STRUCTURAL AND THERMAL ANALYSIS OF GEAR TECHNOLOGY

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Abstract

Gear is a machine element used to transmit motion and power between rotating shafts by means of progressive engagement of projections called teeth. Generally gear transmits motion or power between rotating shafts when the centre between two shafts is comparatively low. The aim of the project is to design a helical gear for marine applications by using empirical formulas. A 2D drawing is drafted from the calculations and a 3D model is designed using 3D modeling software Pro/Engineer. Structural analysis and thermal analysis are done using two materials Nickel Chromium Alloy steel and Aluminum Alloy A360. Structural analysis is done to validate the strength and thermal analysis is done to validate the thermal properties like nodal temperature, thermal gradient and thermal flux.

Key words—Nickel Chromium Alloy Steel, Aluminum Alloy A360, Ansys

1. Introduction

A gear is a rotating machine part having cut teeth, or cogs, which mesh with another toothed part in order to transmit torque. Two or more gears working in tandem are called a transmission and can produce a mechanical advantage through a gear ratio and thus may be considered a simple machine. Geared devices can change the speed, torque, and direction of a power source. The most common situation is for a gear to mesh with another gear; however, a gear can also mesh with a non-rotating toothed part, called a rack, thereby producing translation instead of rotation.The gears in a transmission are analogous to the wheels in a pulley. An advantage of gears is that the teeth of a gear prevent slipping.When two gears of unequal number of teeth are combined, a mechanical advantage is produced, with both the rotational speeds and the torques of the two gears differing in a simple relationship.In transmissions which offer multiple gear ratios, such as bicycles and cars, the term gear, as in first gear, refers to a gear ratio rather than an actual physical gear. The term is used to describe similar devices even when the gear ratio is continuous rather than discrete, or when the device does not actually contain any gears, as in a continuously variable transmission.[1]. The helical gear offers high contact and more friction which avoids slippage when compared to spur gear.[2] The contact stresses were examined using 2-D FEM models. The bending stresses in the tooth root were examined using a 3-D FEM model.[3] Accurate evaluation of stress state and distribution of stress is complex task; we have analyzed the stress pattern by using three dimensional Photo elasticity techniques.[4] The direct design approach that is commonly used for most parts of mechanisms and machines(for example, cams, linkages, compressor or turbine blades, etc.) determines their profiles according to the operating conditions and desired performance.[5] Gear drives are used to various kinds of machines like automobiles, metal cutting tools, material handling equipments, rolling mills, marine power plants etc. The friction and other losses in this type of power transmission equipment is comparatively very low.

2. Design Calculation For Helical Gear:

Helical gear in high speed marine applications

Speed of the pinion = 3500rpm

Power = p =9000KW = 9000×10^3
Gear ratio = 7

Center distance = x = $\frac{d_G+d_P}{2}$
Helix angle = 25^0 = α

Material used = 40ni2cr1m028 steel
Properties = BHN =225

Minimum tensile strength = 900 n/mm^2
Young's modulus = $2\times 10^5\ N/mm^2$
Compressive stress = $\sigma_c = 11000\ kgf/cm^2$

Bending stress = $\sigma_b = 4000\ kgf/cm^2$=392265.9 N/mm^2

Module =m= 18

$t_p = no\ of\ teeth\ on\ pinion = 18$

WKT gear ratio = GR = $\frac{T_G}{T_p} = \frac{N_p}{N_G} = \frac{D_G}{D_P}$

We have GR =7 and $T_p = 18$

$$GR = \frac{T_G}{T_p}$$

$$T_G = 18 \times 7 = 126$$

No of teeth on gear = $T_G = 126$

Diameter of gear $D_G = T_G \times m = 126 \times 18 = 2268mm$

Diameter of pinion =$D_P = T_p \times m = 18 \times 18 = 324$

GR = $\frac{N_P}{N_G}$ center distance = $\frac{2268+324}{2} = 1296$

$N_G = speed\ of\ gear = \frac{3570}{7} = 500rpm$
Normal pitch $P_N = P_c \cos\alpha$
$P_c = circuler\ pitch = \frac{\pi D}{T} = \frac{\pi(D_G+D_P)}{T_G+T_P} = \frac{\pi 2592}{144} = 56.52$

$P_N = 56.52 \cos 25 = 51.22$
Normal pressure angle = ϕN

$$\tan\phi N = \tan\phi \times \cos\alpha \qquad (\phi=20)$$

$$\tan\phi N = \tan 20 \times \cos 25 = 0.328$$

$$\phi N = \tan^{-1}(0.328) = 18.210$$

face width of helical gears:

usually recommended that the overlap should be 15 percent of the circular pitch

$$b\tan\alpha = 1.15p_c$$
$$b = \frac{1.15p_c}{\tan\propto} = \frac{64.998}{tan25} = 139.388$$

the maximum face width may taken as 12.5 m to 20m

b = 20m = 360

formative or equivalent no of teeth for helical gears = $T_E = \frac{T}{cos^3\propto}$

equivalent no of teeth on pinion = $T_{EP} = \frac{T_p}{cos^3\propto} = \frac{18}{cos^3 25} = 24.20$

equivalent no of teeth on gear = $T_{EG} = \frac{T_G}{cos^3\propto} = \frac{126}{cos^3 25} = 169.428$

tooth form factor for pinion for 20^0 full depth involute

$$Y^1P = 0.154 - \frac{0.912}{T_{EP}} = 0.154 - \frac{0.912}{24.20} = 0.1163$$

tooth form factor for gear for 20^0 full depth involute

$$Y^1G = 0.154 - \frac{0.912}{T_{EG}} = 0.154 - \frac{0.912}{169.428} = 0.1486$$

Properties for helical gears:

Pressure angle ϕ = 20

Helix angle α = 25^0

Addendum = 0.8M(maximum) = 144

Dedendum = 1M(minimum) = 18

Minimum total depth = 1.8m =32.4

Minimum clearance 0.2M =3.6

Thickness of tooth = 1.5708M =28.2744

Strength of helical gears:

$$W_t = W_T = (\sigma_0 \times C_v) \times b \times \pi \times m \times \text{Y1}$$

$$W_T = tangential\ tooth\ load$$

$\sigma_0 = allowable\ static\ stress$

$C_v = velocity\ factor$

b = face width

M = module

Y^1 =tooth form factors

Both the pinion and gear are made of the same material the pinion is weaker thus the design will be based upon pinion

The allowable static stress (σ_0) for steel gears is approximately one third of the ultimate tensile strength $= \sigma_0 = \frac{\sigma_u}{3}$

$$\sigma_0 = 516.666\ N/mm^2$$

$$\sigma_u = 1500\ N/mm^2$$

Peripheral speed = V $= \frac{\pi D_P N_P}{60\times1000} = \frac{3.14\times324\times3500}{60\times1000} = 59.346$

The value of velocity factor C depending upon peripheral velocities greater than 20 m/s is given by $\frac{0.75}{0.75+\sqrt{V}}$

$$C_V = \frac{0.75}{0.75+\sqrt{V}} = \frac{0.75}{0.75+59.346} = 0.088$$

$W_T = (\sigma_0 \times C_v) \times b \times \pi \times m \times Y^1 P$

$= 516.666 \times 0{,}088 \times 360 \times 3.14 \times 18 \times 0.1163 = 107589.993N$

The dynamic tooth load on the helical gear is given by

$$W_D = W_T + \frac{21V(bccos^2\propto + W_T cos\propto)}{21V+\sqrt{b\times ccos^2\propto + W_T}}$$

Where v, b, c have usual meaning as discussed in spur gears

C = deformation factor $= \frac{k\times e}{\frac{1}{E_p}+\frac{1}{E_G}}$

K =0.111 for 20^0 full depth involute system

E_G and E_p = are youngs modulus of gear and pinion in N/mm^2

σ = tooth error action in mm

σ = 5 used in precision gears is = 0.032

C= $\frac{0.111\times0.032}{\frac{1}{2\times10^5}+\frac{1}{2\times10^5}}$ =355.2 N/mm

$$W_D = W_T + \frac{21V(bccos^2\propto + W_T cos\propto)}{21V+\sqrt{b\times ccos^2\propto + W_T}}$$

$$107589.993 + \frac{21\times59.346(360\times355.2cos^2 25 + 107589.993cos25)}{21\times59.346+\sqrt{360\times355.2cos^2 25 + 107589.993}} = 178805.7627N$$

The static tooth load or endurance strength of the tooth for bevel gear is given by

$W_S = \sigma_e \times b \times \pi \times m \times Y^1$

$\sigma_e = 1.75 \times BHN = 446.25\ Mpa$ (BHN=225) =flexural endurance limit

$W_S = 446.25 \times 360 \times 3.14 \times 18 \times 0.1163 = 1055996.789$

$$W_w = \frac{D_P\times b\times Q\times K}{\cos 2\propto}$$

D_P, b, Q and K have usual meanings as discussed in spur gears in this case

K =load stress factor $= \frac{(\sigma_{es})^2 \sin\phi N}{1.4}\left(\frac{1}{EP}+\frac{1}{EG}\right)$

$\sigma_{es} = 2.8 \times BHN - 70 = 644\ N/mm^2$

K=$\frac{(644)^2 \sin 18.210}{1.4}\left(\frac{1}{2\times10^5}+\frac{1}{2\times10^5}\right) = 0.925$

Q = $\frac{2\times VR}{VR+1} = 1.75 \quad = (VR = 7)$

$W_w = \frac{324\times360\times1.75\times0.925}{cos^2 25} = 230022.806N$

4.Design for pinion shaft

Tangential load on pinion $W_T = 107589.993N$

Axial load of pinion $W_T = W_T \tan \propto = 107589.993\tan 25 = 50170.037$

Bending moment of pinion shaft $M_1 = W_T \times x$

X = over hang =1296

$M_1 = 50170 \times 1296 = 1394.36630.928$

Bending moment of pinion shaft due to the axial load $= M_2$

$$= W_A \times \frac{D_P}{2} = 50170.037 \times \frac{324}{2} = 8127545.994$$

$$resultent\ bending\ moment\ oon\ pinion\ shaft = M = \sqrt{{M_1}^2 + {M_2}^2}$$

$$= \sqrt{139436630.928^2 + 8127545.994^2} = 139673301.4875$$

Torque transmitted by pinion T = $\frac{P \times 60}{2\pi N_P} = \frac{9000\times 10^3 \times 60}{2\pi 3500} = 24567.788\ N - mm$

Equivalent twisting moment $T_e = \sqrt{M^2 + T^2}$

$= \sqrt{292403108.788^2 + 24567.788^2} = 139673303.648\ N - mm$

We know that equivalent twisting moment $T_e = \frac{\pi}{16} \times \tau {D_P}^3$

$139673303.648 = \frac{\pi}{16} \times 230{D_P}^3$

${D_P}^3 = 3094396.093 = D_P = 145.722mm$

Let us now check for the principle shear stress WKT the shear stress induced

$$\tau = \frac{16T_e}{\pi {D_P}^3} = \frac{16 \times 139673303.648}{\pi 150^3} = 211Mpa$$

direct stress due to axial load $= \sigma = \frac{W_A}{\frac{\pi}{4}{D_P}^2} = \frac{50170.037}{\frac{\pi}{4}150^2} = 2.8$

principle shear stress = $\frac{1}{2}\sqrt{\sigma^2 + 4\tau^2} = \frac{1}{2}\sqrt{2.8^2 + 4 \times 211^2} = 211$

the principle shear stress is less than the permissible shear stress of 230 Mpa therefore the design is satisfactory

WKT the diameter of pinion hub $= 1.8d_p = 1.8 \times 150 = 270$

Length of the hub = $1.25 \times 150 = 187.5 = 190$

If the pitch circle diameter of the pinion is less than or equal to 14.7M+60mm = $14.7 \times 18 + 60 = 325.5$

$$thickness$$

$$equal\ to\ the\ face\ width\ thickness = b = 360mm$$

5.Design for the gear shaft

Let D_G = diameter of gear shaft

We have already calculated that the tangential load = $W_T = 107589.993N$

Axial load = $W_A = 50170.037N$

Bending moment due to the tangential load $= M_1 = W_T \times over\ hang\ (x)$

= 139436630.928 (x = 1296)

Bending moment due to axial load = $M_2 = W_A \times \frac{D_G}{2}$

$$= 50170.037 \times \frac{2268}{2} = 56892821.96$$

$resultent\ bending\ moment\ of\ pinion\ shaft = M = \sqrt{{M_1}^2 + {M_2}^2}$ = 56892839.04

$VR = 7$ *the gear shaft is subjected to a torque to 3 times the torque on the pinion shaft*

Torque on the gear shaft = T

=torque on the pinion shaft$\times VR = 24567.788 \times 7 = 171974.516$

WKT equivalent twisting moment = $T_e = \sqrt{M^2 + T^2}$

$$= \sqrt{56892839.04^2 + 171974.516^2} = 56893098.959\ N - mm$$

We also know that equivalent twisting moment $= T_e = \frac{\pi}{16} \times \tau \times {D_G}^3$

$$56893098.959 = \frac{\pi}{16} \times 230 \times {D_G}^3$$

$${D_G}^3 = 1260439.744 = D_G = 108mm$$

Let us know check for the principle shear stress;

WKT = shear stress $= \tau = \frac{16T_e}{\pi {D_G}^3} = \frac{16 \times 56893098.959}{\pi 108^3}$ =230

Direct stress due to axial load $\sigma = \frac{W_A}{\frac{\pi}{4}{D_G}^2} = \frac{50170.037}{\frac{\pi}{4}108^2} = 5.47$

Principle shear stress $= \frac{1}{2}\sqrt{\sigma^2 + 4\tau^2} = \frac{1}{2}\sqrt{5.47^2 + 4 \times 230^2} = 230Mpa$

The principle shear stress is same as the permissible stress of 230 Mpa the design is satisfactory

WKT diameter of gear hub = $1.8d_G$=194.4

Length of hub =$1.25d_G$=135

Length of hub is < face width i.e = 360mm

6.Introduction To Pro/Engineer

Pro/ENGINEER is the industry's standard 3D mechanical design suit. It is the world's leading **CAD/CAM /CAE** software, gives a broad range of integrated solutions to cover all aspects of product design and manufacturing.

DIFFERENT MODULES IN PRO/ENGINEER

- PART DESIGN
- ASSEMBLY
- DRAWING
- SHEETMETAL
- MANUFACTURING

Model Of Helical Gear

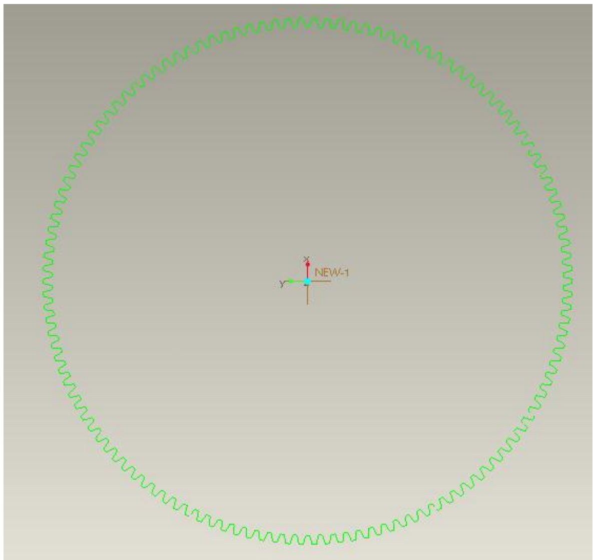

Fig.1 Model of helical gear

Types of Structural Analysis

Static Analysis--Used to determine displacements, stresses, etc. under static loading conditions. Both linear and nonlinear static analyses. Nonlinearities can include plasticity, stress stiffening, large deflection, large strain, hyperelasticity, contact surfaces, and creep.

Modal Analysis--Used to calculate the natural frequencies and mode shapes of a structure. Different mode extraction methods are available.

Harmonic Analysis--Used to determine the response of a structure to harmonically time-varying loads.

Transient Dynamic Analysis--Used to determine the response of a structure to arbitrarily time-varying loads. All nonlinearities mentioned under Static Analysis above are allowed.

- Fracture mechanics
- Composites
- Fatigue
- p-Method
- Beam Analyses

Types of Thermal Analysis

ANSYS supports two types of thermal analysis:

1. **A steady-state thermal analysis** determines the temperature distribution and other thermal quantities under steady-state loading conditions. A steady-state loading condition is a situation where heat storage effects varying over a period of time can be ignored.
2. **A transient thermal analysis** determines the temperature distribution and other thermal quantities under conditions that vary over a period of time.

7.Structural Analysis Of Helical Gear Using Nickel Chromium Molybdnum Alloy Steel

Imported model of helical gear from Pro/Engineer

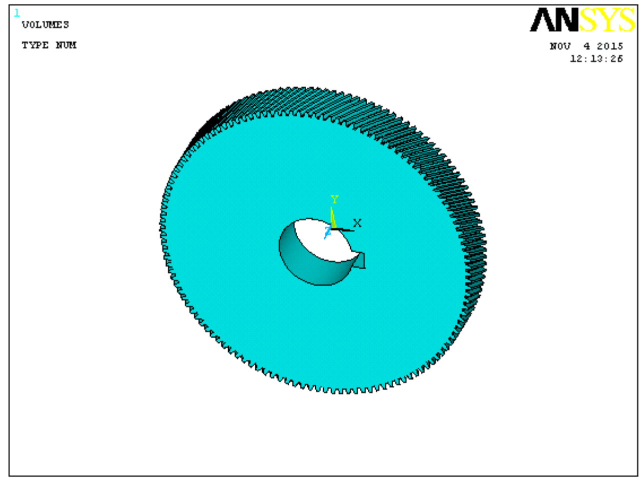

Fig.2 (a) Element type and material property

Element Type – Solid 20 Node 95

Material Properties -

Young's modules – 210000MPa

Poisson ratio - 0.3

Density – 0.000008kg/mm^3

Meshed Model

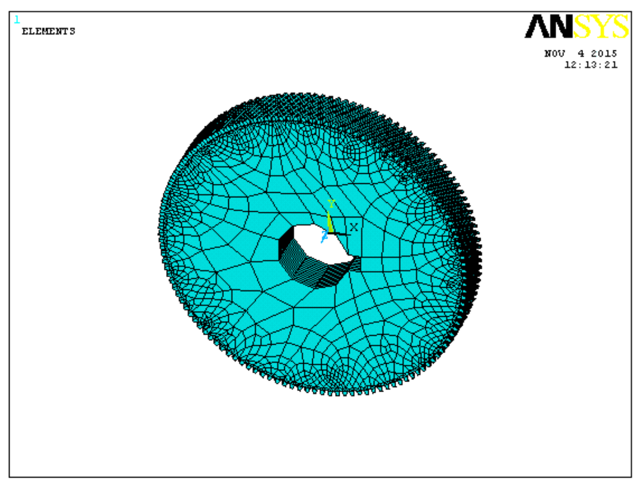

Fig.2(b) Mesh lines and area,volume of helical gear

Click> OK

Loads

Pressure –18.210N/mm^2

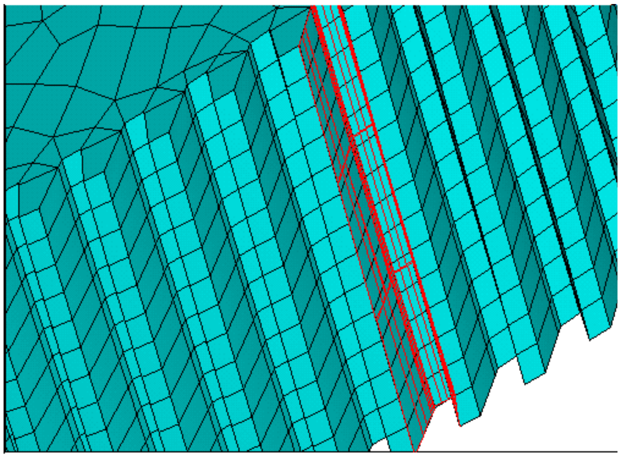

Fig.2(c) Area and volume of helical gear

Solution

Solution – Solve – Current LS – ok

Post Processor

General Post Processor – Plot Results – Contour Plot - Nodal Solution – DOF Solution – Displacement Vector Sum

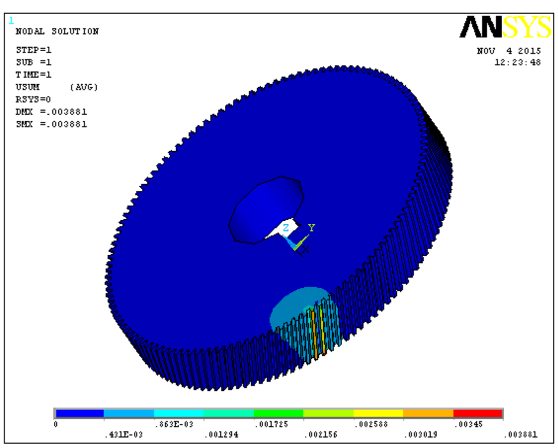

Fig.2(d) Nodal displacement of helical gear

General Post Processor – Plot Results – Contour Plot – Nodal Solution – Stress – Von Mises Stress

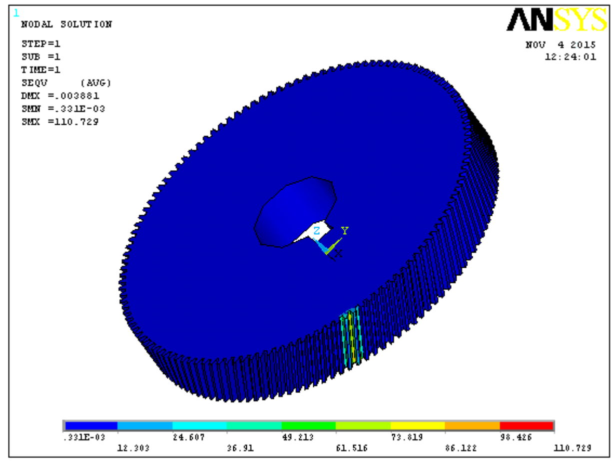

Fig.2(e) Deflection plot and stress on helical gear

8.Thermal Analysis of Helical Gear using Nickel

Chromium Molybdenum Alloy Steel

Imported Model from Pro/Engineer

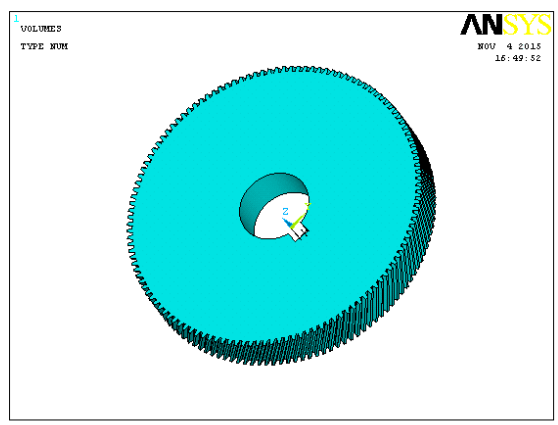

Fig.3(a) Element type and material property

Element Type: Solid 20 node 90

Material Properties: Thermal Conductivity – 0.42w/mmk

Specific Heat – 477j/kg k

Density – 0.000008kg/mm^3

Meshed Model

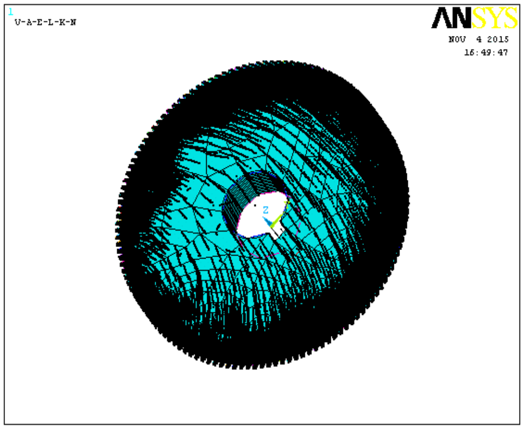

Fig.3(b) Mesh lines and area

Apply Loads

Loads – Define Loads – Apply – Thermal – Temperature

Temperature – 373k

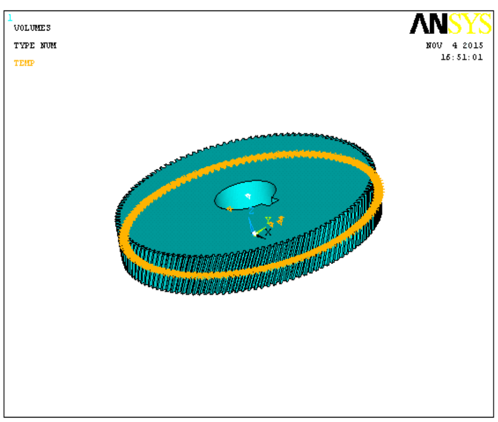

Fig.3(c) Apply load on helical gear

Loads – define Loads – Apply – Thermal – Heat flow – On nodes

Heat flow – 2kj/sec

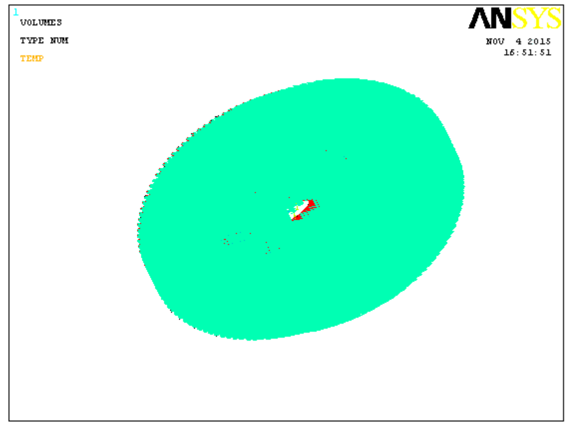

Fig.3(d) Heat flow on helical gear

Loads – define Loads – Apply – Thermal – Convection

– on areas

Bulk Temperature – 273k

Film Coefficient – 222W/mmK

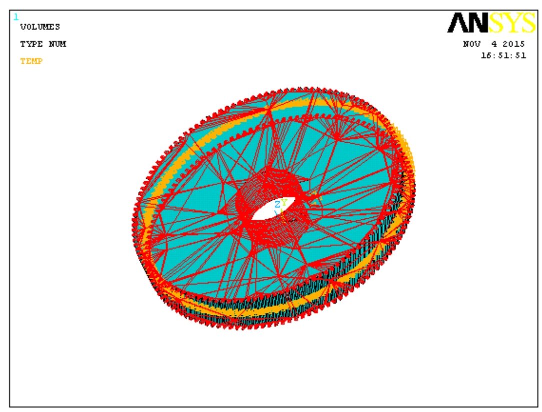

Fig.3(e) Bulk temparature and film coefficient

Solution

Solution – Solve – Current LS – ok, Post Processor

General Post Processor – Plot Results – Contour Plot - Nodal Solution – DOF Solution – Nodal Temperature Vector sum

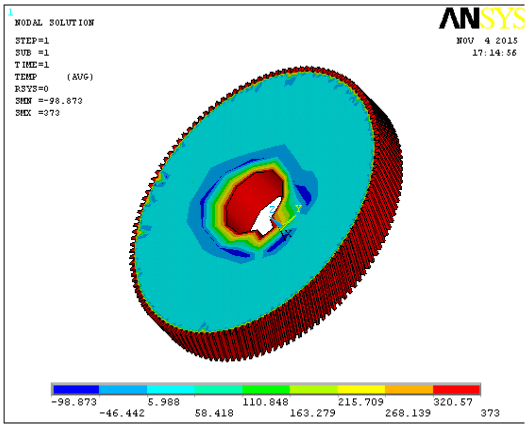

Fig.3(f) Nodal displacement and nodal temparature

General Post Processor – Plot Results – Contour Plot - Nodal Solution – Thermal Gradient Vector sum

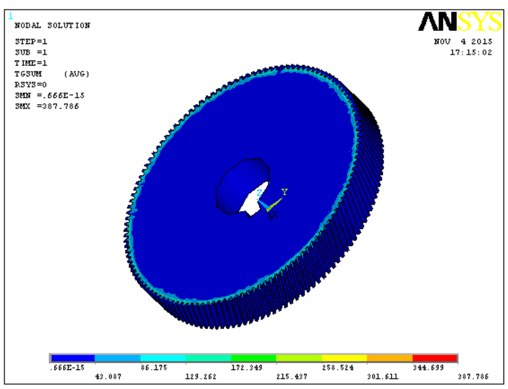

Fig.3(g) Nodal solution and thermal gradient

General Post Processor – Plot Results – Contour Plot -

Nodal Solution – Thermal flux vector sum

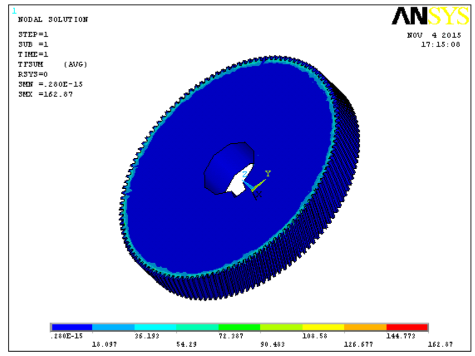

Fig.3(h) Thermal flux on helical gear

9.RESULTS

Table-1

	Displacement (mm)	Von Mises Stress (N/mm^2)	Nodal Temperature (K)	Thermal Gradient (K/mm)	Thermal Flux (W/mm^2)
NickelChromium Molybdnum Alloy Steel	0.003881	110.729	373	387.786	162.87
Aluminum Alloy A360	0.010722	96.003	373	1069	186.056

As per the analysis image

The yield stress for Nickel Chromium Alloy Molybdenum Steel 360Mpa.

The yield stress for Aluminum Alloy A360 is 165Mpa.

CONCLUSION

In our project, we have designed a helical gear used in marine applications using theoretical calculations and modeling of helical gear is done in Pro/Engineer.

- We have performed Structural analysis and thermal analysis on helical gear using Nickel Chromium Molybdenum alloy steel and Aluminum alloy A360.
- By observing the analysis results, the stress values obtained are less than their respective yield stresses for both materials. So we can decide that our design is safe under working conditions.
- By comparing the analysis for both the materials, the stress value is less for Aluminum alloy A360 than Nickel Chromium Molybdenum Alloy Steel and thermal conductivity is more for Aluminum alloy A360 than Nickel Chromium Molybdenum Alloy Steel.
- So we can say that using Aluminum alloy A360 for helical gear is more advantageous than using Nickel Chromium Molybdenum Alloy Steel as per our analysis.
- We also studied the manufacturing process of helical gear and prepared a prototype.

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