Design Calculation of Timing Gears Trains for Tractor Engine (80HP)

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Abstract- The objective of this paper is to calculate the design of timing gears trains for MICO Tractor Engine (80hp), which is an inline, 4-stroke, 4-cylinder diesel engine. The engine is produced maximum rated output power of 59.68kW (80hp) at 1800 rpm. This timing gear trains for Tractor Engine is incorporated with helical gears. It consists of five helical gears are used. Synchronize mesh is used for this timing gear for tractor engine. This paper is consider the design calculation of timing gear. Tooth load is calculated with help of Lewis equation & dynamic tooth load is calculated with help of Buckingham equation. Static analysis of the gear is done to find the von-Mises stress on the tooth of the gear in while meshing with Autodesk software.

Index Terms- helical gears, tooth load, Buckingham equation, Lewis equation, von-Mises stress

I. Introduction

Today, automotive vehicles are widely used in many countries for transportation, farm agriculture and other purposes. So, it is necessary to know about automotive vehicle engine in detail. Automotive technology is also a required to develop the country rapidly.

In general, automotive vehicles are both compression ignition (diesel) engine and spark ignition (petrol) engine. Mostly Tractor Engine uses the internal combustion compression ignition engine. Because it is used heavy duties such as trucks, trains and ships. Fuel is injected from the fuel pump and fuel injectors directly into the combustion chamber at high pressure at the end of the compression stroke. The Tractor Engine is located at the front of the vehicle and connected with clutch housing, gears box and drives both large rear wheels. The engine type is four-cylinders, four-stroke inline engine. The firing order is 1-3-4-2. The brake horsepower of MICO Tractor Engine is 80 hp (59.68kW) at maximum engine revolution of 1800 rpm.

In this engine, the function of gears trains is important to get valves timing for entering the air into the cylinder, to release exhaust gas and is also important engine oil circulate into the engine block. The starting system of MICO Tractor Engine is also important to start the engine with rotation of crankshaft by using gear trains. The camshaft and its associated parts control the opening and closing of two valves. Its associated parts are pushrod, rocker arms, valves springs and valves. The crankshaft through timing gears drives the camshaft and also drives engine oil pump shaft. Cams are made as integral parts of the camshaft are designed in such a way to open the valve at the correct timing and keep them open for the necessary duration. The camshaft in the block is driven either by timing gears or by using timing chain or timing belts. Camshaft timing is relationship between the camshaft and crankshaft. In most engine, the camshaft is driven by the timing chain or timing belts. But in this engine, the camshaft is driven by timing gears. Many newer engine designs place the crankshaft on the cylinder head. These engines are called overhead camshaft (OHC) engines. Engine oil pump shaft is also provided to flow the engine oil (lubricant) into many parts of engine block such as between pistons and cylinders, crankshaft, camshaft, gudgeon pin, connecting rod, rocker arms and valves spring, etc. Engine oil pump have two functions. Firstly, it is suction from the engine oil tank and delivered to the components of engine parts [1]. Engine oil pump are made as no luck of lubricant between rotating parts. Engine oil is circulated under pressure to the rotating bearings, the sliding pistons and the camshaft of the engine. This lubricates the bearings, allows the use of the higher-capacity fluid bearings and also assists in cooling the engine. Therefore, engine oil pump is essential for engine block [2].

In this paper, the author specializes Timing Gears Trains System of MICO Tractor Engine with detail approach.

II. DESIGN CONSIDERATION OF TIMING GEARS TRAINS AND ENGINE OIL PUMP GEAR

When a vehicle is starting from rest, the ignition switch is open, the starter motor is turn the flywheel and the crankshaft also turn counter clockwise. At the time, the timing helical pinions and gears related to each other.

Gears are defined as toothed wheels or multi lobes cams which transmit power and motion from one shaft to another by means of successive engagement of the teeth.

A. Classification of Gears

Gears are broadly classified into four groups;

- Spur gear,
- Helical gear,
- Bevel gear and
- Worm gears.

B. Helical Gear

Helical gear consists of infinite number of narrow spur gear, thus forming a cylindrical helix. There is a basic difference between spur and helical gears. While the teeth of spur gears are cut parallel to the axis of the shaft, the teeth of helical gears are cut in the form of helix on the pitch cylinder [3].

In spur gears, the contact between meshing teeth occurs along the entire face width of the tooth, resulting in a sudden application of the load which, in turn, results in impact conditions and generates noise in high speed applications.

In helical gears, the contact between meshing teeth beings with a point on the leading edge of the tooth and gradually extends along the diagonal line across the tooth. There is a gradual pick-up of load by the tooth, resulting in smooth engagement and quite operation even at high speeds.

There are two basic type types of helical gears parallel and crossed. Parallel helical gears on two parallel shafts. A right hand helical will always mesh with a left hand helical. Crossed helical gears are mounted on shafts with crossed axes. Their teeth may have the same or opposite hand of the helix. The discussion in this chapter is limited to design of parallel helical gears.

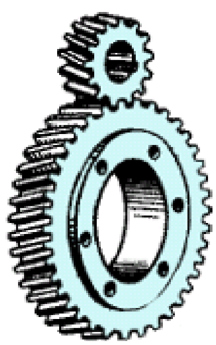

Figure 1: Parallel Helical Gear

Figure 2: Crossed Helical Gear

C. Gear Design

Gear tooth design involves primarily the determination of the proper pitch and face width for adequate strength, durability and economy of manufacture.

In a gear design it is necessary to calculate strength check and dynamic check. Strength point of view find the smallest possible module and corresponding face width [4].

Dynamic point of view find the endurance force F_0, wear force F_w and dynamic force F_d .Required condition F_0, $F_w \geq F_d$.

Economy of manufacture divided into three portions, 1st class commercial cut, carefully cut and precision cut.

D. Velocity Ratio (Transmission Ratio), V.R

Angular velocity ratio is the ratio of angular velocity of the pinion to the angular velocity of its mating gear. It is inversely proportional to the number of teeth on the two gears, and for spur gears it is also inversely proportional to the pitch diameters.

$$V.R = \frac{D_g}{D_p} = \frac{N_g}{N_p} = \frac{rpm_p}{rpm_g} \quad (1)$$

Where, D_p = pitch diameter of pinion, mm or m
D_g = pitch diameter of gear, mm or m
N_p = number of teeth on pinion
N_g = number of teeth on gear
rpm_p = rotational speed of pinion, rev/min
rpm_g = rotational speed of gear, rev/min

E. Base Design

The amount of force that can be transmitted to a gear tooth is a function of the $S_0 y$ product (load carrying capacity) as shown by the Lewis equation. For two mating gears, the weaker will have the smaller $S_0 y$ value [5].

When two mating gears are to be made of the same material, the smaller gear (pinion) will be the weaker and control the design.

F. Force On Gear Teeth

Power is received from engine by the clutch shaft rotating at engine speed. Thus there is a torque in the shaft that can be computer from the following equation [6]. The pitch line velocity is derived from the basic relationship for

$$v = \frac{\pi \times D \times rpm}{60} \quad (2)$$

D = pinion diameter, (mm or m)
rpm = rotational speed for weaker gear (pinion), (rpm)

The torsional moment acting on the shaft can be determined from this equation.

$$M_t = \frac{9550 \times kW}{rpm} \quad (3)$$

Where, M_t = Torsional moment, (Nm)
kW = Power being transmitted, (kW),
rpm = rotational speed for weaker gear (pinion), (rpm)

G. Transmitted Force

The transmitted force acts tangential to the pitch surface of the gear, actually transmits torque and power from the drive to the driver gear and acts in a direction perpendicular to the axis of the shaft carrying the gear. The transmitted force can be found by dividing the power transmitted by the pitch line speed. If the torque (M_t)is Newton-meter and D is in meter [7].

$$F_t = \frac{M_t}{D/2} \quad (4)$$

Where, F_t = transmitted force

The normal force, F_n, and radius force, F_r, can be computer from the known F_t by using the right triangle relations evident

$$F_t = F_t \times \tan(\varphi) \quad (5)$$

$$F_n = \frac{F_t}{\cos(\psi)\times\cos(\varphi_n)} \quad (6)$$

Where, φ = pressure angle of the tool form

H. Formative or Virtual Number of Teeth

The shape of the tooth would be that one generated on the surface of the pitch cylinder of the radius and the number of teeth on this surface is defined as the virtual number of teeth.

$$N_f = \frac{N}{\cos^2(\psi)} \quad (7)$$

Where, N_f = virtual number of teeth, (teeth)

Ψ = helix angle, (degree)

$$D = m \times N \quad (8)$$

Where, N = number of teeth

D = diameter of pinion

m = number module

I. Strength in Gear Teeth

The stress analysis of gear teeth is facilitated by considering the orthogonal force components, F_t and F_r.The tangential force, F_t, produces a bending moment on the gear tooth. The resulting bending stress is maximum at the base of the tooth in the tooth in the fillet that joins the involute profile to the tooth space. Wilferd Lewis developed the equation for the stress at the base of the involute profile, which is called the Lewis equation. In the design of the gear for strength, the pitch diameter is either known or unknown. If the pitch diameter is known, the following form of the Lewis equation may be used [8].

$$\left(\frac{1}{m^2 y}\right)_{all} = \frac{s\times k\times\pi^2}{F_t}\times\cos(\psi) \quad (9)$$

Where, s = allowable stress (N/mm^2)

m = module, (mm, m)

y = form factor based on virtual or formative number of teeth

k = 6 (max;), at helical gear

F_t = transmitted force, (N)

ψ = helix angle, (degree)

Where y is called Lewis form factor, values of which are tabulated in Table A.1.

This form factor y is a function of the tooth shape, which depends primarily on the tooth system and the number of the teeth on the gear [9]. If the pitch diameter is unknown, the following form of the Lewis equation may be used.

$$S_{ind} = \frac{2\times M_t}{m^3\times k\times\pi^2\times y\times N_w\times\cos(\psi)} \quad (10)$$

Where, S_{ind} = Index stress

M_t = Torsional moment, (Nm)

m = module, (mm or m)

y = form factor based on virtual or formative number of teeth

k = 6 (max;), at helical gear

N_w = number of weaker gear (pinion), (teeth)

ψ = helix angle, (degree)

The allowable stress s may be taken as approximately equal to the endurance limit of the material in released loading, crossed for stress concentration effects and multiplied by a velocity factor.

$$S_{all} = S_{all}\times V.F \quad (11)$$

$$S_{all} = S_{all}\times\frac{5.6}{5.6+\sqrt{v}} \quad (12)$$

Where, S_{all} = endurance stress (one third of the ultimate strength of material)

V.F = velocity factor

v = pitch line velocity, m/s

In the design check for strength, the pitch diameter is either known or unknown.

Reduce k value,

$$k_{red} = k_{max}\times\frac{S_{ind}}{S_{all}} \quad (13)$$

Where, $k_{max;}$ = 6 (for helical gear)

Then face width, b

$$b_{max;} = k_{max;}\times\pi\times m \quad (14)$$

$$b_{min;} = k_{red;}\times\pi\times m \quad (15)$$

Where, m = number of module

$$F_0 = s_o\times b\times y\times\pi\times m\times\cos(\psi) \quad (16)$$

Where, F_0 = the endurance force

s_0 = endurance stress (one third of the ultimate strength of material)

b = face width of gear (pinion),(mm or m)

y = form factor based on virtual or formative number of teeth

m = module, (mm or m)

ψ = helix angle, (degree)

The allowable stress may be taken as approximately equal to the endurance limit of the material in released loading, corrected for stress concentration effects and multiplied by a velocity factor.

The limiting endurance strength load, F_0, must be equal to or greater than the dynamic load, F_d.

The limiting wear load, F_w, for helical gears be determined by the Buckingham equation for wear,

$$F_w = \frac{D_p \times b \times K \times Q}{\cos^2(\psi)} \quad (17)$$

Where, D_p = pitch diameter of pinion, (mm or m)

Q = ratio factor

b = face width of gear (pinion), (mm or m)

K = stress fatigue factor, (N/m^2)

$$Q = \frac{2 \times D_g}{D_p + D_g} \quad (18)$$

Where, D_g = pitch diameter of gear, (mm or m)

$$K = \frac{Ses^2 \times \sin(\varphi_n) \times [\frac{1}{E_p} + \frac{1}{E_g}]}{1.4} \quad (19)$$

Where, S_{es} = surface endurance limit of a gear pair,

E_p = Modules of elasticity of the pinion material,

E_g = Modules of elasticity of the gear material,

φ_n = pressure angle measured in a plane normal to a tooth

The surface endurance limit may be estimated from

$$S_{es} = [2.75(BHN_{avg}) - 70] \quad (20)$$

Where, BHN_{avg} = average brinell hardness number of the pinion and gear material

The wear load, F_w is an allowance load and must be greater than the dynamic load, F_w. The dynamic load, F_d, for helical gears is the sum of the transmitted load and an incremental load due to dynamic effects:

$$F_d = F_t + \frac{21 \times v(bC\cos^2(\psi) + F_t) \times \cos(\psi)}{21 \times v + \sqrt{bC\cos^2(\psi) + F_t}} \quad (21)$$

Where, F_d = Dynamic load, (N)

v = Pitch line velocity, (m/sec)

b = face width of gear (pinion), (mm or m)

F_t = Transmitted force, (N)

C = Dynamic factor, (N/m)

$$\tan(\varphi_n) = \tan(\varphi)\cos(\psi) \quad (22)$$

Where, φ_n = pressure angle measured in a plane normal to a tooth

φ = pressure angle of the tool form

ψ = helix angle, (degree)

$$BHN_{avg} = \frac{BHN_p + BHN_g}{2} \quad (23)$$

Where, BHN_{avg} = average brinell hardness number of the pinion and gear material BHN_p = brinell hardness number of the pinion material

BHN_g = brinell hardness number of the gear material

III. DESIGN CALCULATION AND RESULTS

A. Design Calculation of Gear Trains

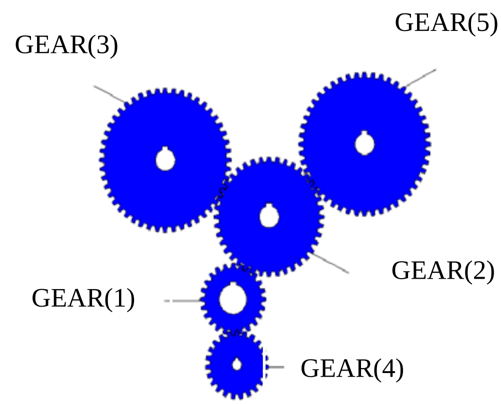

Figure 3: Crankshaft Pinion and Timing Gear,

Where, Gear (1) is Crankshaft gear (pinion)
Gear (2) is Timing gear (pinion)
Gear (3) is Camshaft gear
Gear (4) is Engine oil pump gear (pinion)
Gear (5) is Pump shaft gear

B. Design calculation of crankshaft pinion and timing gear

Type of gear = Helical gear

Pressure angle, φ = 20º full depth

Helix angle, ψ = 23º

Number of crank pinion, N_1 = 24 teeth

Number of timing gear, N_2 = 40 teeth

Crank pinion speed, rpm_1 = 1800 rpm

Ultimate tensile strength, S_u = 1120MPa

Endurance stress, S_0 $= \frac{S_u}{3}$ =373.3333MPa

Yield strength, S_y = 807MPa

Power of Tractor engine = 80hp (59.68kW)

Modules of elasticity, E = 207GPa

Brinell hardness number, BHN = 514

Timing gear speed, rpm_2 $= 1800\text{rpm} \times \frac{N_1}{N_2}$

$$= 1800 \times \frac{24}{40} = 1080\text{rpm}$$

Same material, the smaller gear (pinion) will be the weaker and control the design.

Unknown Diameter case,
Strength Check,

$$M_t = \frac{9550 \times kW}{rpm_1}$$

$$= \frac{9550 \times 59.68}{1800} = 316.6356 \text{ Nm}$$

$$N_{f_1} = \frac{N_1}{\cos^3(\psi)}$$

$$= \frac{24}{\cos^3(23)} = 30.7704 \approx 31 \text{ teeth}$$

$N_{f_1} = 31$ teeth, $\varphi = 20°$ full depth

$N_{f_1} = 31$ teeth, $y = 0.115$

$$S_{ind} = \frac{2 \times M_t}{m^3 \times k \times \pi^2 \times y \times N_w \times \cos(\psi)}$$

$$= \frac{2 \times 316.6356}{m^3 \times 6 \times \pi^2 \times 0.115 \times 24 \times \cos(23)}$$

$$= \frac{4.2092}{m^3} \qquad \text{(A)}$$

Assume V.F $= \frac{1}{2} = 0.5$

$$S_{all} = S_0 \times V.F$$

$$= 373.3333 \times 10^6 \times 0.5$$

$$= 186.6667 \text{ MPa} \qquad \text{(B)}$$

(A) = (B)

$$m = 2.8258 \times 10^{-3} m = 2.8252 \text{ mm}$$

Standard module series 1, 1.25, 1.375, 1.5, 1.75, 2, 2.25, 2.5, 2.75, 3, 3.5,…

Try Standard module, m = 3.5,

$$D_1 = m\, N_1 = 3.5 \times 24 = 84 \text{ mm}$$

$$D_2 = m\, N_2 = 3.5 \times 40 = 140 \text{ mm}$$

$$v = \frac{\pi \times D_1 \times rpm_1}{60}$$

$$= \frac{\pi \times 0.084 \times 1800}{60} = 7.9168 \text{ m/s}$$

$$S_{all} = S_0 \times \frac{5.6}{5.6 + \sqrt{v}}$$

$$= 373.3333 \times 10^6 \times \frac{5.6}{5.6 + \sqrt{7.9168}} = 248.4842 \text{ MPa}$$

Sub in equation (A)

$$S_{ind} = \frac{4.2092}{m^3}$$

$$= \frac{4.2092}{(3.5 \times 10^{-3})^3} = 98.1738 \text{ MPa}$$

$$k_{red} = k_{max;} \times \frac{S_{ind}}{S_{all}}$$

$$= 6 \times \frac{98.1738}{248.4842} = 2.3705$$

$$b_{max;} = k_{max;} \times \text{Standard module, } m \times \pi$$

$$= 6 \times 3.5 \times \pi = 65.9734 \text{ mm}$$

$$b_{min;} = k_{min;} \times \text{Standard module, } m \times \pi$$

$$= 2.3705 \times 3.5 \times \pi = 26.065 \text{ mm}$$

Used b =32 mm

$$F_0 = S_0 \times b \times y \times \pi \times m \times \cos(\psi)$$

$$= 373.3333 \times 10^6 \times 32 \times 10^{-3} \times 0.115 \times \pi \times 3.5 \times 10^{-3} \times \cos(23)$$

$$= 13.9056 \text{ kN}$$

$$F_w = \frac{D_1 \times b \times K \times Q}{\cos^2(\psi)}$$

$$Q = \frac{2 \times N_2}{N_1 + N_2} = \frac{2 \times 40}{24 + 40} = 1.25$$

$$\tan(\varphi_n) = \tan(\varphi) \times \cos(\psi)$$

$$\varphi_n = \tan^{-1}(\tan(20) \times \cos(23))$$

$$\varphi_n = 18.523°$$

$$S_{es} = 2.75 \times (BHN)_{avg} - 70$$

$$= 2.75 \times (514) - 70$$

$$= 1343.5 \text{ MN/m}^2$$

$$K = \frac{Ses^2 \times \sin(\varphi_n) \times [\frac{1}{E_p + E_g}]}{1.4}$$

$$= \frac{(1343.5 \times 10^6)^2 \times \sin(18.523) \times [\frac{2}{207 \times 10^9}]}{1.4}$$

$$= 3957346.63$$

$$F_w = \frac{84 \times 10^{-3} \times 32 \times 10^{-3} \times 3957346.63 \times 1.25}{\cos^2(23)} = 15.6925 \text{ kN}$$

v =7.9168 m/s $e = 0.01$ (Assume Precision Gears)

error = 0.01, $\varphi = 20°$ full depth

Using the Steel C =114kN/m

$$F_t = \frac{M_t}{D_1/2} = \frac{316.6356}{84 \times 10^{-3}/2} = 7.5389 \text{ kN}$$

$$F_d = F_t + \frac{21 \times v \times (b \times C \times \cos^2(\psi) + F_t) \times \cos(\psi)}{21 \times v + \sqrt{b \times C \times \cos^2(\psi) + F_t}}$$

$$= 7538.9 + \frac{21 \times 7.9168 \times (32 \times 10^{-3} \times 114 \times 10^3 \times \cos^2(23) + 7538.9) \times \cos(23)}{21 \times 7.9168 + \sqrt{32 \times 10^{-3} \times 114 \times 10^3 \times \cos^2(23) + 7538.9}}$$

$$= 13.5784 \text{ kN}$$

$F_0, F_w > F_d$ (∴Design is satisfied)

∴ module, $m = 3.5$, $D_1 = 84$ mm, $D_2 = 140$ mm, facewidch, b = 32 mm

IV. ANALYSIS OF GEAR USING AUTODESK

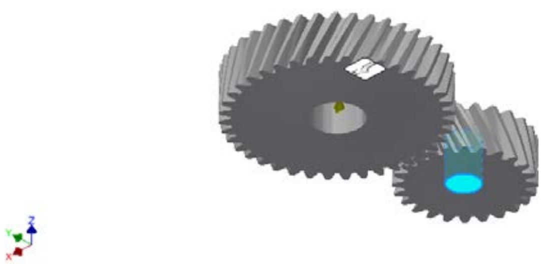

Figure 4: Model Exported in Autodesk

A. Material Properties

Name	:AISI 1050 Steel
Model type	:Linear Elastic Isotropic
Default failure criterion	:Max von Mises Stress
Yield strength	:807 MPa
Tensile strength	:1120 MPa

B. Mesh Information

Avg. Element Size (fraction of model diameter)	:0.1
Min. Element Size (fraction of avg. size)	:0.05
Grading Factor	:1.5
Max. Turn Angle 60 deg	
Create Curved Mesh Elements	:No
Use part based measure for Assembly mesh	:Yes

C. Results

Name	Minimum	Maximum
Volume	909480 mm^3	
Mass	7.13942 kg	
Von Mises Stress	0.00188894 MPa	87.3993 MPa
1st Principal Stress	-8.03851 MPa	51.3274 MPa
3rd Principal Stress	-96.0945 MPa	9.2745 MPa
Displacement	0 mm	0.0272425mm
Safety Factor	2.36844 ul	15 ul

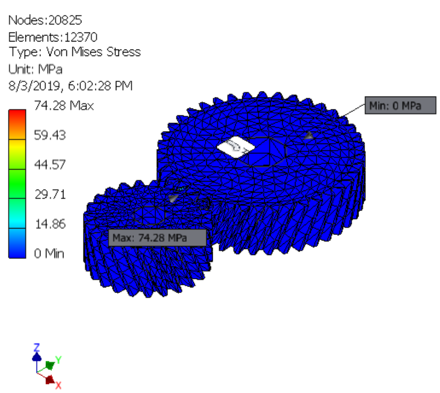

Figure 5: von Mises Stress

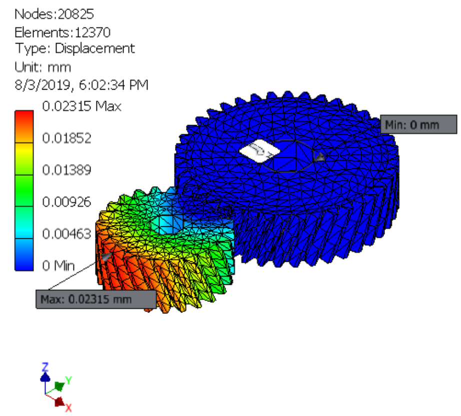

Figure 6: Displacement

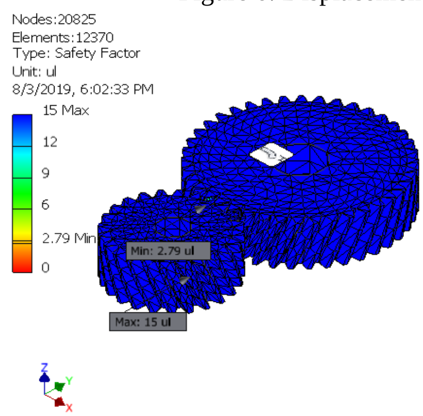

Figure 7: Factor of Safety

IV. CONCLUSION

Present day competitive business in global market has brought increasing awareness to optimize gear design. Current trends in engineering globalization require results to comply with various normalized standards to determine their common fundamental and those approaches needed to identify "best practices" in industries. From the study of effect of various parameters (wear load, dynamic tooth load, beam strength) on the design of helical gears for MICO Tractor Engine (80hp). The induced dynamic tooth load are much lower than those of the results obtained theoretically, thus the design is safe from the structural point of view.

APPENDIX

Table A.1. Form Factors 'y' for. Use in Lewis Strength Equation

Number of teeth	$14\frac{1}{2}$° Full-Depth Involute or Composite	20° Full Depth Involute	20° Stub Involute
12	0.067	0.078	0.099
13	0.071	0.083	0.103
14	0.075	0.088	0.108
15	0.078	0.092	0.111
16	0.081	0.094	0.115
17	0.084	0.096	0.117
18	0.086	0.098	0.120
19	0.088	0.100	0.123

21	0.092	0.104	0.127
23	0.094	0.106	0.130
25	0.097	0.108	0.133
27	0.099	0.111	0.136
30	0.101	0.114	0.139
34	0.104	0.118	0.142
38	0.106	0.122	0.145
43	0.108	0.126	0.147
50	0.110	0.130	0.151
60	0.113	0.134	0.154
75	0.115	0.138	0.158
100	0.117	0.142	0.161
150	0.119	0.146	0.165
300	0.122	0.150	0.170
Rack	0.124	0.154	0.175

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