ANNEX A

The Face Width Parametric Variation Method for Estimating Gear Size

1 Introduction

In order to analyze reliability of a gearset, or to determine the factor of safety guarding against the several kinds of failures, it is necessary to know the size of the gears and the materials of which they are made. In this annex we are concerned mostly with getting a preliminary estimate of the size of gears required to carry a given load. This method does not take into account cumulative fatigue factors, therefore can only be used to design gearsets in which life and reliability are not a major design consideration.

The design approach presented in here is based on the choice of a face width in the range $3p>F>5p$. Gearsets having face widths greater than five times the circular pitch are quite likely to have a non-uniform distribution of load across the face of the tooth because of the torsional deflection of the gear and the shaft, because of machining inaccuracies, and because of the necessity of maintaining very accurate and rigid bearing mounting.

When face width is less than three times the circular pitch, a larger gear is needed to carry the larger load per unit of face width. Large gears require more space in the gear enclosure and make the finished machine bigger and more expensive. For these reasons a face width of three times the circular pitch is a good lower limit on the face width. It should be noted, however, that many other considerations arise in the design that may dictate a face width outside the recommended range.

The gear size is obtained using iteration because both the transmitted load and the velocity depend, directly or indirectly, on the module m. The given information is usually:

- the power H;

- the speed n in [rpm] of the gear to be sized;

- the Lewis form factor Y (see Table A.1 below) for the gear to be sized;

- the permissible bending stress σ_P.

When estimating the gear sizes it is a good principle to adopt a safety factor of 3 or more, depending upon the material and application.

2 Method

This section contains an outline of the steps involved on the procedure, which is based on selection of a trial value for the module and then following the six steps listed below:

i) The pitch diameter d in meters from the equation:

$$d = mN(10^{-3})$$

, where N is the number of teeth and should be defined according to Table A-1 shown below, and m might be selected according to Table A-2 shown below;

ii) The pitch line velocity V in meters per second from the equation:

$$V = \frac{\pi dn}{60}$$

, where n is in [rpm];

iii) The transmitted load W_t in newtons from the equation:

$$W_t = \frac{H}{V}$$

, where H is the power in watts;

iv) The velocity factor from equation:

$$K_v = \frac{6}{6+V}$$

v) The face width F in millimeters from equation:

$$F = \frac{W_t}{K_v m Y \sigma_P}$$

, where σ_P is the permissible bending stress in [MPa];

vi) The minimum and maximum face widths *3p* and *5p*, respectively, in [mm].

These six steps can be programmed if desired to speed up the calculation.

Tab. A.1 – Formulas for Tooth Dimensions for Pressure Angles of 20º and 25º.

Quantity	Formula
Addendum	$a = m$
Dedendum	$b = 1.25m$
Working depth	$h_k = 2m$
Whole depth (min.)	$h_t = 2.25m$
Tooth thickness	$t = m/2$
Fillet radius of basic rack	$r_f = 0.300m$
Clearance (min.)	$c = 0.25m$
Clearance, shaved or ground teeth	$c = 0.35m$
Minimum number of pinion teeth	
$\phi = 20°$	$N_P = 18$
$\phi = 25°$	$N_P = 12$
Minimum number of teeth per pair	
$\phi = 20°$	$N_P + N_G = 36$
$\phi = 25°$	$N_P + N_G = 24$
Width of top land (min.)	$t_0 = 0.25m$

Tab. A.2 – Modules in General Use.

Preferred	1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 16, 20, 25, 32, 40, 50
Next choice	1.125, 1.375, 1.75, 2.25, 2.75, 3.5, 4.5, 5.5, 7, 9, 11, 14, 18, 22, 28, 36, 45