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Research Paper

FINITE ELEMENT ANALYSIS AND FATIGUE ANALYSIS OF SPUR GEAR UNDER RANDOM LOADING

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The gear fitted in the gearbox Armored tracked vehicle is vulnerable to considerable fatigue damage over its life period due to the dynamic excitations caused by the terrain undulations, the rotating wheel and track assemblies. For this purpose, initially static analysis of the model was carried out to validate the model and the boundary conditions correctness. Further Modal analysis is carried out to determine the dynamic characteristics of the gear model. The random load time history is transformed in to frequency domain using Fast Fourier transform to obtain load Power Spectral Density (PSD). Then the stress PSD response is obtained at critical node from the random vibration analysis. Once the spectrum of stress variation is obtained given input to the fatigue analysis and fatigue life is determined by FE package ANSYS 11.0.

Keywords: Spur gear, Static analysis, Modal analysis, PSD analysis

INTRODUCTION

Gears are the most common means of transmitting power in the modern mechanical engineering world. They vary from tiny size used in the watches to the large gears used in marine speed reducers; bridge lifting mechanism and railroad turn table drivers. They form vital elements of main and ancillary mechanism in many machines such as automobiles, tractors, metal cutting machine tools, rolling mills, hoisting and transmitting machinery and marine engines, etc.

The four major failure modes in gear systems are tooth bending fatigue, contact fatigue, surface wear and scoring. Two kinds of teeth damage can occur on gears under repeated loading due to fatigue; namely the pitting of gear teeth flanks and tooth breakage in the tooth root. Tooth breakage is clearly the worst damage case, since the gear could have seriously hampered operating condition or even be destroyed. Because of this, the stress in the tooth should always be carefully studied in all practical gear application. The fatigue

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process leading to tooth breakage is divided into crack initiation and crack propagation period. However, the crack initiation period generally account for the most of service life, especially in high cycle fatigue.

The initial crack can be formed due to various reasons. The most common reasons are short-term overload, material defects, defects due to mechanical or thermal treatment and material fatigue. The initial crack then propagates under impulsive loading until some critical length is reached, when a complete tooth breakage occurs. The service life of a gear with a crack in the tooth root can be determined experimentally or numerically (e.g., with finite element method). The fatigue life of components subjected to sinusoidal loading can be estimated by using cumulative damage theories. Their extension to random load fatigue, through straightforward, may not be very accurate owing to inherent scatter exhibition by the fatigue phenomena. Due to the complexity in geometry and loading on the structure, the finite element method is preferably adopted.

Major contribution being the terrain undulations, the rotating wheel, the track assemblies, the drive line and the engine and also influenced by the incidental variations in the vehicle components, firing ammunition from the vehicle body and hitting of ammunition on the vehicle body.

PROBLEM STATEMENT

The present work deals with the calculation of static and dynamic analysis, and fatigue life estimation of test gear, which is contacting master gear and assuming loading on the gear is random or constant amplitude by Finite Element package ANSYS

AIMS AND OBJECTIVES

1. To carry out the detail finite element analysis like static analysis to validate the model and boundary conditions correctness, Modal analysis to determine dynamic characteristics of the model and transient dynamic analysis to determine dynamic load factor.
2. Simulation of loads of deterministic and non-deterministic nature causing fluctuations bending stresses in gear.
3. To carry out dynamic analysis of gear both under random loading and constant amplitude loading conditions and obtain the stress PSD response at vulnerable locations.
4. To carry out fatigue analysis in order to determine the fatigue life of the test gear with help of above stress PSD responses at vulnerable locations.

3D Modeling of Spur Gears

Modeling of spur gear using NX-cad software

Figure 1: Test Gear

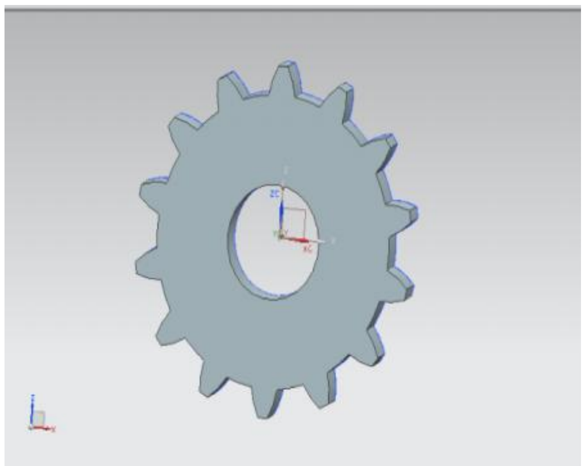
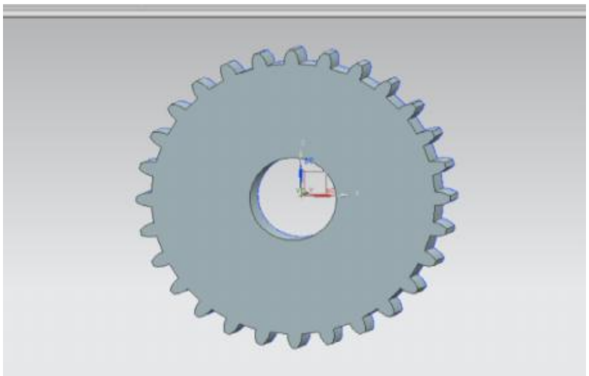
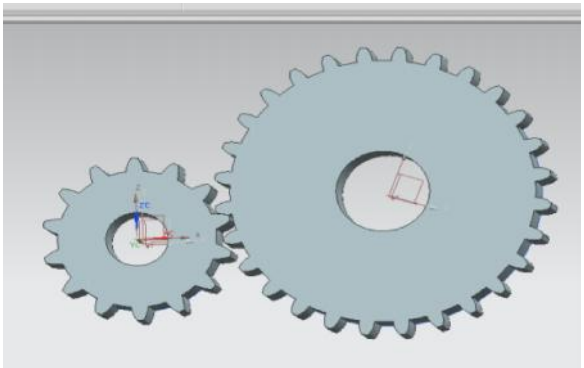


Figure 2: Master Gear



Assembly of Spur Gears

Figure 3: Model of Spur Gears in Unigraphics N x 7.5



FINITE ELEMENT ANALYSIS

In FEM, complex structure or continuum is divided into finite number of small regions called as elements. The material properties and governing equations are considered over these elements and the field quantity is expresses as unknown values at corners of these elements called as the nodes. Due to this the infinite degree of freedom is converted into finite degree of freedom problem. An assembly process, duly considering the loading and constraints, results in a set of equations. Solution of these equations gives us approximate behavior of the continuum.

ASSUMPTIONS

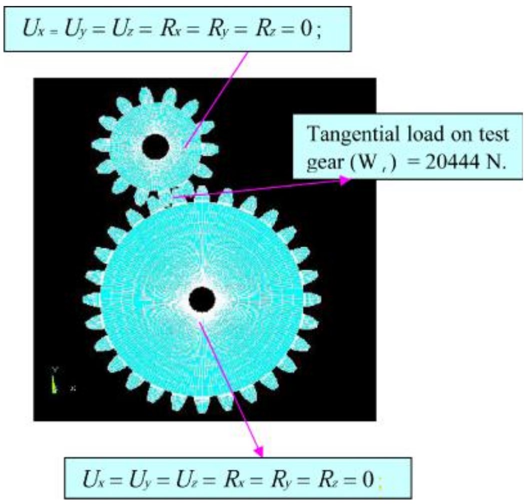
The following conditions have been assumed.

1. The material has been assumed to behave as linear elastic material.
2. The material is homogeneous and isotropic.
3. Heat generation and thermal stress are ignored.
4. The gear to mesh under conditions of good lubrication and the friction is taken as zero.
5. No contact behavior is considered.
6. There are no residual stresses present in the gears.

Table 1: Finite Element Model Properties

Element Type	Shell
Total number of node at each element	8
Number of degrees of freedom per node	6
Total number of elements	2265
Total number of nodes	7648
Largest node number	7648
Total number of degrees of freedom in model	43735

Figure 4: Boundary Condition Model



Static Analysis

A static load is a stationary force or moment acting on a member. To be stationary the force or moment must have an unchanging magnitude, unchanging point or points of application, and an unchanging direction.

Table 2: Composition of Alloys En-36 and En-19

Constituent Material	Percentage Content	
	Master Gear En-36 Steel	Test Gear En-19 Steel
Carbon	0.12	0.43
Silicon	0.22	0.19
Manganese	0.51	0.73
Nickel	3.42	–
Chromium	0.94	1.30
Molybdenum	0.16	0.27

Table 3: Mechanical Properties

Properties	Master Gear	Test Gear
Material name	En-36c	En-19
Young's modulus of elasticity (MPa)	2E 5	2E 5
Poisson's ratio	0.29	0.29
Yield strength (MPa)	1230	1030

Table 4: Cyclic Properties

Properties	Master Gear	Test Gear
Fatigue strength coefficient (SF)	2415.92 (N/mm^2)	1944.32 (N/mm^2)
Fatigue strength exponent (B)	–0.07	–0.25
Fatigue ductility coefficient (EF)	0.002	1.22
Fatigue ductility exponent (C)	–0.47	–0.73
Cyclic strength coefficient (KP)	4263.71 (N/mm^2)	1862.96 (N/mm^2)
Cyclic strain hardening exponent (NP)	0.1	0.14

Table 5: Static Analysis Results

Type of Stress	Maximum Stress (N/mm)	Corresponding Node Number
S_{xx}	540.1	7311
S_{yy}	267.8	7315
$\pm_1$	756.3	7315
$\pm_2$	89.64	7261
Von misses stress	792.4	7311

Figure 5: (Bending) Stress Contour

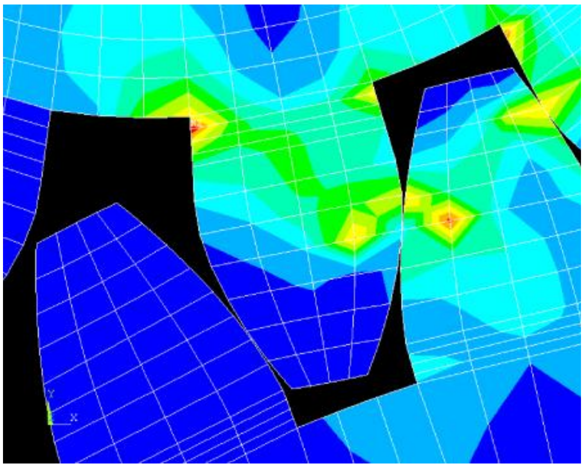


Table 6: Comparison of Analytical and Ansys Result

Type of Stress	Analytical	Ansys Software	% Error
S_{xx}(N/mm^2)	532.3	540.1	1.5

We can observe that the values for maximum bending stress obtained by Ansys are very closer to analytical value and the error is 1.5%, which confirms that the model and boundary conditions applied to the gear, is correct. The stresses induced in the adjacent tooth are about 1.13% of loaded tooth.

DYNAMIC ANALYSIS

Dynamic analysis involves the computation of response of a linear system subjected to time dependent loads.

Table 7: Frequency Corresponding to Mode Shape

Mode Shape	Frequency (rad/sec)	Frequency (cycles/sec)
1	1.753E+03	2.78 E+02
2	4.322E+03	6.87 E+02
3	4.599E+03	7.31E+02
4	5.474E+03	8.71E+02
5	6.557E+03	1.043E+03
6	6.729E+03	1.071E+03
7	1.1928E+04	1.898E+03
8	1.3300E+04	2.116E+03
9	1.3709E+04	2.181E+03
10	1.6830E+04	2.678E+03
11	1.7022E+04	2.092E+03
12	1.9077E+04	3.036E+03
13	1.9135E+04	3.045E+03
14	1.9718E+04	3.138E+03
15	2.0082E+04	3.196E+03
16	2.0707E+04	3.295E+03
17	2.1285E+04	3.387E+03
18	2.1382E+04	3.403E+03
19	2.1508E+04	3.423E+03
20	2.230E+04	3.549E+03

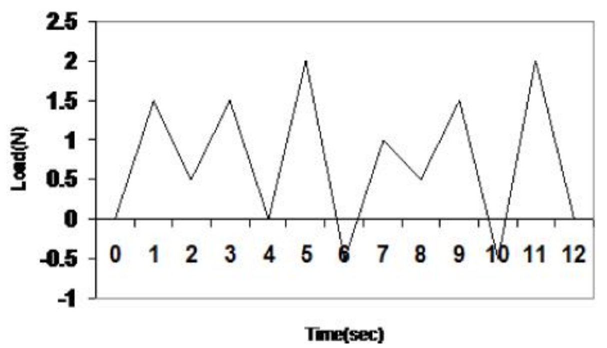
TRANSIENT DYNAMIC ANALYSIS

Transient dynamic analysis may be used to determine the response of structures subjected to arbitrary time-varying loads.

Loading Detail

Geometric model, material properties and the boundary conditions are same as the static analysis except that the contact force is variable in nature but peak load is same as in static analysis. And the time step taken as 0.001 sec.

Figure 6: Graph of Time and Load



Results of Transient Dynamic Analysis

1. The displacement of node which has maximum deformation corresponding to first mode shape (Node number 363) is shown in Figure 7.

2. The stress histories of critical node which has maximum bending stress in case of static analysis (Node number 7311) is shown in Figure 8.

3. Transient dynamic analysis is predominant on static analysis because at same peak load condition the stress is higher as compared with static analysis and the dynamic load factor is 1.2. It is due to mass and acceleration effects induced in dynamic loading.

Figure 7: U_x (Displacement) Response History Plot for Node 363

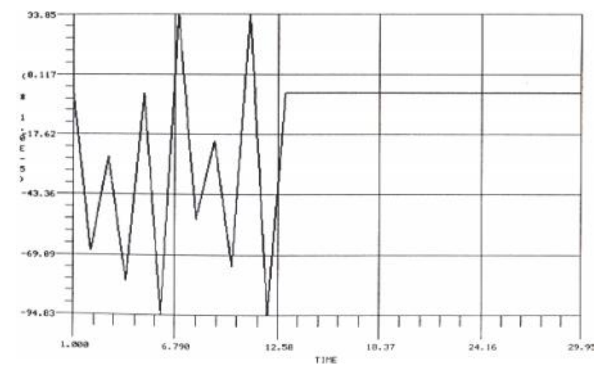
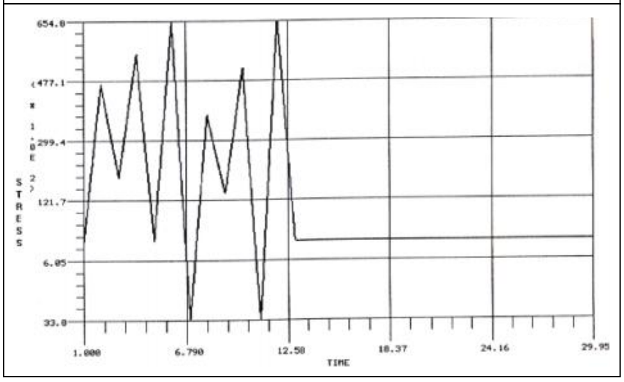


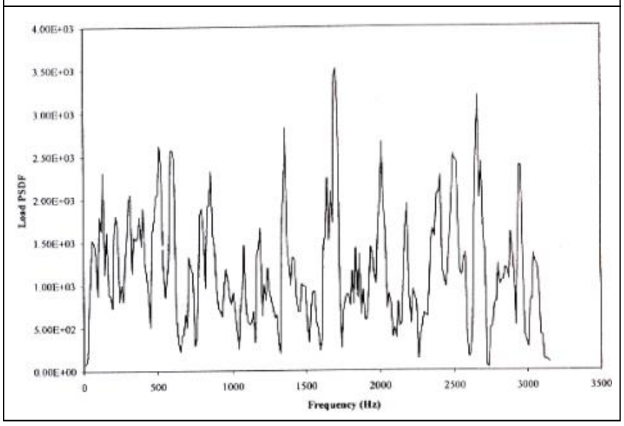
Figure 8: S_{xx} (Bending) Stress Response History Plot for Node 7311



RANDOM VIBRATION ANALYSIS

The load history is converted to load PSD with the help of fast Fourier transform. This load PSD is used in ANSYS package for random vibration analyses. The load PSD is shown in the Figure 9.

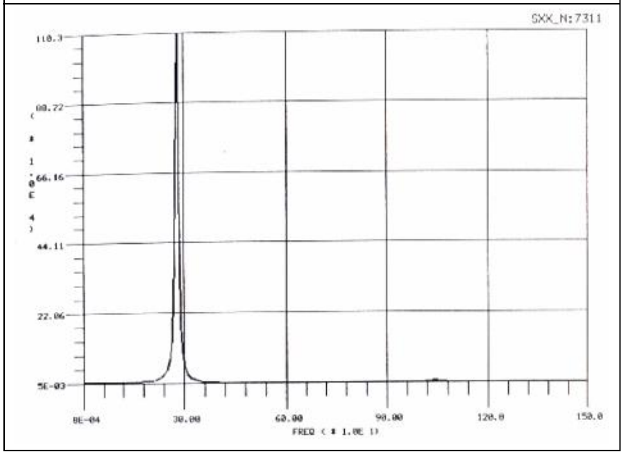
Figure 9: Load PSD



Results of Random Vibration Analysis

The output of random analysis is stress PSD response of node no. 7311 which is shown in Figure 10. As we see that the maximum power content is at the frequency 278.99 Hz corresponding to first mode shape and the power content at other modes are negligible as compared to the first mode because they

Figure 10: Stress PSD Response of Node No. 7311

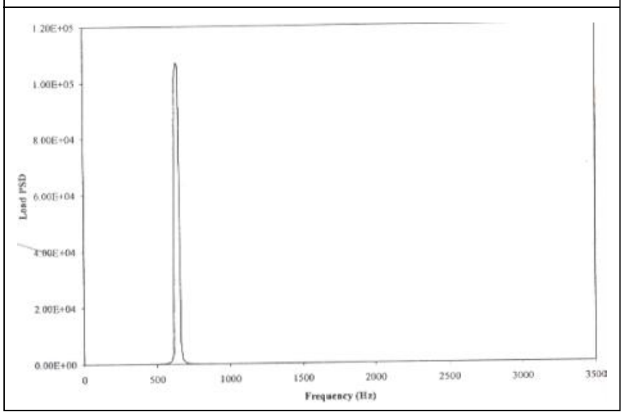


are very small. This means that careful analysis is need in the region of first mode frequency rather than at other modes.

CONSTANT AMPLITUDE LOAD ANALYSIS

Deterministic data are those that can be described by an explicit mathematical relationship. Data representing a physical phenomenon can be categorized has being either periodic or non periodic. Periodic data can be further categorized as being both sinusoidal and complex periodic. Non-periodic data can be further categorized as being both almost periodic and transient.

Figure 11: Load PSD



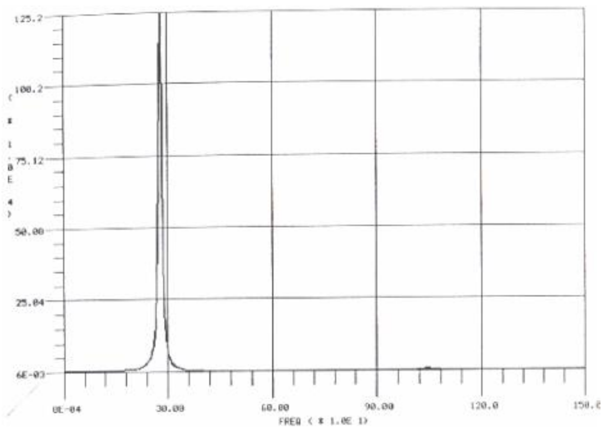
The load history is converted to load PSD with the help of fast Fourier transform. This load PSD is used in ANSYS package for random vibration analyses. The load PSD is shown in the Figure 11.

Results of Constant Amplitude Load Analysis

The output of constant amplitude analysis is stress PSD response of node no. 7311 which is shown in Figure. As we see that the maximum power content is at the frequency 278.99 Hz corresponding to first mode shape and there is very little power content at other modes. This means that careful analysis is need in the region of first mode frequency rather than at other modes.

And for constant amplitude loading, it is observed that energy content is more as compared to random loading. So the stresses induced due to constant amplitude loading are considerably higher than that random loading.

Figure 12: S_{xx} (Bending) Stress PSD at Node 7311



FATIGUE ANALYSIS

Fatigue life evaluation has almost become mandatory for structural components under repeated loading in aerospace, nuclear and other major industries where the structural failures are mainly due to fatigue.

Table 8: Table Cyclic Properties	
Properties	**Test Gear**
Fatigue strength coefficient (SF)	1944.32 N/mm^2
Fatigue strength exponent (B)	–0.25
Cyclic strength coefficient (KP)	1862.96 N/mm^2
Cyclic strain hardening exponent (NP)	0.14

Fatigue Properties

The properties used in crack initiation analysis.

Analysis

There are two types of analysis done in this project.

1. Fatigue analysis for random loading
2. Fatigue analysis for constant amplitude loading

Fatigue Analysis for Random Loading

Taking the same stress PSD as input in Ansys for crack initiation analysis. Geometric model, boundary conditions and material properties are same.

Table 9: Input to Fatigue Analysis	
Type of Analysis	**Crack Initiation Analysis**
Type of correlation	Stress-life (S-N)
Basic FE analysis	Random vibration
Material properties	Table cycle properties

Results of Fatigue Analysis for Random Loading

Due to the stress PSD response given input at critical node 7311, life obtained by FEM is 5.8 x 10^5 cycles where as life obtained by experiment is 6.5 x 10^5 cycles (Hanumanna *et al.*, 2001). And the variation is less. This leads to the validation of the results. Life obtained by FEM is less as compared to the experimental life, this is due to clamped condition of test gear.

Table 10: Fatigue Life Under Random Loading

Life Obtained by Experiment (cycles)	Life Obtained by FEM (cycles)
6.5×10^5	5.8×10^5

Fatigue Analysis for Constant Amplitude Loading

Constant amplitude loading pattern has been shown in Figure 11. Its stress PSD response at critical node 7311 has been obtained in constant amplitude load analysis, which is shown in Figure 10. Taking the same stress PSD as input in Ansys for crack initiation analysis. Geometric model, boundary condition and material properties are same.

Table 11: Input to Fatigue Analysis

Type of Analysis	Crack Initiation Analysis
Type of correlation	Stress-life (S-N)
Basic FE Analysis	Random Vibration
Material properties	Tablecycle properties
Approach	Miner-Palmgren Damage Rule

Results of Fatigue Analysis for Constant Amplitude Loading

Due to stress PSD response given input at critical node 7311, life obtained by FEM is 2.9 x 10 cycles where as life obtained by experiment is 3.25 x 10 cycles (Hanumanna *et al.*, 2001). And the variation is less. This leads to the validation of the results. Life obtained by FEM is less as compared to the experimental life, this is due to the clamped condition of test gear. And the life obtained by constant amplitude loading is much lesser than that obtained by random loading, this is due to stresses induced in constant amplitude loading is higher for same peak load condition.

Table 12: Fatigue Life Under Constant Amplitude Loading

Life Obtained by Experiment (cycles)	Life Obtained by FEM (cycles)
3.25×10^5	2.9×10^5

Table 13: Static Analysis Results

Type of Stress	Maximum Stress (N/mm)	Corresponding Node Number
S_{xx}	540.1	7311
S_{yy}	267.8	7315
$\pm_1$	756.3	7315
$\pm_2$	89.64	7261
Von misses stress	792.4	7311

Table 14: Comparison of Analytical and Ansys Result

Type of Stress	Analytical	Ansys Software	% Error
S_{xx}(N/mm^2)	532.3	540.1	1.5

CONCLUSION

The gear in a gear box fitted an armoured tracked vehicle for the purpose of power transmission and positioned of heavy mass to the desired angle with high accuracy are subjected to fluctuating loads that are random in nature. Therefore, it is analyzed for random loading and also under constant amplitude loading conditions for fatigue analysis and life of gear has been obtained by finite element package ANSYS.

Following are the conclusions:

1. The stresses obtained for static analysis were presented in Table 13.
2. Comparison of analytical and Ansys result was presented in Table 14.

3. The maximum power content is at the frequency 278.99 Hz corresponding to first mode shape and the power content at other modes are negligible as compared to the first mode because they are very small.
4. Due to the stress PSD response given input at critical node 7311, life obtained by FEM is 5.8 x 10 cycles where as life obtained by experiment is 6.5 x 10 cycles.
5. Due to stress PSD response given input at critical node 7311, life obtained by FEM is 2.9 x 10 cycles where as life obtained by experiment is 3.25 x 10 cycles.

SCOPE FOR FUTURE WORK

1. 3-D contact analysis of gears including heat generation and frictional component of load may be investigated.
2. Surface wear is one of the major failure mode in gear systems, therefore surface wear prediction methodology of gear pair may be carried out.
3. Present study may be extended to different complex cases of multi axial fatigue loading of proportional and non-proportional nature.
4. Carryout fatigue reliability assessment of test gear under random loading both by S-N curve approach and Fracture mechanics approach.
5. Fatigue and reliability analysis under non-stationary random loading may be carried out.

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