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INFLUENCE OF SURFACE COATING ON HERRINGBONE GEAR ASSEMBLY OF AN ELECTRIC VEHICLE

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Abstract: Electric vehicles are gaining more and more popularity all around the globe. As the demand for electric vehicles (EVs) continues to rise, it becomes increasingly important to develop efficient and reliable gear systems to meet the high-performance requirements of these vehicles. Herringbone gears, known for their improved load-carrying capacity and reduced noise, are commonly employed in high-speed applications such as EV powertrains. However, gears used in EVs need to withstand high loads, operate at high rotational speeds, are vulnerable to failure due to the surface stresses and bending that occur at their teeth, which results in a surface fatigue failure known as pitting.

This study aims to investigate the stress analysis of Herringbone gears assembly used in single stage step-down transmission of EV's and evaluate the impact of coating on gear surface for reduced stress to enhance their performance and reliability. In this paper, herringbone gears that are utilised in gear box of EV's are designed using low alloy, 4140 normalized steel in Solid works. Finite Element Analysis (FEA) is performed using Ansys to forecast the equivalent (Von-Mises) stress, the total deformation, the equivalent elastic strain, and the regions susceptible to fatigue failure. Numerical and parametric analysis is performed to evaluate the influence of coating thickness, hardness, adhesion, and other factors on mitigating surface fatigue failures in herringbone gears used in EVs. The results showed that with increasing surface thickness of WC/C coating, corresponding Equivalent (Von-Mises) Stress, Total Deformation and Equivalent Elastic Strain are decreasing. The findings of the study underscore the importance of coating techniques in optimizing the durability and reliability of herringbone gears in EVs, leading to the advancement of gear transmission systems and the overall efficiency of electric vehicles.

Keywords: Design, Modelling, Herringbone Gear, Surface Coating, EV's, Solid works, Ansys.

I. INTRODUCTION

As the demand for electric vehicles (EVs) continues to rise, it becomes increasingly important to develop efficient and reliable gear systems to meet the high-performance requirements of these vehicles by increasing the operating life cycles of individual gear components. Herringbone gears have drawn a lot of interest since they outperform conventional spur gears in terms of carrying capacity of the load and noise output. However, when operating at high RPMs, herringbone gears are susceptible to two critical failure modes: pitting and bending failures.

Pitting failure occurs when localized contact stresses exceed the surface strength of the gear teeth, resulting in the formation of cracks, pits, and surface material loss. This failure mode can significantly compromise the durability and performance of the gear system, leading to increased maintenance costs and decreased vehicle reliability. On the other hand, bending failure occurs due to excessive bending stresses, resulting in tooth deflection or even fractures.

To overcome these challenges and enhance the durability of herringbone gears in high RPM applications, the impact of surface coating has been explored. Surface coatings can provide several benefits, including increased hardness, improved wear resistance, reduced friction, and enhanced resistance to pitting and bending failures. Coating technologies such as diamond-like carbon (DLC) coatings, nitriding, or shot peening have shown promising results in mitigating these failure modes.

By improving the understanding of these failure modes and developing effective mitigation strategies, this research aims to contribute to the development of more durable and reliable gear systems for electric vehicles. The findings will help optimize gear design, material selection, and surface coating approaches to ensure optimal performance, efficiency, and longevity of herringbone gears in high RPM applications, thereby supporting the advancement of electric mobility.
Theoretical calculations for contact stresses can be done using: Bending equation and Hertzian Contact Stress equation.

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A. *Bending Equation*

Bending failure and tooth pitting are the two primary failure types in a gearbox. Bending failure occurs via application of higher loads. By contrasting the estimated bending stress with the material's empirically proven permissible fatigue values, bending failure in gears can be predicted. The Lewis formula helps in determining the Bending stresses and is given as:

$$\sigma_b = \frac{P_d F_t}{bY} \tag{1}$$

Where,
F_t= Normal Tangential Load (N)
P_d=Diametrical Pitch
b= Face width
Y = Lewis form factor.
The Lewis equation solely considers static loading and ignores the dynamics of meshing teeth.
The parabola tangential method refers to the location where the parabola curve intersects the tooth root fillet's curve., and it is at this point that the maximum stress is anticipated. At the tooth root fillet's two tips, one is located on either side. Considered to be the worst case scenario is the strain on the region that connects those two sites.

The following is the AGMA equation for bending stresses:

$$\sigma_b = \frac{F_t}{bmJ} K_v K_O K_S (0.93 K_m) \tag{2}$$

where ,
K_a = Application factor,
K_S = Size factor,
K_m = Load distribution factor,
K_V = Dynamic factor,
F_t = Normal tangential load,
J = Geometry factor,
m= Module

B. *Hertzian Contact Stress Equation*

One of the other significant gear tooth failures is pitting, which results from repetitive high contact pressures on the gear tooth surface occurring when a set of teeth is transmitting power. One of the primary reasons of pitting, which frequently affects the operating surfaces of gear teeth, is excessive loading, which raises contact stresses above the threshold. Comparing the calculated Hertz contact stress to the allowed ranges identified through experimentation for the specific material is the current approach for forecasting contact failure in gears. Hertz's equation is used to calculate gear contact stress. In figures 1 and 2, the contact stresses between the cylinders are shown.

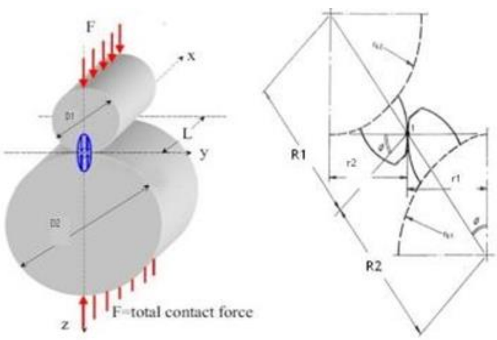

Fig. 1: Cylinders in contact experience compression.

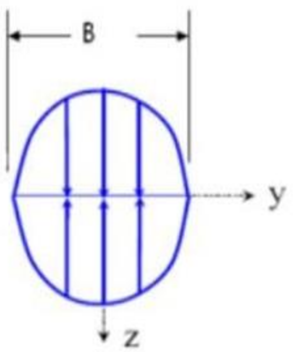

Fig. 2: Elliptical Stresses

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There are always problems when two curved-surfaced objects are pressed together because the deformation produces a contact region that is subject to compressive forces. Because it closely resembles the surfaces of gear teeth, the scenario where the curved surfaces have a cylindrical shape is of special interest to the gear designer. At the pitch point in Fig. 1, two gear teeth are observed mating. Referring to Fig. 2, the narrow rectangle with the measurements B and L stands in for the area of contact under load. Figure 3 shows the elliptical pattern of stress distribution across the breadth.

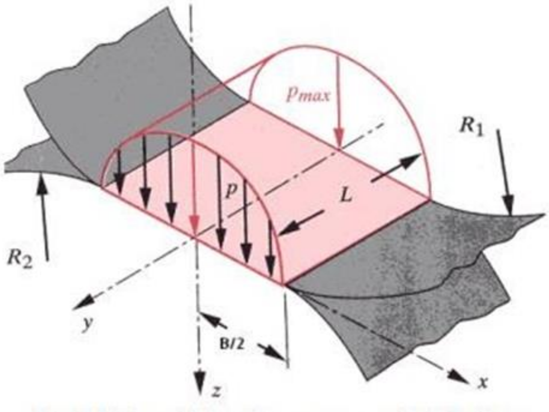

Fig. 3: Ellipsoidal-prism pressure distribution

The current approach to forecasting gear contact failure compares the calculated Hertz contact stress to the permitted values for the particular material discovered through laboratory testing. Hertz's Equation (2) was first developed to calculate the gear contact stress for a contact between two cylinders. The following equation yields the surface compressive stress (Hertzian stress):

$$\sigma_c = \sqrt{\frac{2\,F_t}{\pi b}}\sqrt{\left(\frac{\frac{1}{r_1}+\frac{1}{r_2}}{\frac{1-\upsilon_1^2}{E_1}+\frac{1-\upsilon_2^2}{E_2}}\right)} \tag{3}$$

$$r_1 = \frac{d_1 sin\phi}{2} \tag{4}$$

$$r_2 = \frac{d_2 sin\phi}{2} \tag{5}$$

where,
r_1 = radius of curvature at point of contact on 1st cylinder
r_2 = radius of curvature at point of contact on 2nd cylinder
ϕ = Pressure Angle
E_1 and E_2 = Young's Modulus of both cylinder material
υ_1 and υ_2 = Poissons' Ratio for both cylinder material

The contact stresses that exist between the tooth surfaces of two mating helical gears can be calculated using the Hertz equations mentioned thus far. These gears' contact characteristics can be roughly described as being similar to those of cylinders that have the same radii of curvature at the contact site as the load-transmitting gears. In the case of an involuted curve, the radius of curvature varies continually, changing abruptly in the area of the base circle.

II. METHODOLOGY

In this Paper, commercially available CAD modelling software SolidWorks is used for designing Herringbone gears as per parameters mentioned in table [I] and commercially available CAE software (Ansys Workbench) is applied for simulating gears assembly. The simulation is being performed for uncoated and coated gears respectively. Comparative studies were conducted and tungsten carbide is preferred as the coating material [5], properties of WC/C are depicted in table [II]. Using a static structural analysis system with the specified boundary conditions, the herringbone gear assembly's overall deformation, equivalent (Von-Mises) stress, and equivalent elastic strain were all represented in the

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simulation[3]. Further the parametric analysis is conducted and it is investigated how different coating thickness affects equivalent elastic strain and equivalent (Von-Mises) stress. and plotted.

A. *Designing of Gear*

Considering the module= 5, number of teeth=18 and 20 degrees full depth
Power Transmitted = 15 KW

$$\text{Torque (T)} = \frac{(P\times 60)}{2\pi N} \tag{6}$$

For 10000 rpm,

$$T = \frac{15\times 1000\times 60}{2\times 3.14\times 10000} = 143.3 \text{ N-m} \tag{7}$$

From A Text Book of MACHINE DESIGN by RS Khurmi & JK Gupta
For m=5
Precision gear error=0.015

Pitch Line Velocity (v)=20 m/sec

$$\text{Velocity Factor } (C_v)\ \frac{0.75}{(0.75+\sqrt{v})} = \frac{0.75}{(0.75+\sqrt{20})} = 0.14361 \tag{8}$$

Number of tooth on gear (T_g) = 3 x T_p= 3x18 = 54
Diameter of gear D_g = m x T_g= 5x54 = 270mm

$$\text{Circular Pitch} = \frac{3.14\times D}{T} = \frac{3.14\times 270}{54} = 15.7\text{mm} \tag{9}$$

Face Width (b) = 4πm = 4 x 3.14 x 5 = 62.8 mm

Tangential Force Calculation:

$$\text{Tangential Force } (W_t) = \frac{2000\times T}{D_g} = \frac{2000\times 143.3}{270} = = 1061.41 \text{ N} \tag{10}$$

B. *Modelling of Gear*

SolidWorks is being used for designing the Herringbone Gear Assembly as per the parameters: the number of driver gear teeth in this study is 30, the number of driven gear teeth is 54, the module is 5, and the pressure angle is 20 degrees. The gears are made of Low Alloy Steel, 4140 normalized, with a Young's Modulus of 212 GPa, Poissons Ratio of 0.29, and Density of 7.85 g/cc. Additionally, meshing gears are taken into account to be a unit tooth thickness plane stress problem. Table [I] shows the specifications of the driver and driven gears.

Fig. 4: Herringbone Gear Assembly in SolidWorks

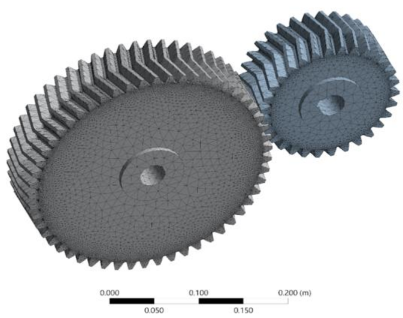

Fig. 5: Meshing of Gear assembly

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TABLE I: Gear Parameters

S.No	Variable name	Description	Value	Unit
1.	z_1	Number of Teeth in Driver Gear	30	-
2.	z_2	Number of Teeth in Driven Gear	54	-
3.	m	Module	5	mm
4.	D_1	Pitch Diameter of Driver Gear	150	mm
5.	D_2	Pitch Diameter of Driven Gear	270	mm
6.	ϕ	Pressure Angle	20	degree
7.	β	Helix Angle	15	degree
8.	F	Face Width	62.8	mm
9.	P	Power Input	15	kW
10.	N	Speed of Driven Gear	10000	RPM

III. RESULTS

The FEA is carried out for Herringbone gears assembly used in single stage step-down transmission of EV's. The Simulations are done on WC/C coated and uncoated gears assembly respectively. Finite element study findings for each condition of herringbone gear are summarized in Table [III]. During the analysis, it is learned that the deformation and rate to gear failure due to surface phenomenon of pitting is smaller in case of WC/C coated gears when compared to that of the uncoated gears.

A. *Results for Uncoated Gears*

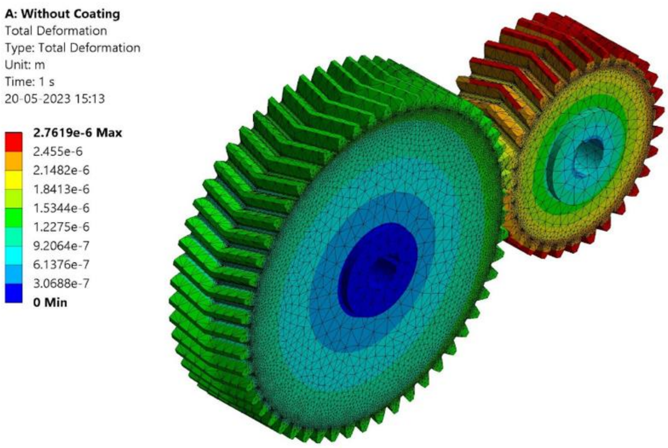

Fig. 6: Total deformation of uncoated gears

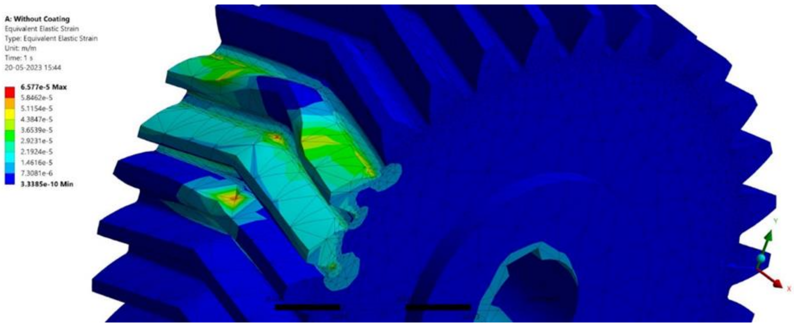

Fig. 7: Equivalent Elastic Strain for Uncoated Gears

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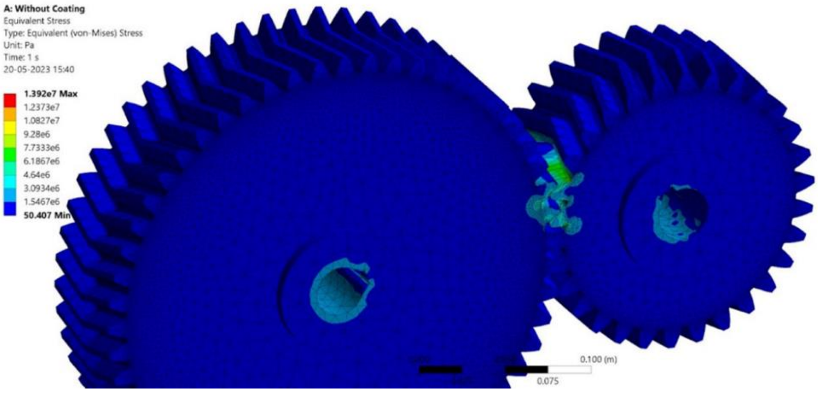

Fig. 8: Equivalent (Von- Mises) Stress for Uncoated Gears

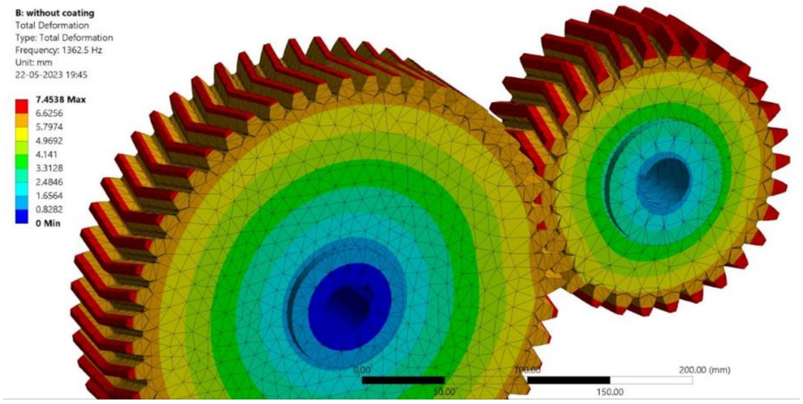

Fig. 9: Modal Analysis for Uncoated Gears

B. Results for WC/C (Tungsten Carbide) Coated Gears

Now Analysis was carried out by coating the Herringbone gear with Tungsten Carbide material having Young's Modulus of 615 GPa, Poissons ratio of 0.31 and Ultimate tensile strength of 344 MPa. Thickness of Coating chosen was 20 microns.

TABLE II: Material Properties of Tungsten Carbide

WC		
Structural		
Isotropic Elasticity		
Derive from	Young's Modulus and Poisson's Ratio	
Young's Modulus	6.15e+11	Pa
Poisson's Ratio	0.31	
Bulk Modulus	5.3947e+11	Pa
Shear Modulus	2.3473e+11	Pa
Compressive Ultimate Strength	2.7e+09	Pa
Tensile Ultimate Strength	3.44e+08	Pa

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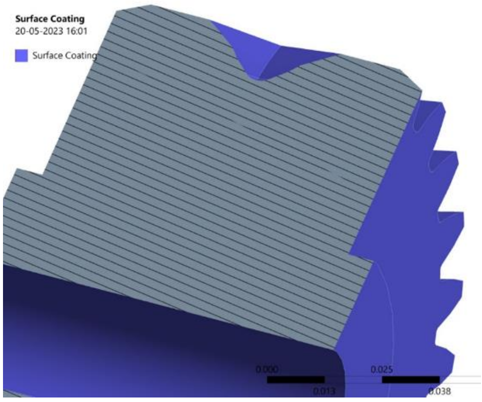

Fig. 10: WC/C Surface Coating

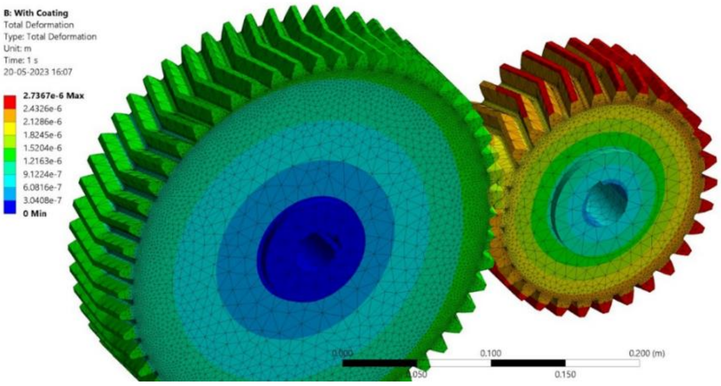

Fig. 11: Total deformation of wc/c coated gears

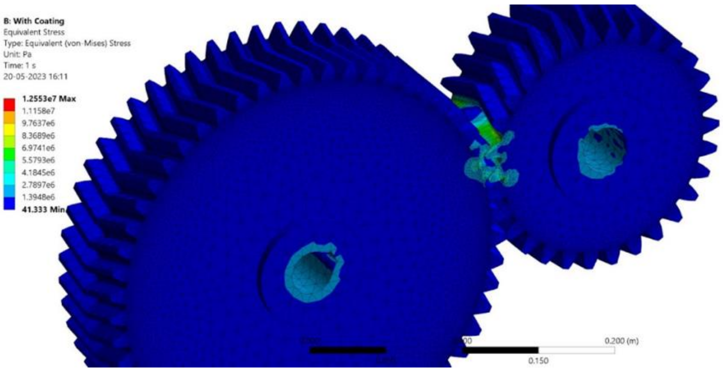

Fig. 12: Equivalent (Von- Mises) Stress for WC/C Coated Gears

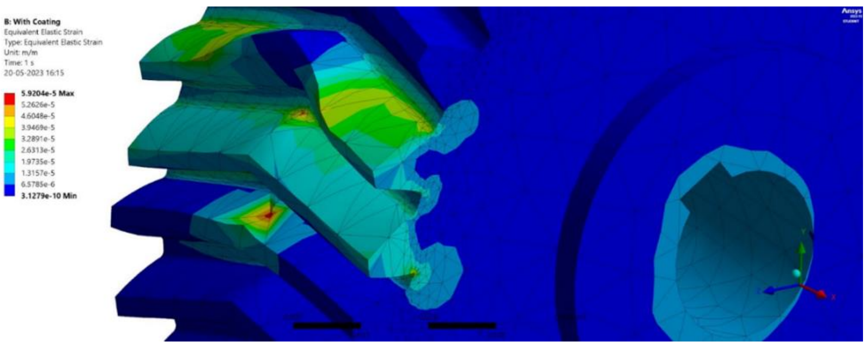

Fig. 13: Equivalent Elastic Strain for WC/C Coated Gears

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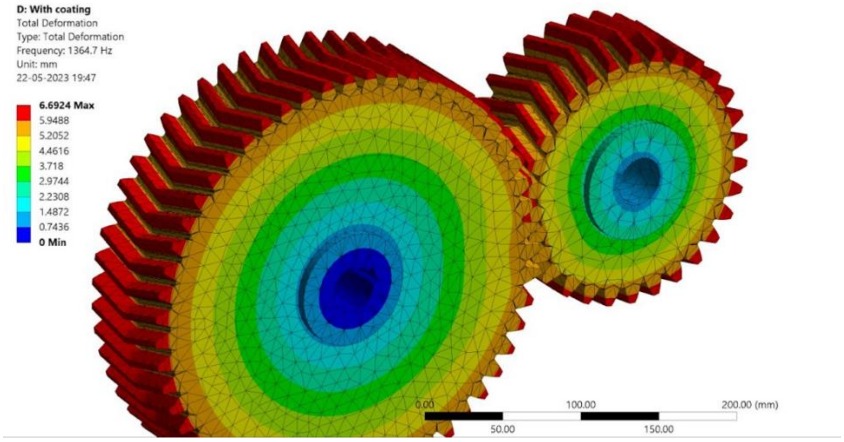

Fig. 14: Modal Analysis for WC/C Coated Gears

IV. DISCUSSION

Parametric study of the results obtained depicts the impact of thickness of the coated material. From the depicted curves as shown in fig. [15][16], it is evident that with increase in the surface thickness of WC/C coating, the corresponding Equivalent (Von-Mises) Stress, Total Deformation and Equivalent Elastic Strain is seen to be decreasing which inclines with the objective of the paper. Therefore, it is evident that material coating on gears increases the operating life of the components accompanied by the decrease in equivalent stress and strains on their surfaces.

TABLE III: Comparative Analysis of Results

S.No.	Without Coating	With Coating (20 m) (Tungsten Carbide)
Total Deformation (m)	2.76 x 10^{-6}	2.73 x 10^{-6}
Equivalent (von-mises) Stress (MPa)	13.9	12.5
Equivalent Elastic Strain (m/m)	6.58 x 10^{-5}	5.92 x 10^{-5}
Total Deformation at 1364 Hz Resonance Frequency (mm)	7.43	6.69

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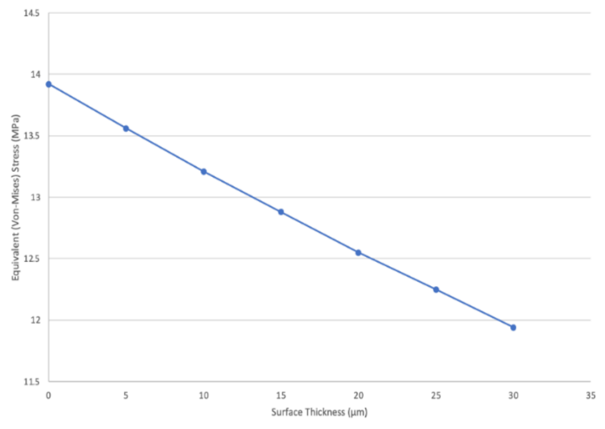

Fig. 15: Equivalent Stresses VS Coating Thickness.

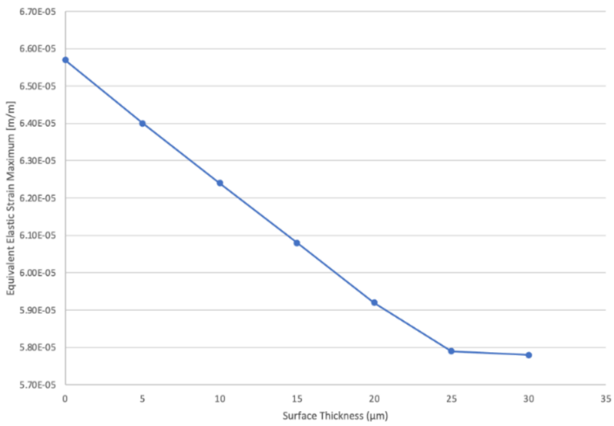

Fig. 16: Equivalent Elastic Strain VS Coating Thickness

V. CONCLUSION

This study has developed the suitable method for reducing the surface stresses for Herringbone gears assembly used in single stage step-down transmission of EV's by means of surface coating. FEA analysis was conducted, for WC/C coated and uncoated gears. To summarise, surface coating is critical in increasing the operational life of gears. subjected to high RPM in electric vehicle (EV) applications. High RPM conditions put significant stress on gears, leading to fatigue failure and reduced lifespan. By applying suitable surface coatings, several benefits can be achieved. Surface coatings can enhance the wear resistance of gears, reducing the friction and minimizing the material loss caused by the repeated contact between gear teeth. This wear resistance is particularly important in high RPM applications where gears experience intense rubbing and sliding forces. The results of this study can be readily opted by the industries of today.

Further this study can be extended to various other industrial applications for different gear drives subjected to high loads and cyclic loading applications. Various composites and carbon based materials can be considered for coating application based on their material economics, strength and surface finishing properties.

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