A Model for Predicting Churning Losses in Planetary Gears

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Introduction

Because of their compactness and axisymmetric arrangement, planetary gearboxes are widely used in automotive and aerospace applications. In the general context of the reduction of energy consumption and polluting emissions, gearbox efficiency has become a major issue. The power losses in planetary gears can be divided into two parts: i) the load-dependent power losses associated with the friction between the gear teeth and the frictional moment in rolling elements bearings; and ii) the load-independent (or no-load) power losses such as those generated by gears and planet-carrier windage, oil trapping and/or churning, seals and the viscous forces in rolling elements bearings. Numerous studies have been conducted on load-independent power losses for one pinion or a pinion-gear pair. Conversely, the studies, which deal with no-load dependent power losses in planetary gears, are sparse. They focus on global power losses produced by the planetary gear train but not on power losses distribution between the several sources previously stated.

It is important to underline that the sources of power losses can be very different from one mechanical transmission to another. One determining factor relies on the kind of gears which are considered. If one focuses on power losses occurring in cylindrical gear trains, numerous relationships can be found in literature. As an example, the work of Changenet on churning losses (Ref. 1) can be cited, whereas the work of Diab (Ref. 2) and Velex (Ref. 3) can be accounted for the study of friction losses.

As far as power losses occurring in planetary gears are concerned, the works conducted by Durand de Gevigney (Ref. 4), Kahraman (Ref. 5) and Talbot (Ref. 6) can be cited. However, all these studies concern oil jet lubricated planetary gear sets and it can be pointed out that no work deals with churning losses in planetary gears which are splash lubricated.

In order to investigate this source of dissipation, the authors have used a specific test rig. Tests were performed for different operating conditions: rotational speed, oil sump level and temperature. Some results have already been published by the authors in a previous paper (Ref. 8). The major conclusions are given here. An increase of the lubricant temperature was shown to produce a moderate decrease in churning loss, whereas it is strongly influenced by the rotational speed. Moreover, as the speed increases, the oil sump tends to disappear and the lubricant is distributed more toward the outer circumference of the housing. Then the oil sump level influence has to be interpreted according to this fluid distribution. The planetary gear set under consideration was designed in such a way that removing components is easy. Then experiments with no sun gear and/or by removing some planets have been performed during this study. These tests have demonstrated that churning losses of a planetary gear train can represented by a set of components consisting of viscous drag losses associated with gears (planets and sun gear) and the planet-carrier, and of oil trapping in inter-tooth spaces between planets and ring gear or planets and sun gear. The power losses caused by planet-carrier drag forces do not appear to be major contributors to the churning loss, whereas the number of planets is of primary importance.

Notations:

$P_{D,P}$, $P_{D,C}$, $P_{D,S}$	Drag losses due respectively to planets, planet-carrier and sun gear [W]
$P_{T,P-R}$, $P_{T,P-S}$	Pocketing losses due respectively to the contact between planets and ring gear and the contact between planets and sun gear [W]
ω_P or ω_C or ω_S	Rotational speed respectively of the planets, planet-carrier or sun gear [rad/s]
R_P or R_S	Pitch radius of the planets or the sun gear [m]
ρ	Oil density [kg/m^3]
Q_v	Flow of trapped oil [m^3/s]
v_p	Linear speed at pitch radius [m/s]
b_P or b_C	Width of the planets, planet-carrier [m]
m	Modulus of the gears [m]
n	Number of planets [-]
ν	Kinematic viscosity of the oil [m^2/s]
h_{tooth}	Tooth height [m]
h	Oil ring thickness [m]
R_{ring}^{root} or R_P^{root} or R_{sun}^{root}	Root radii of the ring, the planets and the sun gears [m]
R_C^{est} or R_C^{int} or R_C	External, internal or average radii of the planet-carrier [m]

In keeping with this first study, this paper aims to establish some analytical relationships to estimate churning losses in splash lubricated planetary gears.

Experimental Investigations

Test rig. A precise description of the test rig shown in Figures 1 and 2 is available in (Ref. 8); only the main features are exposed in this paper. The geometrical characteristics of the epicyclic gear train under consideration are listed (Table 1).

As explained (Ref.7), an electric motor is used to drive the planetary gear through a belt. The gear set is composed of a planet-carrier carrying 3 planets that are mounted on needle bearings.

Figure 1 Photo of the test rig.

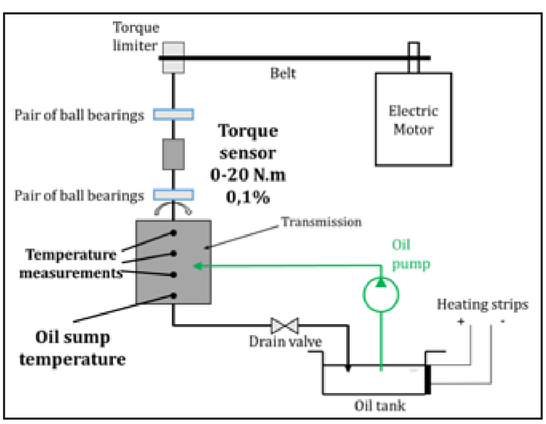

Figure 2 Scheme presenting the test rig.

Table 1 Planetary gear set geometrical features

		Sun gear	Planets	Ring gear	Planet- carrier
Number of teeth	[-]	54	27	108	-
Modulus	[mm]	1.4			Average radius: 70
Face width	[mm]	37	30	32	68
Pressure angle	[°]	20			-
Helical angle	[°]	0			-
Number of planets	[-]	3			-

The input speed is applied on the planet-carrier, whereas the sun gear is connected to the output shaft. In order to measure the power losses generated by this system, a torque sensor is used. Several type-K thermocouples are also used to monitor different temperatures (outer ring of roller bearings, ring gear and oil sump for instance). For this study the maximum input speed reached is 2,000 rpm.

For the oil sump, two kinds of lubricants have been used. Their physical characteristics are shown (Table 2).

Table 2 Physical properties of the oils used in the study

Oil number #	Density (kg/m^3)	Kinematic viscosity at 40°C (cSt)	Kinematic viscosity at 100°C (cSt)
1	837 at 30°C	41	7.4
2	900 at 15°C	200	18

Test protocol. Each test for any configuration is repeated twice in order to check the repeatability of the experiment and the consistency of the measure. For each measurement, the resisting torque is measured once the oil sump temperature is stabilized.

Due to its versatility (Ref.7) several configurations can be investigated with this test rig. As an example, some tests can be conducted by removing one or more planets and by using stationary cylinders (i.e. non-rotating mechanical components) instead of these gears. The different configurations are listed (Table 3).

Table 3 List of tests

No. of the test	Configuration	Speed range (rpm)	Viscosity values for oil 1 (cSt)	Viscosity values for oil 2 (cSt)
1	Planet-carrier with 3 cylinders, no sun gear	[200–2000]	7.4 - 13 - 24.9 - 88.2	111
2	Planet-carrier with 1 planet and 2 cylinders, no sun gear	[200–2000]	7.4 - 13 - 24.9 - 88.2	111
3	Planet-carrier with 3 planets, no sun gear	[200–1500]	7.4 - 13 - 24.9 - 88.2	111
4	Planet-carrier with 3 planets and sun gear	[200–1250]	7.4 - 13 - 24.9 - 88.2	111

As described (Ref. 7), the torque needed to run the planetary gear set is measured with oil sump in the casing for each operating condition and then again in the absence of any lubricant; thus, churning power losses are deduced by subtraction.

The four kinds of tests which are listed (Table 3) are performed in order to establish an analytical model to quantify the power losses generated by each component in the planetary gear set: the planet-carrier, the planets and the sun gear.

Summary of previous results (Ref.7). As it has been mentioned in the introductory section, the authors have already performed some experimental investigations to study churning losses in a planetary gear train. Because of centrifugal effects, they have shown that oil is ejected at the planet-carrier periphery and an oil ring is created (Fig. 3).

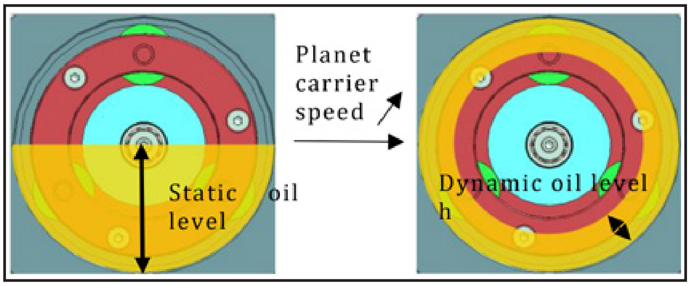

Figure 3 Scheme of the formation of the oil ring with its thickness noted h.

As far as churning losses are concerned, their evolution can be interpreted as a function of this oil ring thickness. Figure 4 presents a typical result for a rotational speed equal to 800 rpm and with oil #1 at 30°C. Three different parts can be underlined from Figure 4:

1. Only the lower part of the planets is submerged in the oil. The churning losses are associated with those due to oil pocketing and drag of the planets.
2. The oil ring reaches the planet-carrier. To the two previous sources of power losses, the drag forces acting on the planet-carrier must be added.
3. The last zone corresponds to an oil ring which reaches the upper part of the planets and the sun gear. These components generate new sources of dissipation.

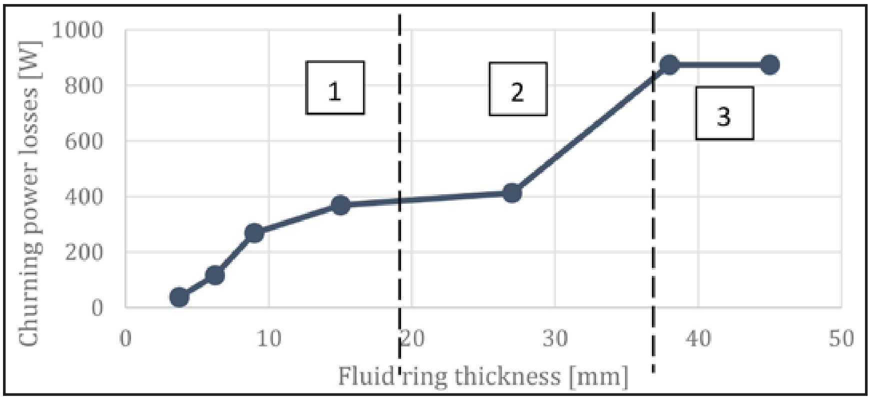

Figure 4 Churning power losses regarding the oil ring thickness.

It can be considered that the assumption of the oil ring is an acceptable approach to analyze churning losses. As a conclusion, this study aims to use this hypothesis in order to establish an analytical model for quantifying the churning power losses in a planetary gear set. As presented in (Ref.7), the churning power losses (P_{ch}) can be expressed as follows by considering the three above-mentioned parts:

In part 1,

$$P_{ch} = n(P_{D,P} + P_{T,P\text{–}R}) \quad (1)$$

Where n is the number of planets, $P_{D,P}$ represents drag power losses of planets and $P_{T,P\text{-}R}$ corresponds to the ones generated by oil trapping in inter-tooth spaces between planets and ring gear.

In part 2,

$$P_{ch} = n(P_{D,P} + P_{T,P\text{–}R}) + P_{D,C} \quad (2)$$

Where $P_{D,C}$ represents drag power losses generated by the planet-carrier.

In part 3,

$$P_{ch} = n(P_{D,P} + P_{T,P\text{–}R}) + P_{D,C} + P_{D,S} + n(P_{T,P\text{–}S}) \quad (3)$$

Where $P_{D,S}$ represents drag power losses of sun gear and $P_{T,P\text{-}S}$ corresponds to the ones generated by oil trapping between planets and sun gear.

In conclusion, several sources of power losses need to be calculated in order to predict the overall churning losses. In this study, the drag loss due to the sun gear has not been developed because of the non-sufficient immersion level to reach the sun gear.

Analytical Expressions of Churning Power Losses

Oil pocketing. In order to predict the power losses due to oil pocketing between gears, the formula of Mauz is used (Ref.8):

$$P_{T,P\text{–}R} = \omega\, R_P .\, 4.12 .\, \rho Q_v^{0.75} v_p^{1.25} b^{0.25} m^{0.25} (\nu)^{0.25} \left(\frac{h_{tooth}}{h_0}\right)^{0.5} \quad (4)$$

With $h_0 = 2.3 \times \mathrm{m}$.

Equation 4 is given to estimate $P_{T,P\text{-}R}$, but it can be also used to quantify $P_{T,P\text{-}S}$.

As far as oil trapping between ring gear and planets is concerned, the oil flow is estimated as follows:

$$Q_v = v_p . b . \min\left(h; R_{ring}^{root} - R_C^{ext}\right) \quad (5)$$

As far as oil trapping between planets and sun gear is concerned, the oil flow is quantified as follows:

$$Q_v = v_p . b . \min\left(h - R_C^{int}; R_{sun}^{root} - R_C^{ext}\right) \quad (6)$$

The above formulae can be used to subtract the oil pocketing power losses to the overall churning losses. From this approach, it is possible to isolate the power losses generated by drag effects. The function "*min*" is used to describe the fact that the oil ring does not always reach the planet-carrier (Eq. 5)or the sun gear (Eq. 6), depending on its thickness.

Drag power losses. To determine this source of dissipation, the work from Changenet (Ref.1) is used and the power losses can be expressed as follows:

$$P_{D,P\text{ or }C} = \frac{1}{2} \rho \omega^3 S_m R^3 C_m \quad (7)$$

In this formula:

- ω is the rotational speed of the considered rotating element: for the planet and for the planet carrier [rad/s]
- R is the pitch radius of the considered rotating element: for the planet-carrier, for the planets and for the sun-gear [m]
- C_m is the drag coefficient [-]
- S_m is the wet surface [m^2]

The use of Equation 7 underlines that for each source of dissipation, the wet surface S_m and the drag co-efficient must be calculated differently.

Planet-carrier. To begin with, the wet surface is calculated as follows:

$$S_{m,C}=\underbrace{\overbrace{0.7}^{3\,holes\,for\,planets}\times 2\pi . R_C^{ext}.b_c}_{cylindrical\,part}+\underbrace{2\times 2\pi\left(R_C^{ext2}-(R_{ring}^{root}-h)^2\right.}_{two\,flanks} \tag{8}$$

As a first approximation, the planet-carrier is assimilated to a rotating disk and its drag coefficient can be deduced from the works of Goldstein (Ref. 9):

$$C_{m,C}=\alpha\times R_e^{-0.16} \tag{9}$$

Where α is a coefficient and Re is the Reynolds number that is calculated by:

$$Re=\frac{2\,R_C^{ext2}\omega_C}{\upsilon} \tag{10}$$

As the geometrical shape of the planet-carrier is more complex than the one of a disk, the configuration associated with test No. 1 (Table 3) is used to isolate the drag of the planet-carrier. The value of the coefficient α was deduced from these tests: α = 0.046.

Planets. To develop the model for estimating the planet drag torque, the measurements performed during tests No. 2 and 3 (Table 3) are used. This source of dissipation can be isolated from the overall measured power losses by subtracting the oil pocketing losses and the planet-carrier drag losses.

To use Equation 7, the wet surface is calculated by using Equations 11–12. The parameters used in these equations are defined (Fig. 5).

$$S_{m,P}=R_P^2(2\theta-\sin 2\theta)+2R_Pb\theta+\frac{2Z\theta h_{tooth}b}{\pi\cos\alpha} \tag{11}$$

$$\theta=\frac{\theta_1+2\theta_2}{2} \tag{12}$$

$$\theta_1=\begin{cases}2\cos^{-1}\left(\frac{R_P-h}{R_P}\right) & \text{if } h\le R_{ring}^{root}-R_C^{ext}\\ 2\cos^{-1}\left(\frac{h_c}{2R_P}\right) & \text{else}\end{cases}$$

$$\theta_2=\begin{cases}0 & \text{if } h\le R_{ring}^{root}-(R_{PS}^{ext}-h_c)\\ \sin^{-1}\left(\frac{H}{R_P\left(1-\frac{h_c}{h_c+2(H)}\right)}\right)-\tan^{-1}\left(\frac{h_c}{2R_P}\right) & \text{else}\end{cases}$$

With:

$$H=R_C^{ext}-h_c-(R_{ring}^{root}-h) \tag{13}$$

Dimensional analysis (Ref. 10) has been used to determine an expression of the drag coefficient:

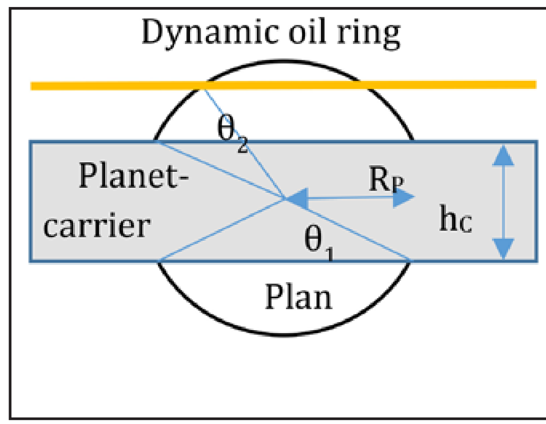

Figure 5 Scheme for the calculation of the planet's wet surface.

$$C_{m,P}=3.88\times Re^{-0.1}\times Fr^{-0.63} \tag{14}$$

$$Re=\frac{2R_{sat}^2\omega_P}{\upsilon}$$

$$Fr=\sqrt{\frac{\omega_P^2R_P}{g}}$$

Re is the Reynolds number that characterizes the oil flow around the planets and Fr is the Froude number. They account for the ratio of inertial forces to viscous forces within the oil and for the ratio of the flow inertia to the gravity.

Comparison between measurements and calculated results. A first set of comparisons is given (Fig. 6). Churning power losses are plotted as a function of the oil ring thickness. These results are given when oil #1 is used and for an input rotational speed equal to 800 rpm.

A satisfactory agreement is observed for all immersion levels. Further comparisons are then made at a given oil ring thickness, i.e.: 24 mm. Figure 7 presents the churning losses evolution as a function of planet-carrier rotational speed. Here again, a satisfactory agreement is shown between calculated values and the measured ones.

Finally, Figure 8 presents the churning losses evolution as a function of the lubricant kinematic viscosity when oil #1 is used.

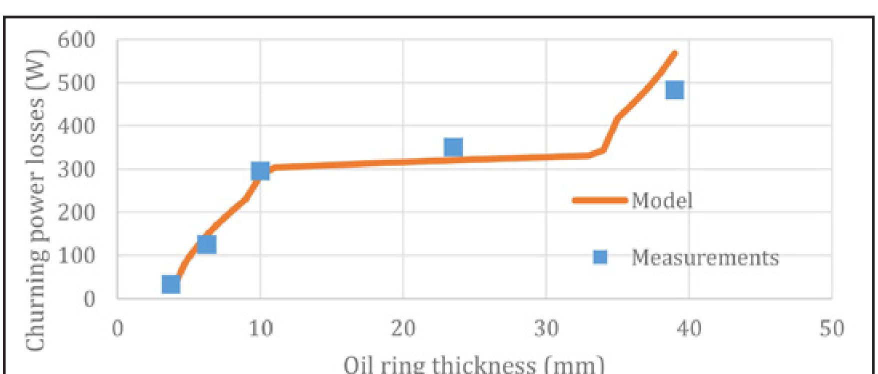

Figure 6 Testing of the previous two models regarding oil dynamic level (at 800 rpm and 88cSt).

As it has been underlined in previous studies (Refs. 1,9), this figure demonstrates that the influence of oil viscosity on churning losses is not very significant.

Conclusion

This study aimed to develop a model for calculating churning power losses for a splash lubricated epicyclic gear train. The proposed model relies on the hypothesis of a dynamic oil level as assimilated to a ring (Ref. 7). The approach is also based on previous studies on churning power losses for cylindrical gears (Refs. 1,11). Some analytical formulations are given to quantify the different sources of dissipation in churning losses, i.e.: oil pocketing and drag torque acting on rotating components. Some comparisons with experimental results obtained on a specific test rig show that this model can be satisfactory.

Future work will consist of investigating the influence of these power losses on the thermal behavior of such planetary gear trains.

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For more information. Questions or comments regarding this paper? Contact Jean-Baptiste Boni at *jean-baptiste.boni@ecam.fr*. **PTE**

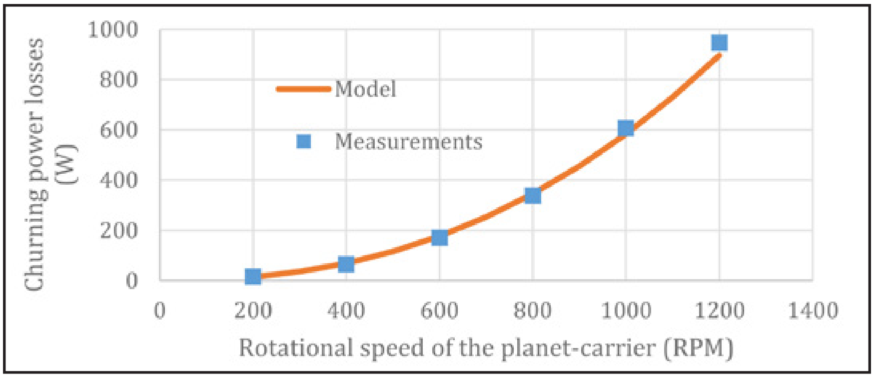

Figure 7 Testing of the previous two models regarding input speed (at 88cSt and h = 24 mm).

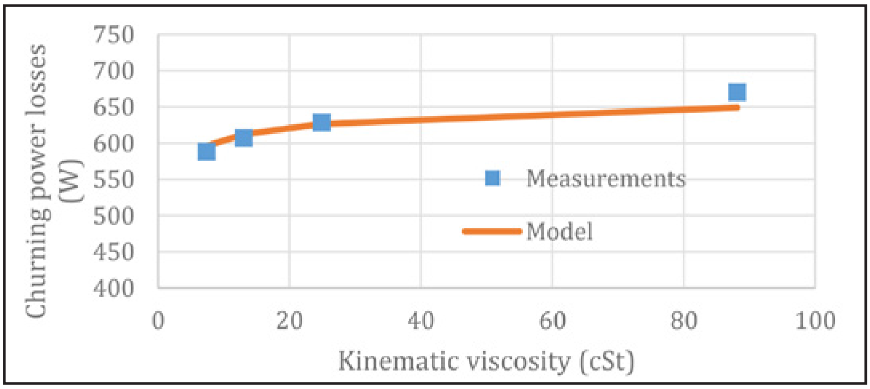

Figure 8 Testing of the previous two models regarding oil viscosity (at 1,000 rpm and h = 24 mm).

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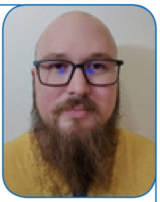

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