er %
5 Nm	1	0.00045146	0.00046520	2.95%	0.00046542	3.00%
	2	0.00046097	0.00046520	0.91%	0.00046542	0.97%
	3	0.00045815	0.00046520	1.52%	0.00046542	1.56%
	4	0.00044854	0.00046520	3.58%	0.00046542	3.63%
	5	0.00043217	0.00046520	7.10%	0.00046542	7.14%
	Average	0.00045026	0.00046520	3.21%	0.00046542	3.26%

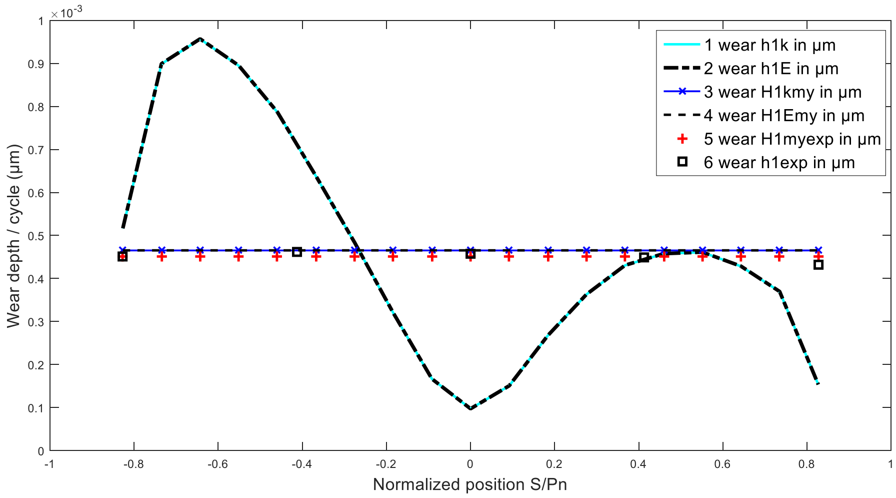

Fig. 15. Evolution of the wear rate experimental and models according to the s/pn positions; driver gear wheel; torque 5Nm; speed 1500 rpm

3.4 Analysis of the Results

In this section, we analyze the results on the determination of the model parameters, followed by those of the results obtained.

3.4.1 Analysis of model characteristics

First of all, we recall that the literature review revealed that serious sizing of gears made of plastic materials and their composites must always take into account the very sensitive thermal effect on their operation, Koffi [1]. This thermal effect particularly affects the wear phenomenon, therefore, our first concern was to identify which material parameter would translate this thermal effect into the models. Thus the Young's modulus is identified and we can take into account the effect of the temperature for its determination in three levels where it appears in the models, the calculation of the sliding distance and the Hertz contact pressure for both models, then for the determination of the wear coefficient ki (k_{my0}) and the exponential wear coefficient λ.

Our numerical algorithms have taken this aspect into account, but our numerous numerical simulations have shown, for very small variations in temperature, sudden divergences of the models between themselves on the one hand and in relation to the experimental results on the other hand. Finally, the effect of temperature is taken into account only for the calculation of the Hertz pressure, which means that both models are stable and predict practically the same results for any operating condition considered.

The second important aspect of the models is that they predict the average wear and therefore the total volume removed. This allows the models to account for the result of random phenomena during the wear process without being affected by their effects. Furthermore, the ideal worn shape from the work of Düzcükoğlu [16], is rarely observed in cases of real operation, but rather it is the worn shape practically identical for all points along the tooth profile which occurs, observes Herba et al. [21]. As a result, the models validly predict the appropriate wear rate during real operation, which makes it possible to predict the lifespan for wear failure or to predict the lifespan for generalized thermal failure.

Finally, a last aspect of the models is their versatility due to the fact that they adapt to different cases of wear (mild and severe wear, wear of the driving wheel and wear of the driven wheel, split load zone) by a specific determination of the coefficients.

3.4.2 Analyzes of the results obtained

Before assessing the results obtained by the models, let us note the following observations on the conformity of the experimental results with the literature review. As reported in our bibliographic study, the experimental work of Singh et al. [13] on POM, HDPE and ABS gears showed that the wear rate for gears made of plastic materials decreases with increasing speed, but increases with increasing load. Simulated disc tests by Manai F [25] showed that under the same operating conditions, wear is more considerable on the driven wheel. Our models are formulated to predict wear rates in this logic, therefore any deviation from an experimental point would result in an illogical relative error between its experimental value and those of the models. Our experimental results generally present this expected trend. For example, for all HDPE materials tested, for the 5Nm load and the three speeds 500 rpm, 1000 rpm, 1500 rpm, the results are all consistent with the expected trend. Then, the results are mostly consistent for the 6Nm torque. Finally, in terms of the results for the 2Nm torque, non-compliance is noted on several points. We recommend that in our further work, the tests be repeated for these experimental points with unexpected results in order to be able to elucidate the causes behind them.

On those, we now appreciate the comparison of the model results with those of the experimental tests for the driven wheel.

The precision of the results is at two levels, namely, the precision for the average values and the precision for the profile point values.

For the material HDPE30B with the torque 5Nm and for the two models, we have the best precisions where the greatest relative error for the average values is 11.76% (model E, HDPE30B, 5 Nm, 500 rpm), and 14.13 % (model E, HDPE30B, 5 Nm, 500 rpm) for the profile point values.

For the material HDPE40B with the torque 5Nm and for the two models, the largest relative error for the average values is 14.65% (model E, HDPE40B, 5 Nm, 500 rpm), and 18.77% (model E, HDPE40B, 5 Nm, 500 rpm) for the profile point values.

For the material HDPE30B with the torque 6Nm and for the two models, the largest relative error for the average values is 15.62% (model E, HDPE30B, 6 Nm, 1500 rpm), and 19.23% (model E, HDPE30B, 6 Nm, 1000 rpm) for the profile point values.

For the material HDPE40B with torque 2Nm, speed 500 rpm, 1000 rpm, and for both models, the largest relative error for the average values is 4.42% (model k, HDPE40B, 2 Nm, 1000 rpm), and 47.74% (model E, HDPE40B, 2 Nm, 1000 rpm) for the profile point values.

The details noted above represent the overall trend of results, for the driven wheel as well as for the driving wheel. We notice that the precision at the level of the average values is higher than for that of the individual points. We also notice that for the torques 5 Nm and 6 Nm (large torque), the individual points oscillate closely around the average value, but for the torque 2 Nm (low torque), many of the individual points are relatively far from the average value. This explains an accuracy of 4.42% for the average value and an accuracy of up to 47.74% for individual points for the 2 Nm torque.

To finish this section, we believe that our experimental results validly validate our models.

4. CONCLUSION

This present study laid the basis for predicting the wear of gears made of plastic materials and their composites. The models developed can already provide solutions to needs on an industrial scale.

From the information that the evaluation of the accuracy of the results gives us, we can draw the following important conclusions:

- On the one hand, the wear of the profile of gear teeth made of plastic materials and their composites is mainly the result of adhesive wear which follows the ideal worn shape of gears, and on the other hand, of wear by random tearing of material due to surface softening under the effect of the instantaneous temperature and the effect of the internal cohesion force.

- The wear rate accuracy per cycle for our models is practically preserved at the average value after a large number of cycles.
- The wear rate accuracy per cycle for our models varies with each cycle (due to random material tearing) for the profile points and after a large number of cycles, their absolute errors will not be the absolute errors per cycle multiplied by the number of cycles, but will have values in the order of hundreds of microns (because of the material tearing phenomenon) and consequently very low relative errors.
- The second and third conclusions allow our models to predict with great precision and great safety, the lifespan in terms of wear failure and generalized thermal failure for gears made of plastic materials and their composites.

The new wear model that has been developed is also based on Archard's law, but compared to Flodin's model, it presents simplicity in determining its exponential wear coefficient λ.

Because of the phenomenon of random removal of material during the wear process of plastic materials and their composites, the mean wear approach was adopted by the models in order to account for the effective total volume of material removed. The results of the two wear models effectively predict the wear behavior in accordance with that obtained by the experimental tests of Singh PK and Herba Y.

For each pair of mesh materials, initial experimental tests are necessary in order to determine the model parameters. The correct determination of parameters is very critical for good model accuracy. For this, we recommend that future studies to optimize the determination of parameters be conducted.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

REFERENCES

1. Kalpakjian S, Manufacturing Engineering and Technology, Third Edition. 1995; 198;216-217.
2. Kess K. Successful Application of Plastic Gear , 3e CMET, Paris, February 1992:721 – 728.
3. The Freedonia Group Inc.: Plastic gear demand by product (million dollars); 1998 –2018.
4. Wafiuddin, et al. Investigation on wear characteristic of biopolymer gear, 4th International Conference on Mechanical Engineering Research (ICMER2017), Materials Science and Engineering 2017; 257:012068
DOI: 10.1088/1757-899X/257/1/012068.
5. Koffi D, Analysis of methods for sizing plastic gears, Technology watch note produced for CETIM (Senlis) – France; 2004.
6. Osman T, Simulation of wear and damage on cylindrical gear teeth - Influence on the static and dynamic behavior of gear transmissions, Doctoral Thesis, INSA Lyon; 2012.
7. Flodin A, Wear Investigation of Spur Gear Teeth: Machine Elements, Department of Machine Design, Royal Institute of Technology, Sweden;2000.
8. Onishchenko V, Tooth wear modeling and prognostication parameters of engagement of spur gear power transmissions. Mechanism and Machine Theory. 2008;43: 1639–1664.
9. Wojnarowski J, Onishchenko V, Tooth wear effects on spur gear dynamics. Mechanism and Machine Theory. 2003;38:161–178.
10. Wu S, Cheng HS A Sliding wear model for partial EHD contact. Transactions of the ASME, 1991;113:134–141.
11. Kukureka SN, Chen YK, Hooke CJ, Liao P, The wear mechanisms of acetal in unlubricated rolling-sliding contact, elsevier, Wear. 1995;185:1-8.
12. Mao K, A new approach for polymer composite gear design: Mechanical Engineering, School of Engineering and Design, Brunel University, Uxbridge, Middlesex UB8 3PH, UK, ELSEVIER, Wear. 2007;262:432–441.
13. Singh PK, Siddhartha, Singh AK. An investigation on the thermal and wear behavior of polymer based spur gears. Tribology International. 2018;118: 264-272.
14. Herba Y, Metrological characterization study of gear wear of natural fiber-thermoplastic/birch composites and nanocomposites, Master's Thesis,

University of Quebec at Trois-Rivières, CANADA; 2022.

15. Agbetossou N, Afio A, Attipou K, Kassegne KA, Koffi D, Tiem S, Modeling and prediction of wear for gears in plastic materials and their composites, International Journal of Engineering Research & Technology (IJERT) Flight. 2020;9(07).
16. Düzcükoğlu H. Study on development of Polyamide gears for improvement of load carrying capacity, Tribology International. 2009;42:1146-53.
17. Kurokawa M, Uchiyama Y, Iwai T, Nagai S, Performance of plastic gear made of carbon fiber reinforced polyamide 12. ELSEVIER, Wear. 2003;254:468–473.
18. Flodin A, Andersson S, Simulation of mild wear in helical gears. Wear. 2000;241:123-128.
19. Flodin A, Andersson S, Simulation of mild wear in spur gears. Wear, 1997 :207:16-23.
20. Agbetossou N et al., New approach based on analytic-numerical method for calculating the sliding distance of teeth profile points during gears meshing. International Journal of Advanced Research. 2019:7;12.
21. Janakiraman V, Li S, Kahraman A, An investigation of the impacts of contact parameters on wear coefficient. ASME J Tribol. 2014;136(3):1–7.
22. Zhang J, Xianzeng L. Effects of misalignment on surface wear of spur gears, ARCHIVE Proceedings of the Institution of Mechanical Engineers Part J Journal of Engineering Tribology. 2015; 208-210;1994-1996
23. Koffi D, Study of the Thermal Behavior of Plastic Spur Cylindrical Gears, Doctoral Thesis (Ph.D.) in Mechanical Engineering, Ecole Polytechnique de Montréal, March 1987
24. Yelle H, Design of thermoplastic gears with an involute tooth profile. University of Waterloo; 1977.
25. Mijiyawa F, Formulation, Characterization, Modeling and prediction of the thermomechanical behavior of plastic and wood fiber composite parts: Application to Gears, Mechanical Engineering, University of Quebec at Trois Rivières, Trois Rivières; 2018.
26. Manai F, Study of the tribology performances of natural fiber-thermoplastic/birch composites: Friction and wear, Master's Thesis, University of Quebec at Trois-Rivières, CANADA; 2019.
27. Zhang J, Xianzeng L, Effects of misalignment on surface wear of spur gears, ARCHIVE Proceedings of the Institution of Mechanical Engineers Part J Journal of Engineering Tribology 2015;208-210:1994-1996.
28. Mao K, Li W, Hooke CJ, Walton D, Friction and wear behavior of acetal and nylon gears, ELSEVIER, Wear. 2009;267:639-645.

Peer-review history:
The peer review history for this paper can be accessed here:
https://www.sdiarticle5.com/review-history/ 110354