Design and Analysis of Gear Tooth for Engine Gear Train to Avoid Failure due to Bending and Pitting by Varying CR

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Abstract — In the Engine Gear Train of Vehicle mostly problem produced to failure of gear due to no perfect mesh, higher load carrying capacity with less installed volume and weight. The gears used in the vehicles should also have lesser noise and vibration. Even though helical gears will meet the requirement, they are prone for additional axial thrust problem. High contact ratio (HCR) is one such gearing concept used for achieving high load carrying capacity with less volume and weight. The previous study deals with various parameters affecting of bending and pitting stresses but increase gear contact ratio with avoid to failure of bending and pitting stress with same engine gear train and same center distance had not been carried out so for. In this paper, we are presenting details on how to develop new gear tooth profile to avoid failure due to bending, pitting stress by Improve Contact ratio of gear for the Engine gear Train. A three dimensional model of Cam and Crank gears is analyzed by using ANSYS 14.5 software. The study involves design, modeling, meshing and post processing of gears using single window modeling concept to avoid contact convergence and related numerical problems.

(Key words: Engine gear train, AMF, Contact ratio)

I. INTRODUCTION

A majority of the heavily loaded transmissions engine applications use gears with a contact ratio (CR) range of 1.3 to 1.7. So, the number of teeth in engagement at any instant is either one or two. Many gear designs use increased pressure angle for increasing the load carrying capacity of gears with fixed module and center distance, but the contact ratio decreases. Tooth dynamic loads and noise increase due to decreased pressure angle. Hence increasing the load Carrying capacity of gears for the above conditions can be done by the design of gears with a higher contact ratio.

Tolerance stack up is one of the most important exercises of design team to ensure the interchangeability of the components and the sub-assemblies in the product having a large product family. Optimized allocation of tolerances between the components in a product helps to reduce the conflicts between the designer and the manufacturer. Automotive Engine gear train is one of the most important and critical aggregate of any vehicle in which the assembly of gear train performs the function as required. In the gear train, numbers of gears are meshed with each other and transfer the torque on wheels as per the required performance of the vehicle. When the gears are meshed and rotates some noise would generate because of the rolling and sliding action of the gears.

All the referred papers refer to various regeneration methods of modified gear tooth by which we can avoid failure due to bending & pitting stress using change gear parameters. In our case to avoid failure of gear tooth due to bending and pitting by Influence of addendum modification factor(AMF), Module and Helix angle on CR and comparing on the. Also we will be able to improve contact ratio and height load carrying capacity of gear and ANALYSIS by using ANSYS 14.5.

II. ENGINE GEAR TRAIN

The engine timing gear train is one of the vital set of components in the engine. It is used to transfer power from Driver to the Driven and to maintain the valve timing in the engine. Hence, power and rotational motion from the crankshaft to the camshaft has to be transferred constantly without any abruptions for the engine to function properly. Timing of the engine can be maintained by gears, belts or chains. The majority of medium and almost all heavy duty engines have timing gears owing to its capability to withstand higher loads and since it has higher longevity compared to belts and chains. The gear train can be placed either in the front or the rear of the engine. Earlier engines and some present heavy duty engines feature gear train in the front of the engine. Most of the heavy duty engines have the timing gears in the rear of the engine. The crankshaft and camshaft positions are the main controlling factors for the layout of the timing gear train. Additionally, the layout of the gear train is dependent on the packaging possibilities in the engine compartment the gears of the gear train are mounted on shafts supported by the cylinder block. Bearings are very essential as well in order to minimize loss of power from friction and also to ensure smooth running of the gear. The bearings are mounted between the gears and the shafts.

The Engine Gear Train is formed by mounting gears on a frame so that the teeth of the gears engage. It transmitted torque Crankshaft gear to Camshaft and Camshaft gear transmit to both Air compressor gear and FIP (Fuel injection pump) constantly without any abruptions as shown in Figure 1.

The gear train is a source of noise generation due to motion between gears causing impacts. It is also one of the major contributors to engine air borne noise and structural vibrations after the combustion and injection noise. There are many sources of noise and vibration generation from a gear train, some are mentioned below:

- Gear rattle between meshing gears
- Gear whine at higher speeds
- Noise generated from auxiliary components
- Noise and vibration propagation from auxiliary components to the gear train.

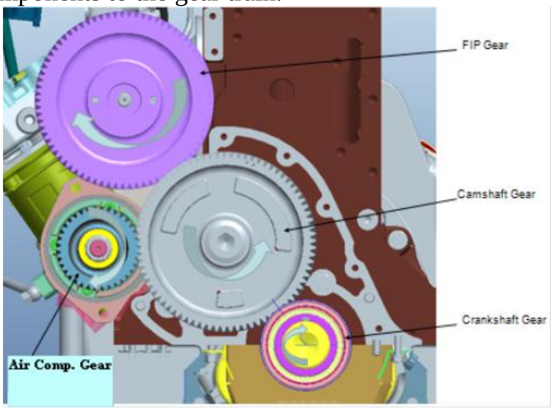


Figure 1. Engine Gear Train

III. SPECIFICATION OF GEAR DATA

Existing Camshaft and Crankshaft gear specification as shown Table 1.

Parameters	Cam Shaft	Crank Shaft	Units
No. of Teeth (z)	78	39	
PCD (d)	181.77	90.88	mm
Module (m)	2.05	2.05	mm
Pressure Angle (α)	20	20	°C
Helix Angle (ψ)	28.4	28.4	°C
Face width(b)	26	28	mm
Addendum (a)	2.61	1.7	mm

Table 1. Cam – Crank gear specification

Normal forces acting on engine gear pair as shown Table 2.

Gear Pair	Normal Force
Crankshaft -Cam Shaft Gear	4000 N
CAM Shaft -FIP Gear	2000 N
CAM Shaft -Air Compressor Gear	3000 N

Table 2. Forces on gear pair

3.1 Stress calculations

Two fundamental stress equations are used in the AGMA methodology, one for bending stress and another for pitting resistance (contact stress).

3.1.1 Bending stress

$\sigma b = Ft * Ko * Kv * Ks * KH * KB /(b* mt * J)$ --- (1)

Where,

Ft = Tangential Force (N)

Ko = overload factor

Kv = Dynamic factor

Ks = Size factor

KH = Load – Distribution Factor

KB = Rim- Thickness Factor

b = face width (mm)

mt = Transvers metric module

J = geometry factor for bending strength

$\sigma b = 164.54\ N/mm^2$ ----------------------- (for cam gear)

σb = 152.28 N/mm2 ---------------------- (for crank gear)

3.1.2 Pitting stress

$\sigma c = ZE * \sqrt{(Ft * Ko * Kv * Ks * KH * ZR) /(dp* b * I)}$ –(2)

Where,

ZE = elastic coefficient

ZR = Surface condition factor

dp = PCD on pinion (crank gear)

I = Geometrical factor for pitting resistance

E = Modulus of elasticity = 2* 10^5 N/mm2

v = Poison ratio = 0.3

σc = 384.13 N/mm^2 ----------------------- (for cam gear)

σc = 383.51 N/mm^2 ---------------------- (for crank gear)

IV. CONTACT RATIO CALCULATIONS

Contact ratio is defined as a measure of the average number of teeth in contact during the period in which a tooth comes and goes out of contact with the mating gear relative to the pitch. The contact ratio for a general gear is around two, if the contact ratio for gears is greater than two then it is referred as high contact ratio gears. Increasing the contact ratio of gears has proven to reduce noise and also have better performance.[1]

The contact ratio of the gear pair plays an important role in increasing the load carrying capacity of gears [11].

High contact ratio can be achieved by different ways namely:

- Increasing the number of teeth;
- Lowering the pressure angle;
- Increasing the addendum factor.

If increasing the number of teeth directly effect on gear thickness means to Increasing bending stress Hence gear failure due to bending other option to Increase gear teeth to use composite material and check Stress on it to do safe for Bending. Second and third is better to use Increasing the Addendum factor at same Center distance also change gear profile and do stress analysis for safe Bending as well as pitting.

4.1 Addendum modification factor –

The addendum modification gear teeth is carried out in order to avoid undercut, improve strength and running properties or to adjust the Centre distance. The addendum modification is generally recommended in the following cases:

a) Gears with critical number of teeth;

b) Non-standard Centre distance;

c) To obtain balanced strength;

d) To obtain reduction in sizes.

However, load capacity and running equalities do not attain optimum values, but the gear pairs of this system have essentially better equalities than X-2000 gear pairs.

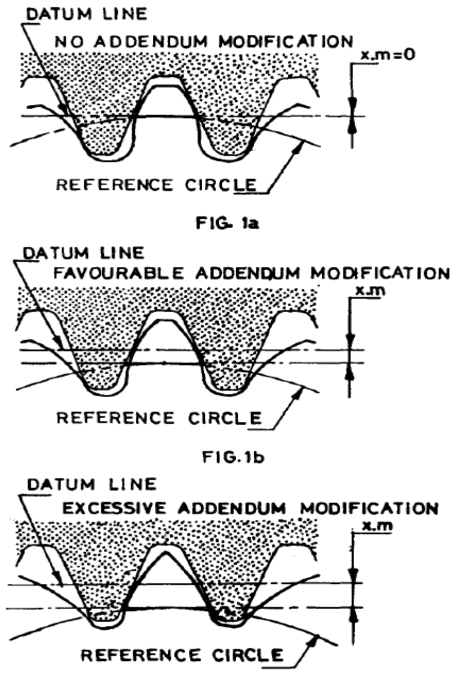


Figure 2. Addendum modification

4.2 Influence of Addendum Modification on CR:

Variation of addendum modification factor to calculate gear contact ratio for Cam and Crank gear pair as shown Table 3.

	CASE-I		CASE-II		CASE-III	
Parameters	Cam Shaft	Crank Shaft	Cam Shaft	Crank Shaft	Cam Shaft	Crank Shaft
No. of Teeth (z)	78	39	78	39	78	39
PCD (d)	181.77	90.88	181.77	90.88	181.77	90.88
Module (m)	2.05	2.05	2.05	2.05	2.05	2.05
Press Angle (α)	20	20	20	20	20	20
Helix Angle (ψ)	28.4	28.4	28.4	28.4	28.4	28.4
Addendum (a)	2.61	1.7	2.62	1.698	2.663	1.66
Dedendum(d)	3.39	4.71	3.70	4.67	3.65	4.72
Gear Ratio (u)		2		2		2
Sum of AMF		-0.631		-0.631		-0.631
Sum of addendum modification		-1.293		-1.293		-1.293
Addendum Modification	-0.135	-1.158	-0.128	-1.165	-0.128	-1.2
Contact ratio		1.74		1.753		1.755

Table 3. Influence of Addendum Modification on CR

There is minor improvement of contact ratio (1.74 to 1.755), so we will go to change module for improve contact ratio.

4.3 Influence of Module on CR:

When change module of gear 2.05 to 2.5 and 2.5 to 2.75 as shown Table 4 effect to change all gear parameters like No. of teeth, addendum, deddendum also change addendum modification factor as same helix angle. Below table shows details of change module to improve contact ratio also improved bending stress because of decrease no. of teeth and increase gear thickness.

	CASE-I		CASE-II		CASE-III	
Parameters	CAM Shaft	Crank Shaft	CAM Shaft	Crank Shaft	CAM Shaft	Crank Shaft
No. of Teeth (z)	78	39	64	32	59	30
Helix Angle (ψ)	28.4	28.4	28.4	28.4	28.4	28.4
PCD (d)	181.77	90.88	181.89	90.94	184.44	93.78
Module (m)	2.05	2.05	2.5	2.5	2.75	2.75
Addendum (a)	2.61	1.7	3.215	2.334	3.55	2.68
Dedendum(d)	3.39	4.71	4.437	5.49	4.86	5.92
Addendum Modification (Xm)	-0.135	-1.158	-0.135	-1.24	-0.135	-4.62
Contact ratio		1.74		1.81		1.83

Table 4. Influence of Module on CR

4.4 Influence of Helix angle on CR:

To Influence of helix angle at identical module, that cases improved gear contact ratio up to 1.88 also effect of no. of teeth, Addendum & dedendum as shown below table 5.

	CASE-I		CASE-II		CASE-III	
Parameters	CAM Shaft	Crank Shaft	CAM Shaft	Crank Shaft	CAM Shaft	Crank Shaft
No. of Teeth (z)	78	39	66	33	68	34
Helix Angle (ψ)	28.4	28.4	25	25	22	22
PCD (d)	181.77	90.88	182.06	91.03	183.35	91.67
Module (m)	2.05	2.05	2.5	2.5	2.5	2.5
Addendum (a)	2.61	1.7	3.21	2.33	3.21	5.49
Dedendum(d)	3.39	4.71	4.44	5.49	4.44	5.49
Addendum Modification (Xm)	-0.135	- 1.158	-0.135	-1.24	-0.135	-4.62
Contact ratio		1.74		1.84		1.88

Table 5. Influence of Helix angle on CR

As above study, we observed that Module is directly proportional to CR and Helix angle is inversely proportional to CR.So We are Design New gear profile parameters having increasing module up to 2.75mm and decreasing helix angle up to 22º is better option to increase contact ratio.

V. DESIGN OF NEW GEAR TOOTH

To calculate gear parameters for selected module and helix angle. Gear parameters value as shown in Table 6.

Parameters	CAM Shaft	Crank Shaft
No. of Teeth (z)	62	31
Helix Angle (ψ)	22	22
PCD (d)	181.77	90.88
Module (m)	2.75	2.75
Addendum (a)	3.55	2.68
Dedendum(d)	4.86	5.93
Addendum Modification (Xm)	-0.135	-2.5403
Contact ratio		1.9

Table 5. CR of New gear Design

The parametric specification of 62 and 31 teeth helical gear. These design specifications has been arrived from KISSsoft application software. The profile of the helical gear with 62 teeth (cam gear) and 31 teeth (crank gear) having Circular and Trochoidal root fillet are called in 3D Modeling software Pro-E and generate 3D model as shown in the Figure 3.

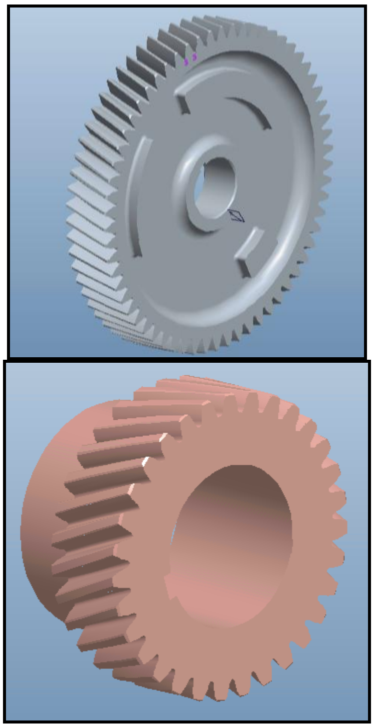


Figure 2. 3D model of Cam – Crank gear

5.1 Stress calculations

To calculate Bending stress for new gear by using Equ. (1)

σb = 139.47 N/mm^2 ---------------------- (for cam gear)

σb = 128.9 N/mm2 ---------------------- (for crank gear)

Also calculate Pitting stress by using Equ. (2)

σc = 396.71 N/mm^2 ----------------------- (for cam gear)

σc = 395.87 N/mm^2 ---------------------- (for crank gear)

Each parameter has to be individually analysed and the result shows that bending stress of old gears is greater than new one for both Cam and Crank gears i.e. New gear design is better factor of safety than old one.Aslo compare pitting stress for old and new gear is not much increased and factor of safety also minor decreased, To observe contact ratio is increased for new Cam and Crank gear, so pitting failure is neglected.

VI. FEA ANALYSIS FOR GEAR TOOTH

Finite element analysis (FEA) is one of the most popular engineering analysis methods for linear & nonlinear problems. FEA requires a finite element mesh as a geometric input. This mesh can be generated directly from a solid model for the detailed part model designed in a three-dimensional (3D) CAD system. Since the detailed solid model is too complex to analyze efficiently, some simplification with an appropriate idealization process including changing material and reducing mesh size in the FE model is needed to reduce the excessive computation time.

To applied tangential force on old cam and new cam gear and to show von misses stress as Figure4 and Figure5 respectively.

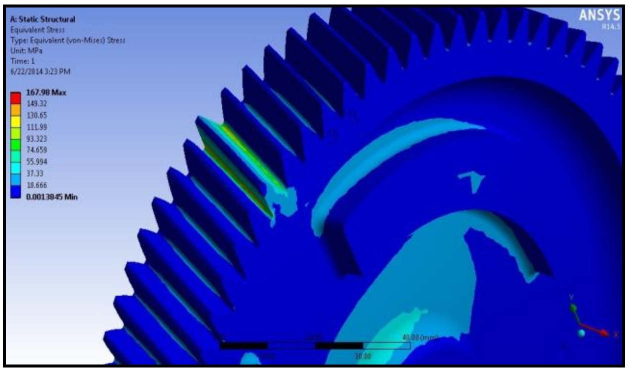


Figure 4. Stresses of old Cam gear tooth

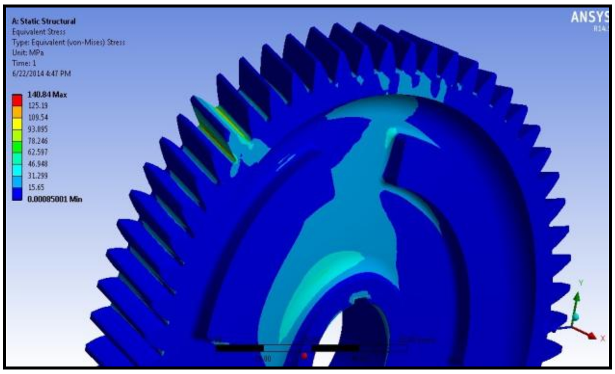


Figure 5. Stresses of new Cam gear tooth

To applied tangential force on old crank and new crank gear and to show von misses stress as Figure6 and Figure7 respectively.

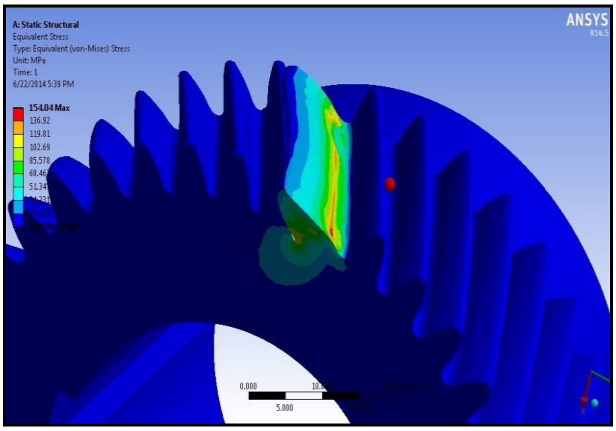


Figure 6. Stresses of old Crank gear tooth

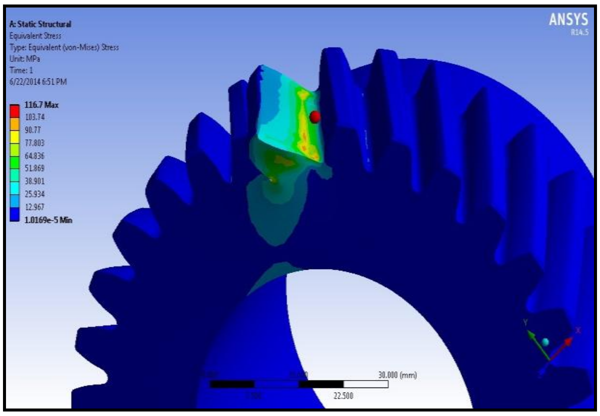
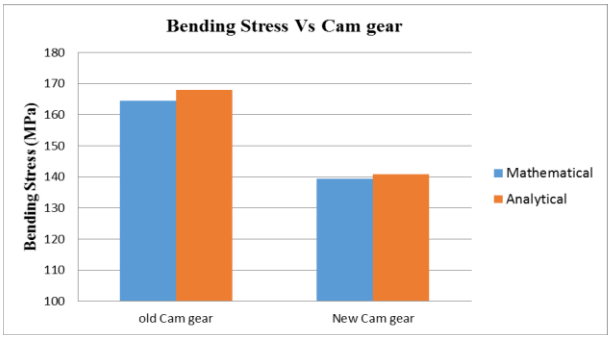


Figure 7. Stresses of new Crank gear tooth

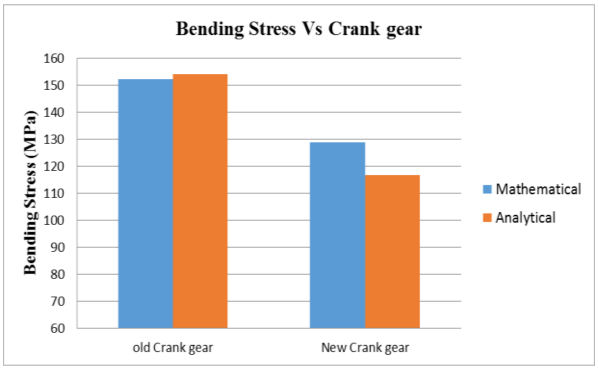
Bending stress of Cam gears are 167.98Mpa and 140.84Mpa for old and new Cam gear respectively and bending stress of Crank gears are 154.04Mpa and 116.7Mpa for old and new Crank gear respectively so once again proved new gear pair is safer then old by using ANSYS 14.5.

VII. RESULT AND DISCUSSION

Mathematical result compare with analytical result for Cam and Crank gears. As observation bending stress decreases of new gear as compare to old. That means new gear pair is safer than old gear pair. Graph 1 shows bending stress Vs old and new Cam gear and Graph 2 shows bending stress Vs old and new Crank gear.



Graph 1. Bending stress Vs old and new Cam gear



Graph 2. Bending stress Vs old and new Crank gear

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CONCLUSION

1. Bending stress on new gear tooth is less as compare to old gear tooth. Average percentage reduction in stress value is about 15.3% for Cam gear and 15.4% for Crank gear.

2. The contact ratio of new gear pair (1.9) is more as compare to old gear pair (1.74).

3. As we varied module of both mating gear from 2.05 to 2.75 keeping remaining parameters has same, we have observed that bending stress on the tooth of Cam and Crank gear is reduced where as it does not an appreciable in the pitting stress. The contact ratio of Cam and Crank gear is increase up to 1.88.

4. As we varied helix angle of both mating gear from 28.4° to 22° keeping remaining parameters has same, we have observed that bending stress on the tooth of Cam and Crank gear is reduced where as it does not an appreciable in the pitting stress. The contact ratio of Cam and Crank gear is increase up to 1.86.

5. Increase module and decreasing helix angle is better solution for improve contact ratio and gear pair safe for bending.

6. The ANSYS result shows close approximation with mathematical calculation values of stresses which has been applied view as with bending stress on gear tooth.

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