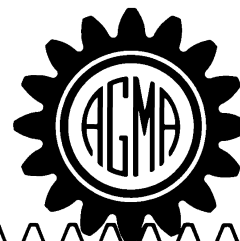


Detailed Procedure for the Optimum Design of an Epicyclic Transmission Using Plastic Gears

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American Gear Manufacturers Association



TECHNICAL PAPER

Detailed Procedure for the Optimum Design of an Epicyclic Transmission Using Plastic Gears

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[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

Abstract

This paper shows all the steps suggested to get an optimum (volume based) design for an epicyclic (planetary) transmission using plastic materials. The design was developed using the tooth proportions proposed in ANSI/AGMA 1006-A97, Tooth Proportions for Plastic Gears and taking into account the recommendations given in AGMA 6023-A88, Design Manual for Enclosed Epicyclic Gear Drives and ANSI/AGMA 2101-C95, Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gears taking into account the effect of changing the number of planets, the bending fatigue and contact strength of the plastic materials, and the temperature effects on the size of the gears.

The design procedure starts with a preliminary analysis of the performance of the gears in a proposed (not optimized) transmission, going step by step to an optimum design for the given load conditions and expected minimum life.

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Introduction

The design of an epicyclic transmission involves some special considerations like assembling possibility, factorizing, and idler planets considerations not necessary for a conventional parallel axles transmission. If additionally we include the expansion characteristic of a plastic material, the design of a plastic epicyclic transmission may become very involving.

By following an example systematically, we will cover all the considerations needed for the design of an optimum epicyclic transmission.

Nomenclature

N_S	Number of teeth in the sun
N_P	Number of teeth in the planet
N_R	Number of teeth in the rim
ω_S	Rotating speed of the sun
ω_P	Rotating speed of the planet
ω_R	Rotating speed of the rim
ω_C	Rotating speed of the carrier
M_g	Overall gear ratio
NP	Number of equally distributed planets
CR	Ratio operating/theoretical center distance
F_W	Operating face width

Problem Description

Type of transmission	Planetary
Gear ratio	4 ($\pm 3\%$)
Rim pitch diameter	70 mm (Max)
Face width	Minimize
Sun input speed	600 RPM
Sun torque (Max.)	6.25 N·m
Bending durability	2000 Hr
Carrier material	Aluminum
Ambient temperature	25°C
Operating temperature	75°C
Lubrication	Oil lubricated

General Design Guidelines

The basic speed equations for an epicyclic transmission may be found in the literature [1]:

$$\omega_S N_S = \omega_R N_R + \omega_C (N_S + N_R) \quad (1)$$

$$\omega_S N_S = \omega_C N_S + \omega_P N_P \quad (2)$$

From eqn. 2, in the planetary:

$$\omega_R = 0; \text{ Therefore,}$$

$$\frac{\omega_S}{\omega_C} = \frac{N_S + N_R}{N_S} \quad (3)$$

$$\text{And } \omega_P = \frac{(\omega_S - \omega_C) N_S}{N_P} \quad (4)$$

In addition, for equally distributed planets with standard center distances, the assembly constraints given by eqns. 5 and 6 must be accomplished:

$$N_R = N_S + 2 N_P \quad (5)$$

$$\frac{N_S + N_R}{N_P} = \text{Integer} \quad (6)$$

In practice, designs with non-standard center distances and dropped tooth planets are commonly used; in that case, eqn. 5 may be ignored.

Materials

The selection of the type of plastic to be used for the gears may be information for a full article and is beyond the scope of this paper. For this example, the selection of the material was based in the recommendations of the plastic vendor, who suggested two possibilities:

An unfilled high molecular weight acetal copolymer for maximum toughness (HMW POM) and a 25% glass reinforced acetal copolymer (GF POM). One important consideration is that for sliding operation like gears, it is a good practice to combine two different materials or grades in both members. In this case, due to the application, it is also important to have a very strong cage (rim gear); therefore, based in their strength, the materials were assigned as follows:

Rim GF POM
Planet HMW POM
Sun GF POM

From the information provided by the plastic vendor, the properties listed in Table 1 are relevant during the gear design.

As may be observed, the Young's modulus and strength of the plastics reduces with the temperature in agreement with the behavior stated in AGMA [2] and shown in Fig. 1. Therefore, for a conservative calculation, we use the material properties at the highest operating temperature highlighted in Table 1.

Table 1. Relevant properties of the materials

Property	Material			
	HMW POM		GF POM	
Temperature (deg C)	40	75	40	75
Young's Modulus (MPa)	2414.5	1339.33	2700	1500
Poisson's Ratio	0.35	0.35	0.35	0.35
Endurance limit 10^7 cycles (MPa)	61	34.92	55	47.97
Contact stress dry (MPa)	19	15.9	19	15.9
Contact stress lubricated (MPa)	63.2	52.5	63.2	52.5
Linear thermal exp, coeff.	1.2E-04	1.2E-04	3.0E-05	3.0E-05
Aluminum L.T.E.C.	2.4E-05			

Procedure

The AGMA [3] suggest that for plastic gears the basic rack must have the proportions shown in Fig. 2 and summarized as follows:

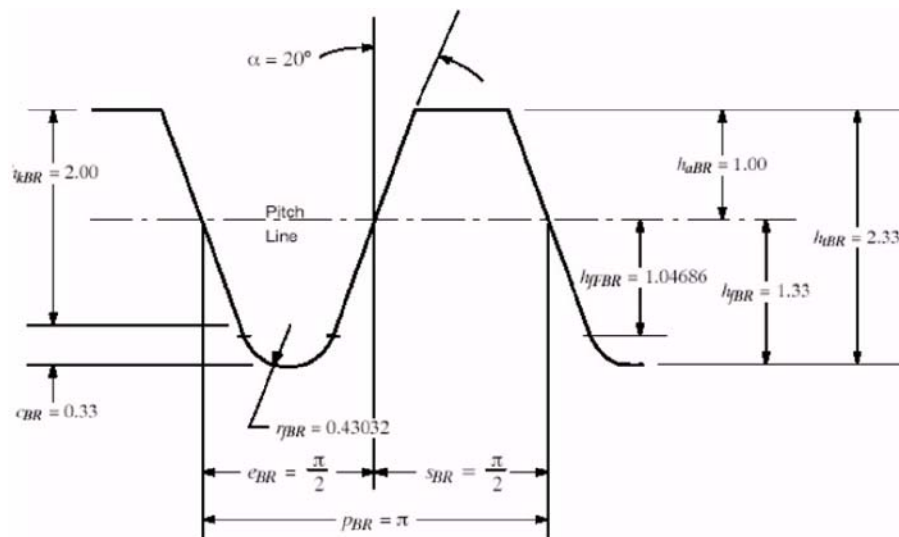


Figure 2. AGMA PT basic rack (from AGMA [2]).

Addendum coefficient 1.33
Dedendum coefficient 1.00
Tip radius coefficient 0.43032 (full)
Normal pressure angle 20°

All the calculations shown in this paper are based on this tooth geometry with a non-undercut constrain. Additionally, although the minimum recommended tooth thickness at the tip of the tooth is $0.275M$ [4], in order to consider as many options as possible without a pointed tooth, during the development of this study a minimum of $0.1M$ was used.

Using eqn. 3 with the nominal gear ratio and the maximum pitch diameter of the rim, we get the options given in Table 2 iterating from $N_S = 11$ to 30.

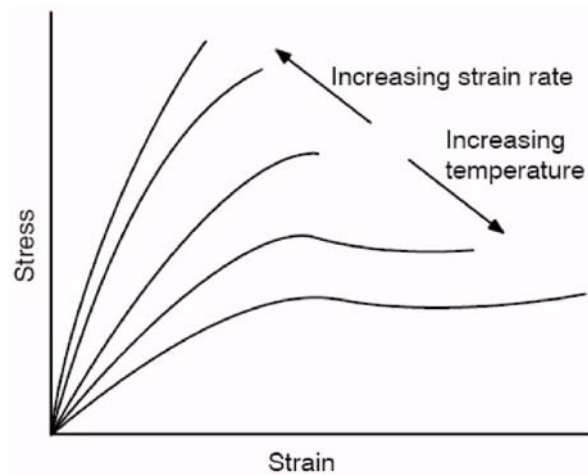


Figure 1. Effect of strain rate and temperature on stress-strain curves (from AGMA [2]).