A Review: Optimization of Gear Face Width and Material Selection under the Influence of Bending Stress

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Abstract— Gears are the critical parts of transmission system. Gears are used to transmit power or torque through direct contact between paired gears. Global competition in automotive market has brought the awareness to optimize the transmission design by optimizing the gear pair without compromising with its life. An optimization consists of selection of proper material and face width of gear pair by keeping the bending stress within safety limit and also by keeping the size or weight of gear pair smaller as much it can be without compromising with its performance. The purpose of this study is to establish a relation between different materials and face widths with the help of bending stress calculations (As per ISO 6336-3), also with some help of computer software viz. Excel spreadsheet. Calculated curves show the relationship between different materials Vs Permissible bending stresses and maximum bending stresses Vs Face widths Vs Materials to select most suitable one as per application requirement. Curves also show increase in price with respect to increase in face width. With the help of these curves one can easily select material and face width as per their application.

Key words: Material, Stresses, Bending Stress, Face Width, Gears & Transmission

I. INTRODUCTION

Bending stress or tooth root stress can cause breakage of gear tooth which can further cause the destruction of all gears in a transmission. To overcome this, a larger safety factor for bending stress is taken into account while designing a gear pair against the safety factor for contact or pitting strength.

There are number of studies going on related to the stress calculation of gear. Researchers and manufacturers have shown keen interest in the calculation of form parameter of gears [1] like bending and contact stress. First study in this area was done by Rose and Buckinghom [2] in 1920. Later in 1989, a first method for load and stress distribution between the engaged tooth of spur and helical gears was developed [3]. This method deals with the stress distribution with the change in gear parameters. It also includes the other parameter calculations like viz. tooth profile modification, crowing on gear tooth, tooth deflection and load distribution along tooth contact. Value of bending strength on a gear root is directly proportional with the material used. In 2008, Engineers from Mahindra and Mahindra published a journal regarding the strategies to select a material for a gear pair [4]. A material model shows the material behavior in low cycle regime [5] is also helped. This material model consist isotropic and kinematic hardening with mechanics of material damage to simulate elastic-plastic response of material, as well as damage nucleation and accumulation. Similarly gear face width is inversely proportional to the Root bending stress. With Alloy steel gear, increase in face width result in decrease in bending stress [6] by keeping other gear parameters like Module, Helix angle and gear ratio constant.

Besides the individual studies of researchers and manufacturers, there are several national standards used in industries for gear Geometry and capacity calculation. Among the ISO (International Standard Organization), AGMA (American Gear Manufacturers association), DIN (Deutsches Institut für Normung), and JIS (Japanese Industrial Standards); ISO and AGMA are most popular and widely used as per Kawalec [7]. ISO 6336 is considered the best method for gear capacity calculations. With the latest revision of ISO 6336-2006, it is become partially equivalent to DIN 3990 [8] and partially equivalent to AGMA 2101-D04 [9], as per Euro Trans [10]. ISO 6336 consists of several parts. Detailed information of each part and where they can be used; is described below:

- ISO 6336-1: Basic principles, introduction and general influences [11]
- ISO 6336-2: Calculation of surface durability (pitting)
- ISO 6336-3: Calculation of tooth bending Strength [12]
- ISO 6336-5: Strength and quality of materials [13]
- ISO 6336-6: Calculation of service life under variable load [14]

II. DESIGN METHODOLOGY

As described above, ISO 6336-3 shows how to calculate tooth root stress σ_F and permissible tooth root stress $\sigma_{FP.}$ Theory behind this equation is same as cantilever beam theory and similar to AGMA 2001, with several significant differences.

$$\sigma_F = \sigma_{F0}.K_A.K_V.K_{F\beta}.K_{F\alpha} \quad (1)$$

$$\sigma_{F0} = \frac{F_t}{b.m_n}.Y_F.Y_S.Y_\varepsilon.Y_\beta \quad (2)$$

$$\sigma_{FP} = \frac{\sigma_{Flim}.Y_{ST}.Y_{NT}}{S_{F\,min}}.Y_{\delta relT}.Y_{RrelT}.Y_X \quad (3)$$

$$\sigma_{FP} = \frac{\sigma_{FG}}{S_{Fmin}} \quad (4)$$

Where, σ_{F0} is the nominal stress at tooth root,
σ_F is the calculated tooth root stress,
σ_{FP} is the permissible tooth root stress, and
σ_{Flim} is the nominal stress number.
σ_{FG} is the Limit strength tooth root of material

In this study, the nominal stress number σ_{Flim} was obtained from ISO 6336-5. It is derived from testing reference test gears. It is the bending stress limit value relevant to the influence of material, the heat treatment and surface roughness of test gear root fillets.

$$\sigma_{Flim} = A\,.\,x + B \quad (5)$$

Where, x is the surface hardness in HBW or HV
A, B are constant (ISO 6336-5, table 1)

If the value of maximum calculated tooth root stress (σ_F) is less than the Limit strength tooth root (σ_{FG}) of material, it is presumed that no tooth breakage will be occurred within predicted life. i.e.

$$S_F = \frac{\sigma_{FG}}{\sigma_F} \geq S_{Fmin} \qquad (6)$$

S_{Fmin} is the minimum safety factor which is to be agreed on between designer and customer. The minimum safety factor was set to 2 in this study, since the analyzed gear pair is being designed of heavy duty vehicle.

F_t is nominal tangential force and can be calculated by

$$F_t = \frac{2000.T}{d} \qquad (7)$$

Where, T is torque transmitted by gear.
D is reference diameter of gear.

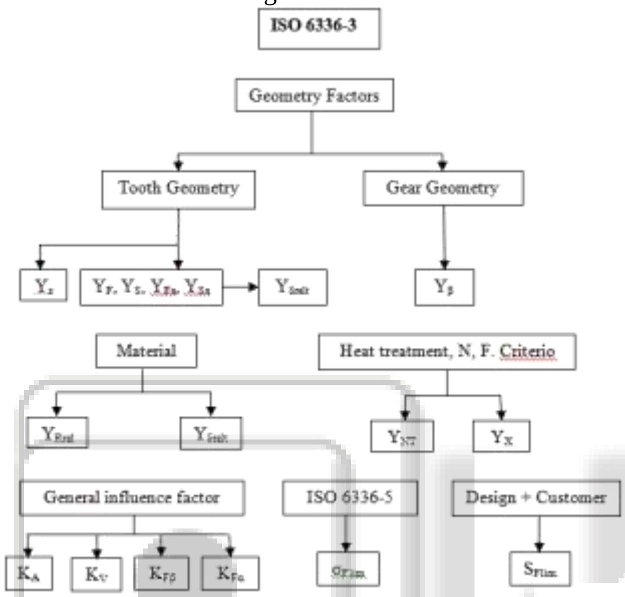

Fig.1. Flowchart to establish modified factor in ISO 6336-3.

Meaning of symbol used in above equations are described in Table 3. Since it is very difficult to find out impaccable value of these factors. As the factors in the ISO 6336-3 standard can be determined by three methods A, B and C. Out of these three A method is superior to B and B is superior to C. The method A is highly accurate one. To derived factor as per method A, a full scale test and high skills are required; therefore it seldom used only in highly precise gearbox. On the other hand, method C is almost same as method B except some approximation are used for some factors. In this study, method B is used when advanced knowledge or calculation is required; and method C is used when no method is mentioned in method B for factor calculation.

At first step, factors are sorted in three groups

1) Factors derived from theoretical group are Y_S, Y_β, Y_ε, etc.
2) Factors derived with a theoretical basis from experimentation are K_A, K_V, $K_{F\alpha}$, $K_{F\beta}$, etc.
3) Factors derived to adjust real work conditions from experimentation are Y_F, $Y_{\delta relT}$, Y_{RrelT}, etc.

A flowchart of the factors is also presented in Figure 1 in order to give an overview of the ISO 6336-3 (Tooth bending strength) calculations.

III. INPUT TO ROBUSTNESS ANALYSIS

A. *Assumptions for gear geometry calculations*

– It is assumed that the gears are geometrically ideal with no profile error or tooth spacing error.
– Uniform line contact between mating gear.

The gear pair is being analyzed (Table 1 and Figure 2) is an automotive gear design. The material properties and working characteristics of the gear are shown in Table 2. The applied torque and other factor shown in Table 1 are calculated values, unlike the some values in Table 2, which are taken from ISO 6336-5.

		Drive Gear	Driven gear
Number of teeth	Z	23	49
Module	m_n	3.0	
Pressure angle, (deg.)	α_n	20	
Helix Angle, (deg.)	β_b	0	
Reference circle diameter,(mm)	D	69	147
Addendum modification coefficient,	X	+0.480	+0.303
Center Distance, (mm)	a	110	
Torque, (Nm)	T	145	-
Minimum required safety factor,	S_{Fmin}	2.0	

Table 1: Input gear geometric parameters as per DIN 3960 [15]

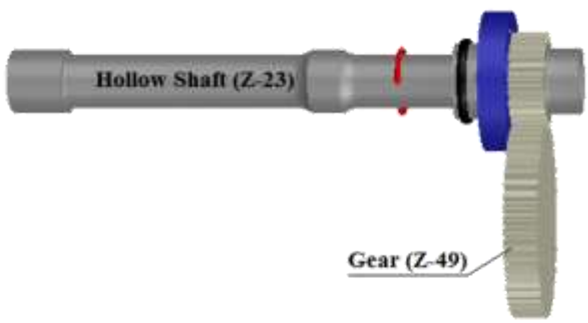

Fig. 2: Illustration of analyzed gear pair.

Material	16MnCr5	20MnCr5	SAE 8620	En353	Steel Grade 3	C45	42CrMo5
Tensile strength (N/mm^2)	1000	1200	980	1200	1035	700	900
Yield Point (N/mm^2)	695	850	785	835	867	490	700
Young's Modulus (N/mm^2)	206000				206843	206000	
Poisson's ratio	0.3						
Surface hardness (HRC)	59	60	60	60	60	57	16
Material Type	Case carburized Steel					Through hardened Steel	

Table2. Material parameters used in calculation

For this analysis, Seven different materials are selected which are being widely used in industries. The selection of materials are done on the basis of their hardness type and manufacturing category. SAE 8620, En353 and 20Mncr5 are case carburizing steels and are being widely used for Automotive Gears. Due to their core hardness value

≥ 30 and 25 HRC, they have very good shock absorbing capability. Whereas 42CrMo4 and C45 are through hardened steel. The main applications of these materials in machineries and LCV. Steel grade 3 AGMA 2001 on the other hand used as an aircraft gearbox material. It can be used compact and light gearbox required.

IV. RESULTS

The calculation analysis of bending stress in done by using above equation. Basic gear inputs parameters viz. module, gear ratio, input torque, reference diameter, helix angle, and pressure angle are kept same in every calculations carried out during this work. The changes in every calculation are material property and Face width. Number of iterations are carried out to find out optimum gear Face width for respective material to justify Safety limit factor $S_{Flim} = 2$. The results found during this analysis is described in Table 3.

Description	Sym.		Driven gear					
Material		16MnCr5	20Mncr5	SAE 8620	EN353	Steel Grade 3	C45	42CrMn5
Tooth root Stress (N/mm^2)	σ_F	297.95				492.77	208.22	
Nominal Stress at tooth root (N/mm^2)	σ_{F0}	184.50				417.12	87.22	
Limit strength tooth root (N/mm^2)	σ_{FG}	829.67		964.73	829.67	1009.25	713.90	
Permissible tooth root stress (N/mm^2)	σ_{FP}	592.62		689.10	592.62	720.89	509.93	
Nominal tangential force (N)	F_t	4202.9						
Safety for tooth root stress	S_F	2.0						
Face width (mm)	b	26		16.5	20	11.5	55	
Application factor	K_A	1						
Dynamic factor	K_V	1						
Face load factor	$K_{F\beta}$	1.457				1.181	1.974	
Transverse load factor	$K_{F\alpha}$	1.108				1.000	1.210	
Foam factor	Y_N	1.48 (as per clause 6: ISO 6336-3)						
Stress correction factor	Y_S	2.32 (as per clause 7: ISO 6336-3)						
Contact Ratio factor	$Y\varepsilon$	1.0						
Helix angle factor	Y_β	1.0						
Stress correction factor	Y_{ST}	2.0						
Life factor	Y_{NT}	1.0						
Notch sensitivity factor	$Y_{\delta relT}$	1.008						
Relative surface factor	Y_{RrelT}	0.957				0.972	0.957	
Size factor	Y_X	1.0						

Table 3: Meaning of Symbols used in equations and calculated values as per ISO 6336

From the analysis outcome, it is observed that between the selected materials Steel grade 3 AGMA 2001 and SAE 8620 have the maximum bending stress limit value (σ_{Flim}) and Steel alloy C45 and 42CrMo4 possesses the lowest value. By using iterative method, an optimum value of Face width is find out for every selected materials by keeping the safety factor constraint S_F >=1.4 as per minimum requirement. As Steel grade 3 AGMA 2001 has maximum bending stress limit value (σ_{Flim}), it satisfied the condition at b =11.5 mm. Similarly when we move toward other materials, σ_{Flim} goes on decreasing and value of b goes on increasing. At same input parameters and same working condition, C45 satisfied the condition at b = 55mm. Variation of minimum stress value for different materials at different face widths are shown in fig. 3. Similarly calculated Tooth root stress at selected Face Width is illustrated in fig.4.

For an optimum gearbox, it is an ideal condition that a gear pair transmits maximum power with minimum face width and of light weight. It is obvious that with increase in face width gear effective width increases and further gear weight. This increases the overall size and weight of gearbox. However high end materials also have more tooling and raw material cost as compared to materials at lower side. Special tooling is required to machine these materials which further increase the overall gear manufacturing cost. Rate comparison of different material is done in Fig. 6. These raw material prices are current Indian market price and are subjected to change.

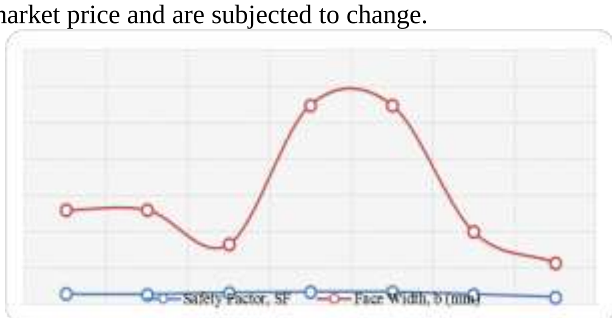

Fig. 3: Limit Bending stress curve for different material and Face Width

Fig. 4 Calculated Tooth root stress σF for different material

Fig. 6 Raw material price in india

V. CONCLUSIONS

Till date there isn't any specific method for material selection other than hit and trial. Material selection is done on the basis of designer experience or after endurance testing. Similarly, Face Width calculation is based on the basic ISO formula i.e. $10m_n$ >Face Width >$20m_n$. The Value below lower limit results in a coarse pitch and above higher limit can cause concentration of load at one end. But these errors can be avoided by adapting advanced machining and assembly procedure.

On the other hand, this analysis represents the easiest way to select material and Face width. By using this method, a designer can select optimum material and Face width as per application requirement. For example- if the cost is the major constraint in gear design as compared to space, than the lowest material can be used with higher Face Width. And where space or working environment (Shock load) is an issue, high end material like SAE 8620, 20MnCr5 or material with same properties can be selected. This method will reduce the time wastage and money of gear manufacture by eliminating the endurance testing done to find out the optimum Face Width and material.

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