Contact Stress Analysis on Spur Gear

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Abstract— Spur gears are most commonly used transmission elements in engineering applications. Two types of fatigue failure occur in spur gears i.e., bending fatigue and contact fatigue. Bending fatigue leads to breakage of gear teeth, contact fatigue is a surface fatigue failure like pitting, scoring etc., The structural response of gear to external load depends upon its geometrical features, which in turn depends upon module. In the present work, influence of module and Pressure angle on geometric features has been studied considering identical operating parameters. Based on geometrical features, structural response of gears of different modules will be analyzed in terms of tooth stresses, fatigue life

Keywords: spur, contact, Stress, tooth

I. INTRODUCTION

A. Gear:

Gear is toothed wheel that connects with another toothed instrument so as to alter the velocity or course of transmitted movement.

Fig. 1: Spur gear

An apparatus is a segment inside a transmission gadget that transmits rotational power to another rigging or gadget. A rigging is not quite the same as a pulley in that an apparatus is a round wheel which has linkages ("teeth" or "machine gear-pieces") that cross section with other rigging teeth, permitting power to be completely exchanged without slippage. Contingent upon their development and course of action, equipped gadgets can transmit strengths at various velocities, torques, or in an alternate heading, from the force source. The most widely recognized circumstance is for a rigging to work with another apparatus.

1) Spur gear:

Spur gears are the most widely recognized sort of riggings appeared in Fig. 1.1. They are utilized to transmit revolving movement between parallel shafts i.e. they are generally round & hollow fit as a fiddle, and the teeth are straight and parallel to the hub of turn. Some of the time numerous goad apparatuses are utilized without a moment's delay to make vast rigging decreases. Outfitting is a standout amongst the most basic parts in a mechanical force transmission framework, & in most modern pivoting apparatus.

B. Gear tooth failure:

The failure of gear takes place due to the following reasons:

1) Wear
2) Overloads
3) Fatigue

1) Fatigue Failure in Gears:

There are three types of fatigue failure in gears.

1) Bending fatigue
2) Contact fatigue
3) Fatigue wear

C. Factors influencing contact fatigue:

There are few factors are influencing on contact fatigue

- Load distribution on gear tooth
- Contact stress
- Surface finish and lubrication

II. LITERATURE SURVEY

H. E. Staph [1] built up a PC system to plan outside goad gears having typical contact proportions (<2) and high contact proportions (≥ 2). Impacts of changes in rigging parameters on a few execution variables of high contact proportion apparatuses were considered. At that point results were contrasted and those for the proportional typical contact proportion gears. The high contact proportion gear acquired by expanding the addendum of an identical typical contact proportion gear have lower twisting and compressive anxieties (good) and expanded grinding heat era and glimmer temperatures (unfavorable).

A. K. Elkohly [2] gave answer for the counts of burden sharing b/w teeth in cross section for high contact proportion gears. In this examination the total of tooth avoidance, dividing blunder & profile alteration was thought to be equivalent for all sets in contact. Likewise, the aggregate of typical burdens taken by sets was thought to be equivalent to the most extreme ordinary burden. Solidness variety along way of contact was considered. Tooth filet stress, contact anxiety were resolved utilizing tooth geometry after individual burden were figured. The outcomes got from trial examination were contrasted and explanatory results.

III. PROBLEM DESCRIPTION

A. Problem definition and objectives:

Gears grow high burdens at contact district of pair of teeth, when it is subjected to outer burdens. Because of these high anxieties at contact district, there is a higher possibility of exhaustion disappointment at these areas. The rehashed stress that happens on the contact locale of goad apparatuses pair tooth surface is for all intents and purposes observed to be the central variable in weariness disappointment of the rigging tooth. The primary goal of the present is to fulfill the impact of configuration parameter like module and weight edge on contact anxiety of goad apparatuses the indistinguishable working parameters of force, rate, gear proportion, face width and materials.

The goal incorporates:

- Determination of stress at the contact region by considering moving load and contact ratio.
- Study of influence of module on contact stresses.
- Study of influence of pressure angle on contact stresses.
- Study of influence of module and pressure angle on fatigue life of gear teeth.
- Study the effect of sliding velocity on power loss.

IV. METHODOLOGY

gear module has been shifted from 3mm to 10mm in ventures of eight and three distinctive weight points of 14.5˚, 20˚ and 25˚ have been chosen in the present work to examine their impact on contact push and weakness life. Contact stress and fatigue life.

Sl. No.	Parameter	Pinion	Gear
1	Face width (L)	60mm	60mm
2	Centre distance (a)	325mm	325mm
3	Power (P)	120KW	120KW
4	Speed (N)	650 RPM	406.25RPM

Table 1: Gear specification

Material properties	Pinion	Gear
Materials	Cast Steel 8630	Cast Iron
Modulus of elasticity	206 GPa	166 GPa
Poisson's ratio	0.3	0.3
Density	7850 kg/m^3	7250 kg/m^3
Ultimate stress (σ_{ut})	1141 MPa	240 MPa
Fatigue strength coefficient ($\sigma_f{}'$)	1936 MPa	920 MPa
Fatigue strength exponent (b)	-0.121	-0.106
Fatigue ductility co-efficient ($\epsilon_{f'}$)	0.420	0.213
Fatigue ductility exponent (c)	-0.693	-0.43
Cyclic strength co-efficient (K')	1502 MPa	712 MPa
Cyclic strain hardening (n')	0.122	0.102

Table 2: Material properties

In the present work, contact anxiety is resolved at the distinctive purposes of contact along the way of contact. The way of contact is isolated into equivalent no. of divisions. The ordinary burden acting at the each of these contact focuses is resolved considering the contact proportion. The span of ebb and flow for every purpose of contact is additionally decided for mating teeth. Contact stress along way of contact then decided in view of these heaps and span of ebb and flow. Fig. 3.1(a) demonstrates the line of contact AI for the riggings in lattice which has been isolated into no. of divisions and Fig. 3.1(b) demonstrates the comparing purposes of contact.

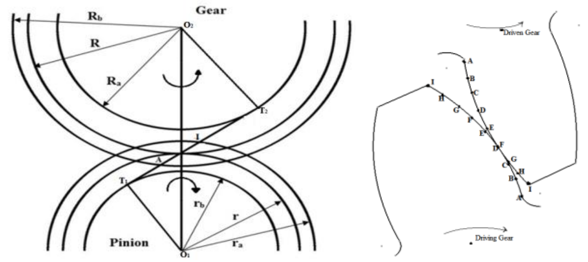

Fig. 2: (a) Line of contact (b) Points of contact

Contact ratio for the selected gear pair is determined by calculating length of path of contact and base pitch.

$$\text{Contact Ratio} = \frac{\text{Length of path of contact}}{\text{Base pitch}}$$

A. Fatigue life:

The fatigue life of the component is estimated by using Ramberg-Osgood equation and the Coffin-Mansion equation.

The strain amplitude is determined by using Ramberg-Osgood equation.

$$\epsilon_a = \frac{\sigma_a}{E} + \left[\frac{\sigma_a}{k'}\right]^{\frac{1}{n'}}$$

$$\text{Where, } \sigma_a = \frac{\sigma_{H\,max} - \sigma_{H\,min}}{2}$$

From Coffin-Mansion equation,

$$\epsilon_a = \frac{\sigma_{f'}}{E}(2N_f)^b + \epsilon_{f'}(2N_f)^c \quad .....(3.9)[21]$$

Where,

ϵ_a→Strain amplitude
E →Young's modulus in GPa
$\sigma_f{}'$ →Fatigue strength co-efficient in MPa
b → Fatigue strength exponent
$\epsilon_{f'}$→Fatigue ductility co-efficient
c → Fatigue ductility exponent

The calculation of contact ratio, radius of curvature, contact stress, fatigue life etc, for different modules are shown in annexure.

V. CONTACT STRESS ANALYSIS ON SPUR GEAR

A. Mesh specification:

Cross section is a critical stride to take care of numerical issues. This can discretise the given continuum into littler district called limited components. The first continuum is then considered as a collection of these components associated at a limited number of joints called hubs. The sort, size, number and course of action of the component rely on upon exactness of the arrangement required.

Fig. 3.6 demonstrates the hex-prevailing cross section utilized as a part of this present work. Fine work is utilized at the region of contact (area in the part where anxieties are basic) and coarse lattice at the rest of the regions. This blend of fine & coarse cross section gives precise results with ideal utilization of computational assets. After the lattice procedure, 140936 nodes and 27988 components were noted.

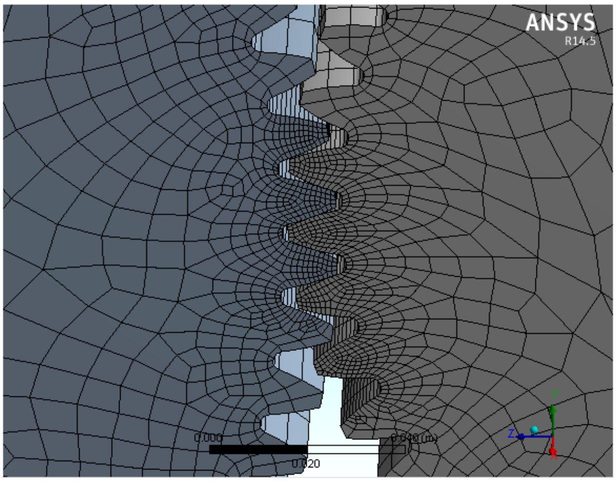

Fig. 3: 1Hex-dominant meshing

B. Boundary conditions:

Following boundary conditions are applied for the present analysis shown in Fig.3.7

- Frictionless support at the pinion centre, which that pinions, is free to rotate on its axis.
- Fixed support at the gear centre, which means generates resistance to the torque.
- Moment is applied on the pinion at centre, which means that gearbox is transmitting power.

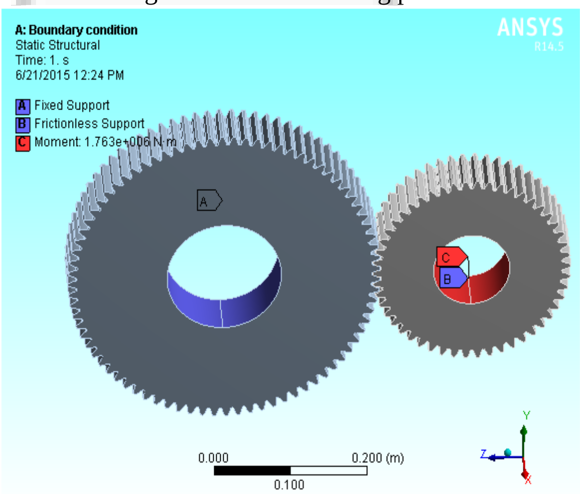

Fig. 4: Boundary conditions

VI. RESULTS AND DISCUSSION

The consequences of the hypothetical and numerical did to ponder the impact of module and weight edge on the configuration parts of the goad apparatuses are talked about in the accompanying segment.

A. Contact ratio:

Fig. 6.1 demonstrates the impact of module and weight point on contact proportion of the goad gears. The contact proportion is the normal number of teeth that are in contact when the apparatuses are coincided and rotated. With increment in module number of teeth will diminish, bringing about decreasing contact proportion. Additionally with increment in weight point, the length of arcof contact decreases, resulting in reduced contact ratio.

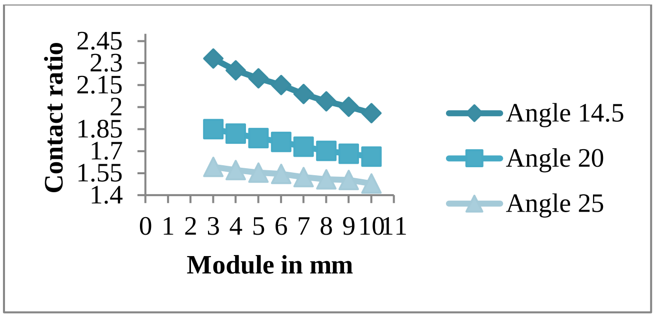

Fig. 5: Contact ratio V/s Module for different pressure angles

B. Radius of curvature:

Fig. 6.2 to 6.5 demonstrates the variety of range of ebb and flow of a reaching pinion and rigging teeth along the line of contact. A to E speaks to approach period and E to I speaks to break period. It can be seen from the Figs, that amid methodology period, span of shape lessening with increment in module for pinion though, it increments with increments in module for apparatus. Amid break period, the pattern is inverse to that saw amid methodology period.

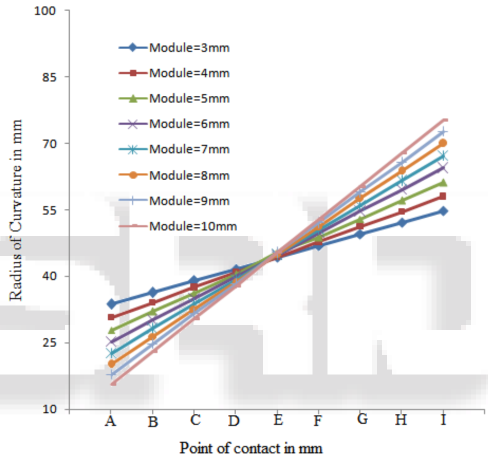

Fig. 6: Variation of radius of curvature of pinion tooth along path of contact [pressure angle 14.5˚]

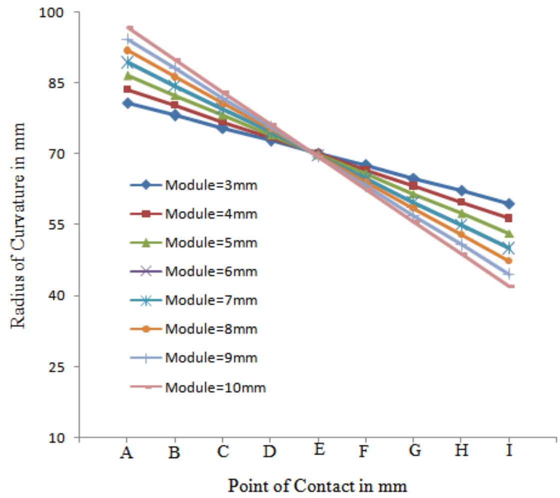

Fig. 7: Variation of radius of curvature of gear tooth along path of contact [pressure angle 14.5˚]

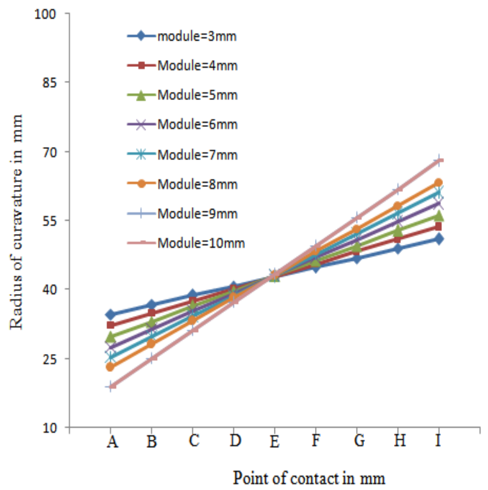

Fig. 8: Variation of radius of curvature of pinion tooth along path of contact [pressure angle 20˚]

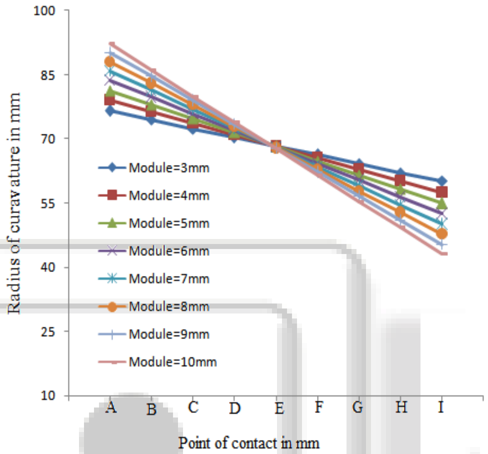

Fig. 9: Variation of radius of curvature of gear tooth along path of contact [pressure angle 20˚]

C. Contact stress:

These varieties of contact anxiety as for module and weight edge are because of the variety in tooth geometry and the contact proportion with module and weight edge.

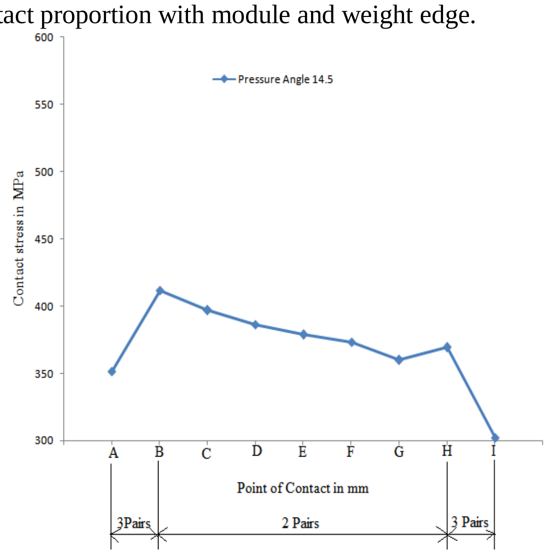

Fig. 10: Variation of contact stress along the path of contact [m =5mm, Ø=14.5˚]

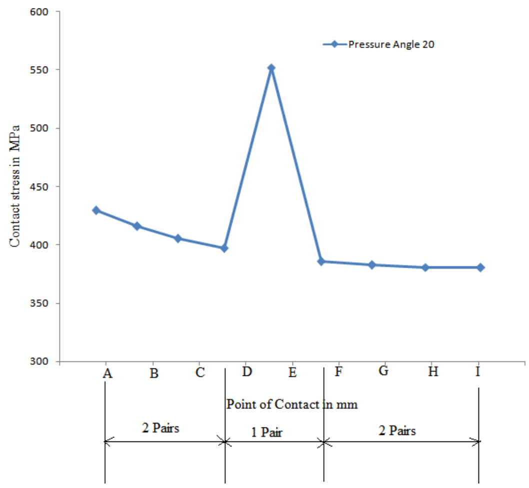

Fig. 11: Variation of contact stress along the path of contact [m =5mm, Ø=20˚]

D. Fatigue life:

There are two sorts of methodology for assessing of weakness life. One is the anxiety life methodology and other is strain life approach. In this present work strain life methodology is considered since the contact stress surpasses the yield anxiety of the apparatus material. The strain based methodology perceives the capacity of the material to experience confined plastic distortion without complete disappointment of the structure.

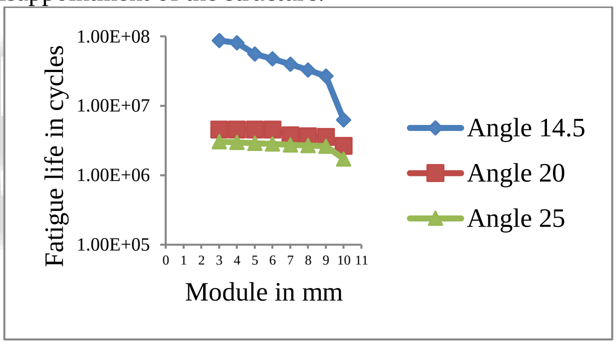

Fig. 11: Fatigue life V/s Module for different pressure angle

E. finite element analyses:

The contact stress and fatigue life of the spur gear pair was carried out using FEM. The von-mises stress distribution at the point of meshed gear pair can be observed, and these results are compared with theoretical results.

1) Contact stress:

Investigation has been done subsequent to giving limit conditions to the models (give settled backing for rigging focus, frictionless backing for pinion focus and Moment 1762950 N-mm for focus of pinion). It can be watched that a greatest Von-mises anxiety of 566.62 MPa is available at the pitch point for module 5mm at weight edge 20˚. The elastic anxieties are the fundamental driver of setting disappointment, on the off chance that they are sufficiently expansive. Fig. 4.14

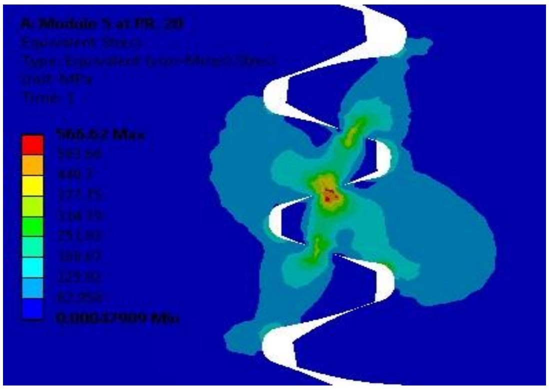

Fig. 12: Von-mises stress for module 5 at pressure angle 20˚

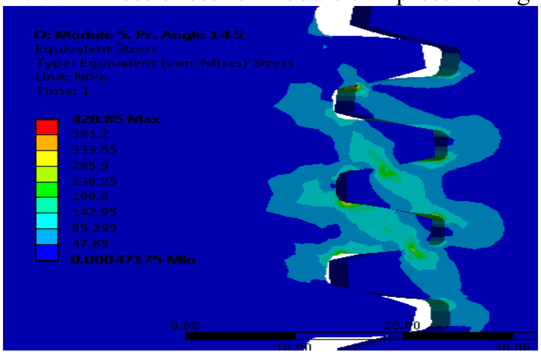

Fig. 13: Von-mises stress for module 5 at pressure angle 14.5˚

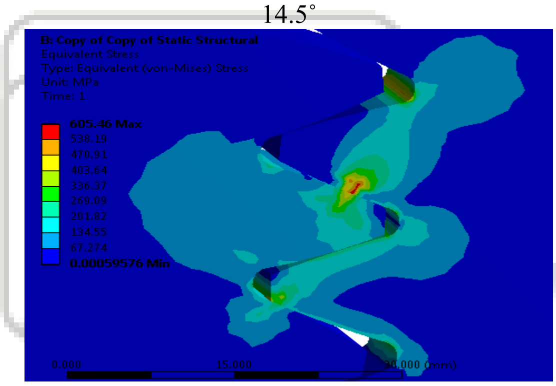

Fig. 14: Von-mises stress for module 5 at pressure angle 25˚

Investigation has been completed subsequent to giving limit conditions to the models (give settled backing for apparatus focus, frictionless backing for pinion focus and Moment 1762950 N-mm for focus of pinion). It can be watched that a most extreme Von-mises anxiety of 605.46 MPa is available at the pitch point for module 5mm at weight edge 25˚. The tractable hassles are the primary driver of setting disappointment, on the off chance that they are sufficiently substantial. Fig. 4.16 The correlations of results are appeared in

Table 4.1 for module 5 for various weight Angle.

Pressure angle	Maximum contact stress in MPa		% of variation
	Theoretical results	ANSYS results	
14.5˚	411.24	428.35	4.3%
20˚	552.15	566.62	2.5%
25˚	576.71	605.46	4.%

Table 3: Comparison of Max. Contact stress [Module=5mm]

2) *Fatigue life*

The Maximum Stress at the contact point, decided in the past segment is utilized for weariness life estimation of the apparatus utilizing strain life approach. Fig. 4.17 demonstrates the exhaustion life evaluated for an apparatus of 5mm module and 20˚ weight point and is shaped to be 9E05 cycles.

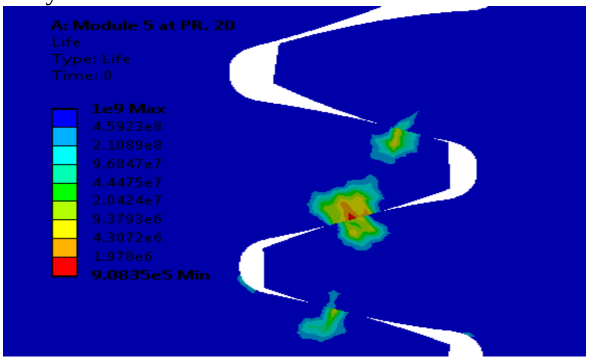

Fig. 15: Life of spur gear pair for module 5 at pressure angle 20˚

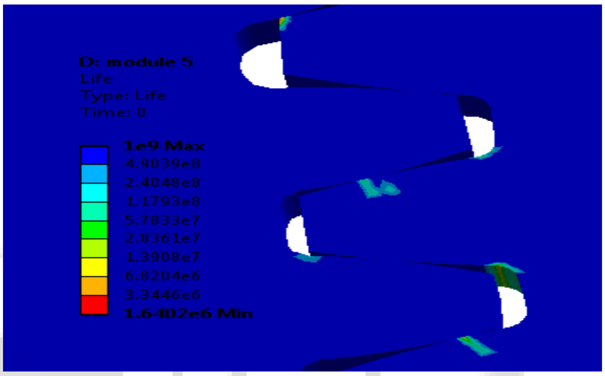

Fig. 16: Life of spur gear pair for module 5 at pressure angle 14.5˚

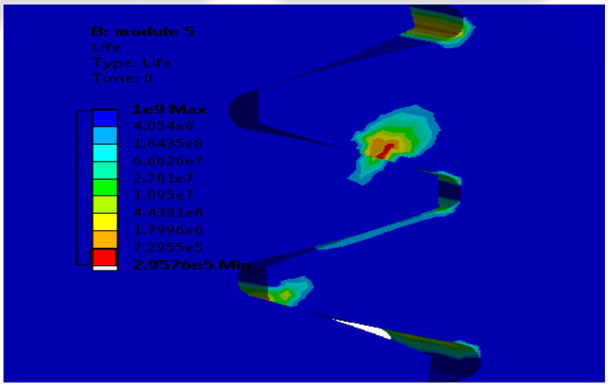

Fig. 17: Life of spur gear pair for module 5 at pressure angle 25˚

The Max. Stress at the contact point, decided in the past area is utilized for exhaustion life estimation of the rigging utilizing strain life approach. Fig. 4.19 demonstrates the weakness life assessed for a rigging of 5mm module and 25˚ weight point and is shaped to be 2.95E07 cycles.

F. Comparison of analytical and FEM results:

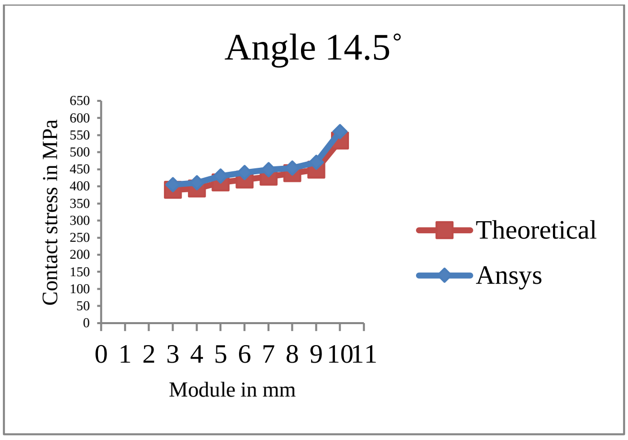

Fig. 18: Contact stress V/s Module at pressure angle 14.5˚

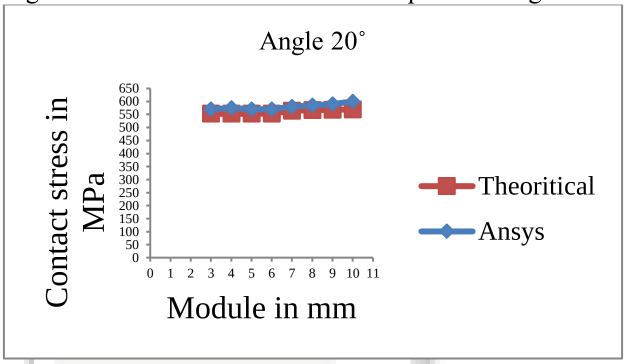

Fig. 19: Contact stress V/s Module at pressure angle 20˚

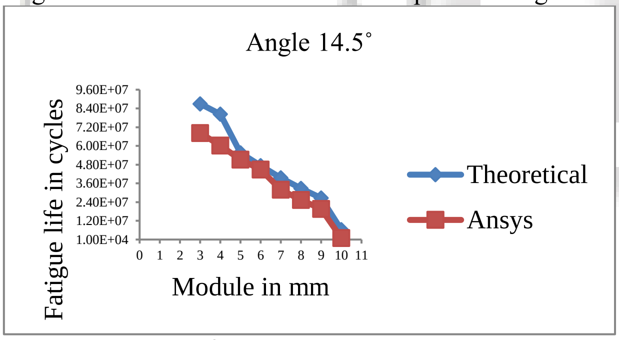

Fig. 20: Fatigue lifeV/s Module at pressure angle 14.5˚

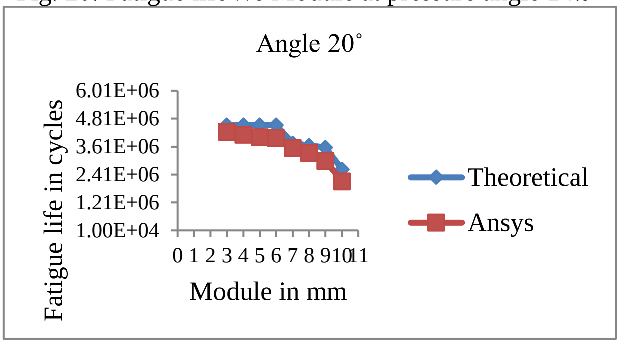

Fig. 20: Gear life V/s Module at pressure angle 20˚

G. Frictional contact stress:

At the point when two apparatus teeth are in cross section, there will be relative moving and additionally sliding movement. Frictional power between the mating teeth restricts the sliding movement. When this frictional power is considered notwithstanding the ordinary power at the line of contact, there will be change in the anxiety incited at the contact surface. Frictional power relies on upon coefficient of grating, which thusly relies on material match and sort of oil. Fig. 4.26 demonstrates the variety of contact anxiety, considering distinctive coefficient of rubbing for 5mm module gear pair.

H. Influence of module and pressure angle on power loss

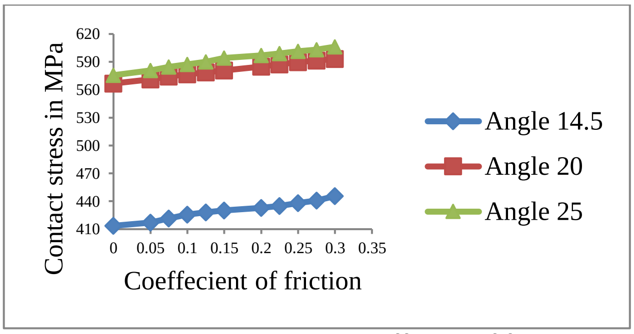

Fig. 21: Contact stress V/s Co-efficient of friction

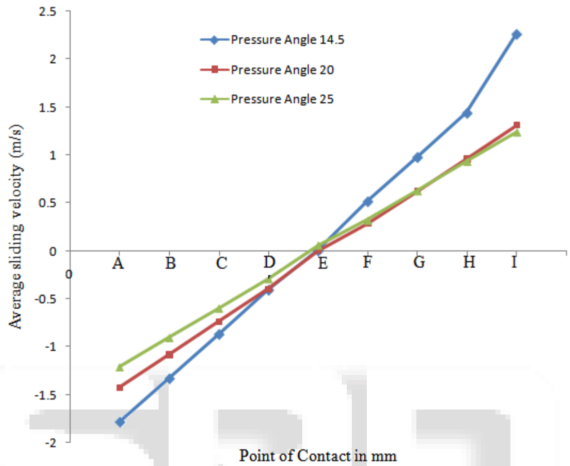

Fig. 22: Variation of sliding velocity along the pathcontact [module=5mm]

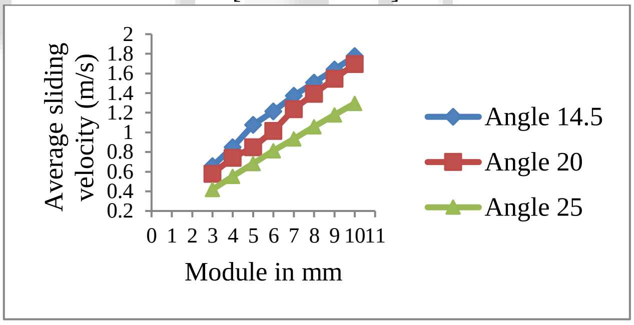

Fig. 23: Average sliding velocity v/s Module

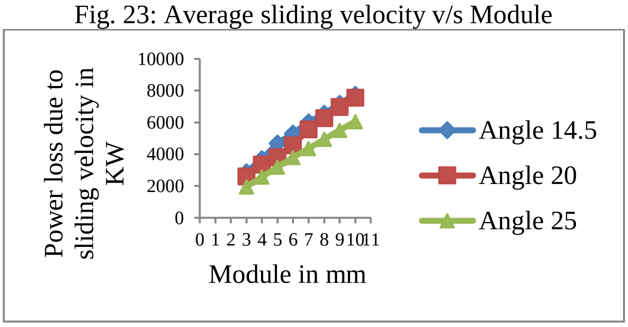

Fig. 24: Power loss due to sliding velocity V/s Module

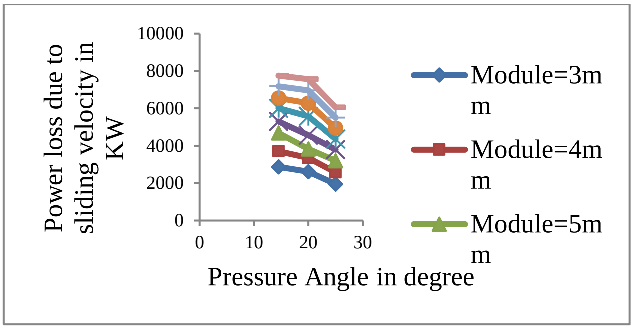

Fig. 25: Power loss due to sliding velocity V/s Pressure angle

VII. Conclusions and Future Scope

- The outcomes demonstrate that the contact proportion between goad apparatuses is conversely corresponding to the module and weight edge.
- The outcomes show that contact stress between goad apparatuses is straightforwardly relative to the module and also weight edge.
- The outcomes demonstrate that weariness life of goad apparatus is contrarily relative to the module and in addition to the weight edge.
- The outcomes demonstrate that the contact stress increments with coefficient of grating.
- The outcomes demonstrate that the normal sliding speed between goad rigging is straightforwardly corresponding to the module and contrarily relative to the weight point.
- Frictional force misfortune increments with expanding module and declines with expanding weight point.
- The outcomes acquired from expository and numerical are observed to be predictable.

VIII. Scope for Future Work

The following areas are worthy of further research. Further analytical, numerical and experimental investigations shall related to,

- Experiment investigation on pitting failure.
- Investigate the crack growth on the contact surface due to pitting phenomenon.

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