Design Analysis of Flywheel Ring Gear Failure

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Abstract- The work portraits the basic gear design calculation based on the ISO 6336 gear rating standards. Result come out after re-calculation of ring gear, FOS safety for beam strength static and dynamic as well as pitting stress is less than 1.2. On the basis of bench marking of the ring gear analysis improves the FOS for the same. Gaining a better understanding of product failures is important to help prevent future failures. To model spur gears using SABR, a three-dimensional model is adopted. The tooth load distribution is highly sensitive to gear body deflections in torsion. The bending stress on gear tooth is also due to the manufacturing errors. Quasi –static iterations of the ring gear pairs how significant decrease in the damage on the gears for the given torque load. Studying the history of product failures may generate some insight in to the reasons for this failure and create a list of factors that may increase the opportunity for success, but there are no guarantees.

Keywords: **Design Analysis, Flywheel Ring Gear, Design Calculation.**

I. INTRODUCTION

Engines is feedback system which is once started, rely on the motion from each cycle to initiate the next cycle. In a four-stroke inline engine, the third stroke releases chemical energy from the fuel, powering the exhaust stroke and the first two strokes of engine then the next cycle, as well as powering the engine's external load. When getting this feedback started, the first two strokes must be continuing the power in some other way. The starter motor is used for initial purpose and then is not needed once engine is running.

Diesel engine flywheel ring gear and starter gear are normally located at the rear of the engine. These gears are responsible for start engine. Some large engines may incorporate starter gears in the front.

- Starter Motor
- Starter motor Pinion gear
- Flywheel
- Ring gear

Diesel engine gear generally cast or forged alloys that are generally heat tempered and surface hardened, carburized, nitride or carbonitrided. The teeth are milled in manufacturing to spur design. Combination of both these design are used in engine is very important factor. Spur gear designs don't have force thrust force but they have less contact area and produced more noise compared to helical design because of hammering effects caused by tooth flanks collision.

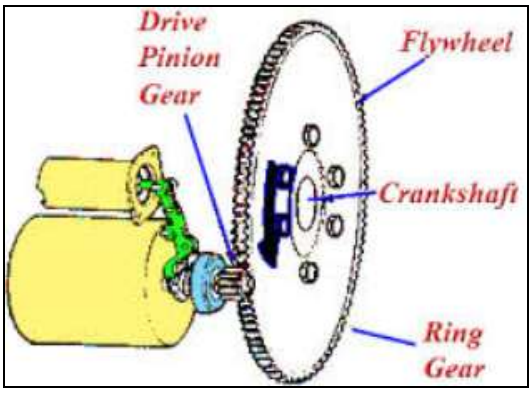

Figure 1- Flywheel ring gear and Starter Motor Pinion

The design of gears requires an iterative approach to optimized design parameters, which governs both the kinematics as well as strength performance. Due to the complex combinations of these parameters, conventional design practice to become complicated and time consuming. It involves selection of appropriate information from large amounts of engineering standards data available in engineering catalogues and handbooks. While the knowledge of gearing design is vast, however, there is acute paucity of

research on comparative analysis between various standards for gear design. The ISO and DIN standards almost similar and ISO forms the base for the British standards.

The ISO 6336 standards give the load capacity of spur and helical gears. Several methods for the calculation of load capacity, as well as for the calculation of various factors, ne starter gear. The noise produced by the starter gear is permitted by ISO, so the directions in standards are complex and flexible. The formula in ISO 6336 is intended to establish a uniformly method for calculating the pitting resistance.

II. LITERATURE SURVEY

DEDENADUM MODIFICATION AND STANDARD SPUR GEAR:

When generating tooth flanks on a cylindrical gear by means of a tooth-rack like tool, a straight pitch line parallel to the datum line of tool rolls on the reference circle. The distance $(x.m_n)$ between the straight pitch line and datum line of tool is the addendum modification and x is the addendum modification coefficient.

An addendum modification is positive, if the bottom line of the tool is displaced from the reference circle towards the tips, and it's negative if the datum line is displaced towards the roots of the gear. This is true for both the external and internal gears. In the case of internal gears the tip points is at the inside.

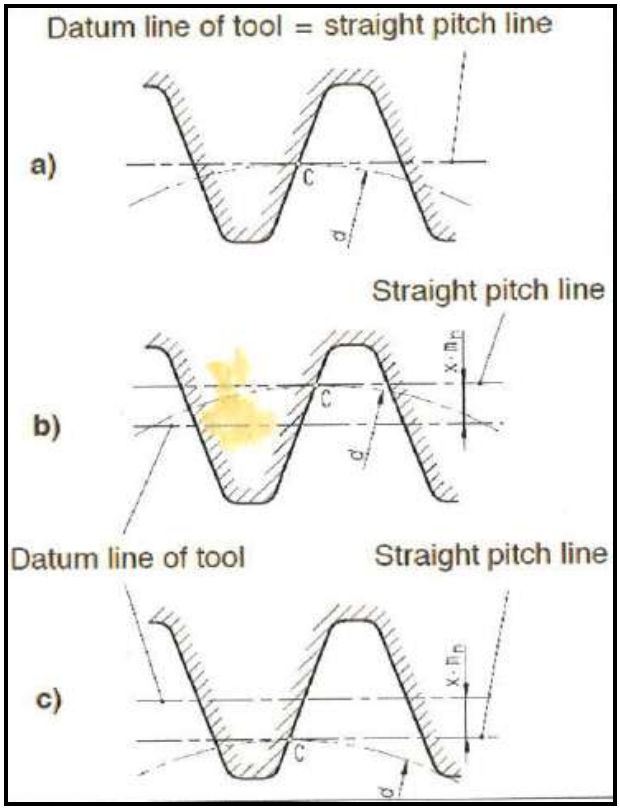

Figure 2 - Different position of the datum line

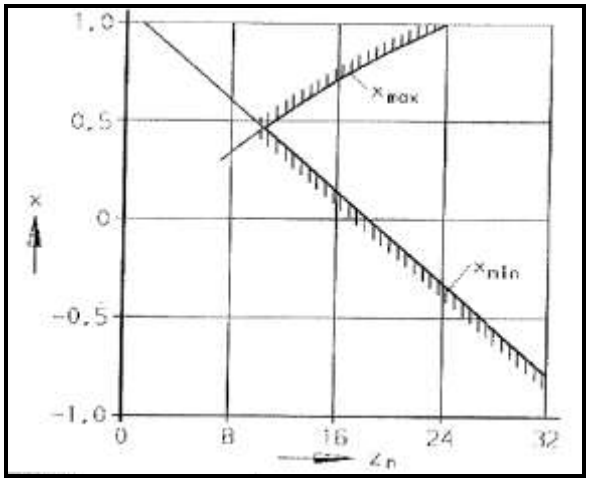

Figure 3 - Addendum modification limit X_{max} and X_{min} for external gears depends on the virtual number of teeth

The addendum modification limits X_{min} and X_{max} are represents dependent on the virtual number of teeth,

Z_n= Z / (cosβ.$cos^2\beta_b$)

The upper limit X_{max} takes into accounts the intersection circle of the teeth and applies to a normal crest within the normal section of s_{an} = 0.25m_n. When the following below the lower limits X_{min} this result in an undercut which shortens the usable involutes and weakens the tooth roots.

Z_n A positive addendum modification results in a greater tooth width and thus in an increase in the tooth root carrying capacity. In the case of small number of teeth these

K= (a-a_d)/m_n-(X_1+X_2)

Considerably stronger effects than in the case of larger ones. One mostly strives for a greater addendum modification on pinion than on gears in order to achieve equal tooth root carrying capacity for both gears.

In the case of an unmodified gear pair (a zero gear pair), both gear have as addendum modification coefficient X_1=X_2=0 (Zero gears). In the case of gear pair at reference center distance, both gear have addendum modification (modified gear), that is with X_1+X_2=0 i.e. X_1=-X_2 for modified gear pair and the sum is not equal to zero, i.e. X_1+$X_2 \neq 0$. One of the cylindrical gears in this case may, however, have an addendum modification Zero.

The gear ratio is define as the ratio of the number of teeth of the gear Z_2 to the number of teeth of the pinion Z_1, thus u = Z_2/Z_1.
Working pitch circle with diameter,

d_w = 2.r_w

Are those transverse intersection circles of cylindrical gear pair, which have the same circumferential speed at their mutual contact point. The working pitch circles divide center distance a=rw1+rw2 in the ratio of the tooth numbers, thus

d_{w1}=2.a/(u+1) and

d_{w2}=2.a.u/ (u+1)

In the case of both an unmodified gear pair and a gear pair at reference center distance, the center distance is equal to the zero center distance,

a_d=(d_1+d_2)/2

And the pitch circle is simultaneously the reference circle, i.e. d_w=d. However, in the case of a modified gear pair and the center distance is not equal to the zero center distance. The pitch circles are not simultaneously the reference circles.

If in the case of modified gear pairs the bottom clearance C_p corresponding to the standards basic rack tooth profile is to be retained, then an addendum modification is to be carried out.

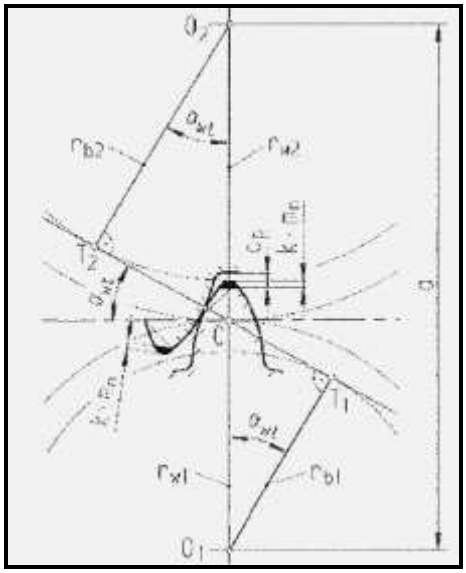

Figure 4 - Transverse section of an external gear pair with contacting left handed flanks

For unmodified gear pairs and gear pair at reference center distance, k=0. In the case of external gear pairs k<0, i.e. the tip diameter of both gears become smaller. In the case of in a cylindrical gear pair either the left or the right flanks of the teeth contacts each other on the line of action each lying symmetrical in a relation to the center line through O_1O_2. The line of action with contacting lefts flanks is the tangent to the two base circles at point T_1&T_2. The common tangent on the pitch circles it includes the working pressure angle α_{wt}.

The working pressure angle α_{wt} is the transverse pressure angle at a point belonging to the working pitch circle. It is determined by

cosα_{wt} = db_1/d_{w1} = db_2/d_{w2}

In the case of unmodified gear pairs and gear pairs at reference center distance. The working pressure angle is equal to transverse pressure angle on the reference circle i.e. α_{wt} = α (9).

GEAR BACKLASH

Backlash – A purposeful error. Properly functioning mechanical systems need to have a certain 'clearance' between the components transmitting under load. Clearance is necessary to avoid interference, wear, and excessive heat generation, ensure proper lubrication, and compensate for manufacturing tolerances etc.

clearance in the gear mesh means that the gap between the teeth of one gear is small amount larger than the tooth width of the mating gear.

The gear rattle motion between the gears due to backlash clearance, resulting from meshing and un meshing of gear teeth results in creation of an impact noise referred to as a gear hammering.

It is clearly observed that lower the backlash lower will be the gear rattle, but on the contrary as the backlash increase drastically the gear rattle decreases as well. This is because there is large gap between the teeth and hence the gear teeth will not rattle between the two teeth, but there will be large impacts on the driving side of the gear teeth. It is observed that at an optimum level of backlash the gear rattle peaks drastically and it is not favorable for the gear train from noise and longevity point of view.

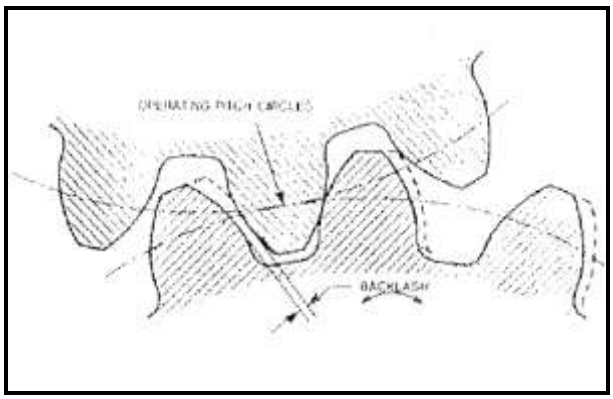

Figure 6: Gear backlash

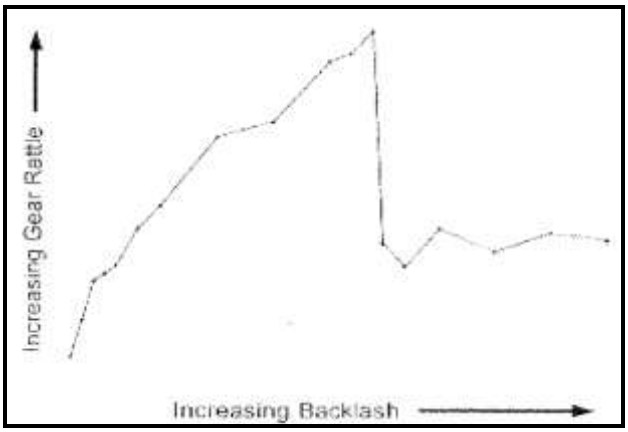

Figure 7: Variation of gear rattle with increasing backlash

III. PROBLEM DEFINATION & CONCEPT

Bench Marking Data				
Company Name	Total No of teeth	Gear Width (b)	Pressure Angle (α)	Root Radius
MHEPL	145	12	14.5	1.1
KOIL	122	15.5	18	1.01
Eicher	142	15	22.98	1.165
Cummins	141	15.8	19.5	1.17175
Greaves Cotton	135	15.5	16.51	0.89

Table: Bench Marking of Gear Width and Pressure Angle

Engine	Gear Width	Pressure Angle	FOS		
			Beam strength Statics	Beam strength Dynamic	Pitting Stress
FOS Existing	12	14.5	1.309	1.231	1.30
FOS Proposal-1	16	14.5	1.74	1.59	1.5
FOS Proposal-2	12	20	1.60	1.52	1.3
FOS Proposal-3	16	20	2.14	1.95	1.5

Table 2 – FOS Proposal

MHEPL has already started work on failure came in past few year for providing high level quality to the customer. Initially ring gear failure was analysis on the basic of manufacturing process and metallurgy analysis, observed data got as per the SOP and material specification as per the given material list in the assembly drawing. Finallydecision has taken redesign analysis of Flywheel ring gear. On the basis of redesign calculation, this redesign calculation compared with same application product part.

Observed below table Factor of safety is less than 1.5 for exit gear width and pressure angle. On the basis of bench marking suggested three proposals to increase FOS.

III. MODELING OF FLYWHEEL RING GEAR

Spur gears are the most common type of gears used in industry and they have straight, flat-topped teeth set parallel to the shaft. With help of following input data we can easily create 3 D model by using any CAD software. Now this difficult task to design create accurate 3D model. Using UG CAD software created 3D model. While design 3D model following steps followed

- calculate the geometric parameters of the gear,
- draw an accurate image of the gear with a detail of the teeth,
- adjust the tooth thickness by use of correction
- calculate the tooth root bending stress with the Lewis method,

To create the gear need the following parameters in the input form:

- the number of teeth z,
- the module m,
- the pressure angle α,
- the rack shift coefficient x/m for the adjustment of the tooth thickness,
- the coefficient of fillet radius of the rack cutter R/m,
- the static nominal torque C applied,
- the face width b of the tooth.

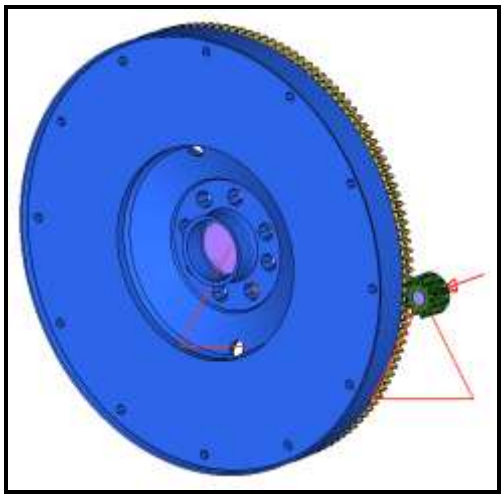

Figure 8: Design Model

IV. RESULTS & DISCUSSIONS

In the present project is analyzed the problem of three-dimensional behavior of a spur gear system with sever load condition. In this project we are going to study the effects of teeth load, axial load, and velocity due to design errors. We will study if the failure results from severe operating conditions, like overload combined with teeth misalignment. To model spur gears using SABR, a three-dimensional model is adopted. Tooth load distributions are both highly sensitive to gear body deflections in torsion and bending but also to the manufacturing errors.
The mechanical systems which include gear sets are various and their operating conditions are subject to manufacturing problems, assembly and installation problems. So, there is very important to understand in the functioning conditions the static and Quasi-static Analysis behaviour of mechanical systems with gear.

Quasi-static Analysis: This is analysis data of flywheel ring gear & pinion. As per Lewis calculation we are getting very low bending safety factor. When it is less than 1.2 starter ring has been fail for bending damage % 188.67 as shown in below table. Then we are taking.

Input Data				
Data	**Pinion**	**Gear**	**units**	**Condition**
Torque (T)	40	539.51	N-m	cracking torque
Speed (N)	1050	80	rpm	475
Gear Width (b)	12	12	mm	condition1. (m8<b<m12) & 3 to 5 times of pitch circle
Normal Module (m)	3	3		Given In diagram
Normal pressure angle (α)	14.5	14.5	degree	Form Factor will be change
Helix angle (β)	0	0	degree	Given In diagram
number of teeth (Z)	11	145		Given In diagram

Pitch Circle Diameter (d')	33	435	mm	Given In diagram
Addendum circle diameter	41	441	mm	Addendum circle diameter=m*(Z+2)
Deddendum circle diameter	28	428	mm	Deddendum circle diameter=m*(Z-2.5)
Base circle diameter	32	421	mm	Base Circle Diameter= d'cos(α)
Diameteral pitch (P)	0.33	0.33		P=(Z/d')
Addendum (ha)	3	3	mm	ha=1m
deddendum (hd)	3.47	3.47	mm	hd=1.25m
module (m)	3	3		m=(1/P)
speed ratio (i)	13	13		i= Zg/Zp
Power transmitted	4.5	4.5	KW	Given In diagram

Table: Input Data

No	Result	Baseline		Iteration 1		Iteration 2	
		Ring Gear	Pinion	Ring Gear	Pinion	Ring Gear	Pinion
1	Required Life (hours)	81.699		81.699		81.699	
2	Required Life (million cycles)	1.0	13.181	1	13.181	1	13.181
3	ISO Total Bending Damage (%)	188.67	0	0	0	0	0
4	ISO 6336-6 Bending Safety Factor	0.982	3.14	1.228	2.812	1.216	3.106
5	ISO Total Contact Damage -Flank P (%)	27.947	115.991	21.766	497.2	25.52	192.652
6	ISO Total Contact Damage -Flank N (%)	0	0	0	0	0	0
7	Min. ISO Contact Safety -Flank P	1.101	0.989	1.122	0.886	1.109	0.952
8	Min. ISO Contact Safety -Flank N	10000	10000	10000	10000	10000	10000
9	ISO Bending Life (Hours)	43.304	Infinite	Infinite	Infinite	Infinite	Infinite
10	ISO Bending Life (million cycles)	0.53	..	..	..	..	..
11	ISO Contact Life (Hours)	292.332	70.436	375.355	16.432	320.14	42.408
12	ISO Contact Life (million cycles)	3.58	11.36	4.59	2.65	3.92	6.84
13	Tangential Tooth Load (N)	2096.9		2096.9		2096.9	
14	Axial Load (N)	0		0		0	

15	Tangential Velocity (m/s)	4.7	4.7	4.7

Table: Quasi-static Analysis (SABR) ISO 6336 Method

Iteration 1 & Iteration 2 in this case increase bending factor safety more than 1.2 so bending damage % is o. But contact safety flank-P is fail to all. Quasi-static analysis is showing need to modify design as iteration 1 & iteration 2.

static Analysis: This data is showing for current design. Even for the reduced cranking torque, static analysis of the ring gear pair show failure as the stresses are more than the material yield limit. The stresses are higher in the initial stage itself i.e. the interference fit application. Refer below table for more details.

No.	Applied Load	Calculated Load	Displacement	Stress	Yield Strength (Mpa)
1	Interference Fit -0.295 mm		0.28	373	340
2	Pinion Torque = 15 Nm	Ring Gear Torque = 200 Nm	0.28	459	
3	Pinion Torque = 19 Nm	Ring Gear Torque = 250 Nm	0.28	482	
4	Pinion Torque = 23 Nm	Ring Gear Torque = 300 Nm	0.28	506	
5	Pinion Torque = 27 Nm	Ring Gear Torque = 350 Nm	0.28	530	
6	Pinion Torque = 30 Nm	Ring Gear Torque = 400 Nm	0.28	548	
7	Pinion Torque = 35 Nm	Ring Gear Torque = 460 Nm	0.28	593	

Table: Static Analysis (Stress Comparison for Different Torque)

IV. CONCLUSIONS

1) Quasi –static iterations of the ring gear pairs how significant decrease in the damage on the gears for the given torque load.
2) Bending safety factor of the ring gear has increased from 0.982 to minimum of 1.2 in both iterations while the contact safety factor does not increase much.
3) Both iterations show considerable increase in the gear life as compared to the base line analysis.
4) Even for the reduced cranking torque, static analysis of the ring gear pairs how failure as the stresses are more than the material yield limit. The stresses are higher in the initial stage itself i.e. the interference fit application.
5) Design the gear as per SAE standard J543 for ring gear designs
6) Select a ring gear suitable to the starter and the pinion from the supplier, finding a best fit for this particular application.
7) Improvement in manufacturing or assembly processes, materials and heat treatment.

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