ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

Contact Stress Analysis of Helical Gear by Using Finite Element Analysis and Numerical Methods



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ABSTRACT:

Gears are mainly used to transmit the power in mechanical power transmission systems. These gears play a most predominant role in many automobile and micro electro mechanical systems. One of the main reasons of the failure in the gear is bending stresses and vibrations also to be taken into account. But the stresses are occurred due to the contact between two gears while power transmission process is started. Due to meshing between two gears contact stresses are evolved, which are determined by using analyzing software called ANSYS. Finding stresses has become most popular in research on gears to minimize the vibrations, bending stresses and also reducing the mass percentage in gears. These stresses are used to find the optimum design in the gears which reduces the chances of failure. The model is generated by using CATIAV5 and ANSYS is used for numerical analysis. The analytical study is based on Hertz's equation. Study is conducted by varying the geometrical profile of the teeth and to find the change in contact stresses between gears. It is therefore observed that more contact stresses are obtained in modified gears. Both the results calculated using ANSYS and compared according to the given moment of inertia.

I.INTRODUCTION:

Gears are most commonly used for power transmission in all the modern devices. The toothed wheels are used to change the speed or power between input and output. They have gained wide range of acceptance in all kinds of applications and have been used extensively in the high-speed marine engines.

In the present era of sophisticated technology, gear design has evolved to a high degree of perfection. The design and manufacture of precision cut gears, made from materials of high strength, have made it possible to produce gears which are capable of transmitting extremely large loads at extremely high circumferential speeds with very little noise, vibration and other undesirable aspects of gear drives. A gear is a toothed wheel having a special tooth space of profile enabling it to mesh smoothly with other gears and power transmission takes place from one shaft to other by means of successive engagement of teeth.Gears operate in pairs, the smallest of the pair being called "pinion" and the larger one "gear". Usually the pinion drives the gear and the system acts as a speed reducer and torque converter.

Advantages of gear drives are the following are the advantages of the gear drives compared to other drives. Gear drives are more compact in construction due to relatively small center distance; Gears are used where the constant velocity ratio is desired, Gears can be operated at higher speeds and It has higher efficiency.Disadvantages of gear drives are these not suitable for the shafts which are at longer center distance, Manufacturing is complex. It needs special tools and equipment, require perfect alignment of shafts, requires more attention to lubrication, and the error in cutting teeth may cause vibration and noise during operation. The gears, according to the peripheral velocity of the gears, may be classified as: (a) Low Velocity, (b) Medium Velocity and (c) High Velocity.The gears, according to the type of gearing may be classified as (a) External Gearing, (b) Internal Gearing, and (c) Rack and Pinion.

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II.HELICAL GEARS:

Helical gears offer a refinement over spur gears. The leading edges of the teeth are not parallel to the axis of rotation, but are set at an angle. Since the gear is curved, this angling causes the tooth shape to be a segment of a helix. The angled teeth engage more gradually than do spur gear teeth. This causes helical gears to run more smoothly and quietly than spur gears. Helical gears also offer the possibility of using non-parallel shafts.

With parallel helical gears, each pair of teeth first makes contact at a single point at one side of the gear wheel; a moving curve of contact then grows gradually across the tooth faceto a maximum then recedes until the teeth break contact at a single point on the opposite side. It may span the entire width of the tooth for a time. Finally, it recedes until the teeth break contact at a single point on the opposite side of the wheel. Thus force is taken up and released gradually.

Quite commonly helical gears are used with the helix angle of one having the negative of the helix angle of the other; such a pair might also be referred to as having a right-handed helix and a left-handed helix of equal angles. The two equal but opposite angles add to zero: the angle between shafts is zero – that is, the shafts are parallel. Where the sum or the difference (as described in the equations above) is not zero the shafts are crossed. For shafts crossed at right angles the helix angles are of the same hand because they must add to 90 degrees.

Helical gear nomenclature:

Helix angle, ψ, Angle between a tangent to the helix and the gear axis is zero in the limiting case of a spur gear. Normal circular pitch, pn is Circular pitch in the plane normal to the teeth.

Transverse circular pitch, p is Circular pitch in the plane of rotation of the gear.

Sometimes it just called circular pitch.
pn= pcos(ψ)k

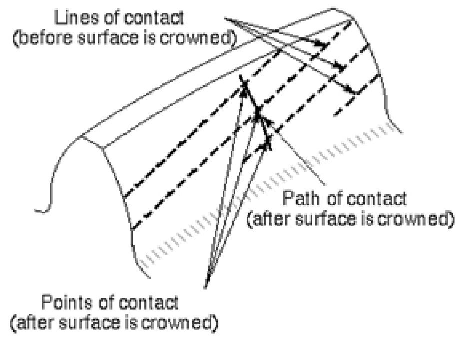

Fig 1: Helical gear nomenclature

Helical Gear geometrical proportions:

p = Circular pitch = d g. p / z g = d p. p / z p
pn = Normal circular pitch = p .cosβ
Pn =Normal diametrical pitch = P /cosβ
px = Axial pitch = p c /tanβ
mn =Normal module = m / cosβ
αn = Normal pressure angle = tan -1(tanα.cos β)
β =Helix angle, dg = Pitch diameter gear = z g. m
dp = Pitch diameter pinion = z p. m
a =Center distance = (z p + z g)* m n /2 cos β
aa= Addendum = maf=Dedendum = 1.25*m

III.THEORETICAL DESIGN CALCULATIONSCALCULATIONS FOR STEEL [40 Ni2 Cr1 Mo28 STEEL]:

Input parameters:

Power P=9000KW, Speed of Pinion N= 3500 r.p.m
Gear Ratio, i=7, Helix Angle, β=25°

Material Selection:

The material for pinion & Gear is 40 Ni2 Cr1 Mo28 steel. Its design compressive stress & bending stresses are [σc=11000kgf/cm2], [σb=4000kgf/cm2]

Properties for 40 Ni2 Cr1 Mo28 Steel:

Density of 40 Ni2 Cr1 Mo28 Steel (ρ) = 7860 kg / m3, Young'sModulus = 215*105 N /m m2, Poisson's ratio (ν) = 0.30, i= 7 ψ=0.3, [MT]=MTkdk = 97420 KW/N kdk=1.3.

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Now, minimum centre distance based on the surface compressive strength is given by
a≥(i+1) 3√[0.7/σc]2 x E[Mt]/(iψ) ≥ 88.41 cm
Minimum module based on beam strength:
mn ≥1.15cosβx3√[MT/(YVσbψmZ1)]
Let Z1 =18, ψm=10
Virtual number of teeth ZV =Z1/cos3β=18/0.744=25.
Lewis form factor YV =0.154 -0.912/ ZV =0.4205mn≥1.15cos25x3√[325661.14/(.4205*4000*10*18)]
mn ≥10.67 mm, But for mn =11-16 mm, σc and σbare ≥ [σc]&[σb] also FS <FD which makes design unsafe.So mn=18mm=1.8 cm
No of teeth on pinion= Z1=2acosβ/ mn(i+1)=12.
But in order to avoid interference, Z1 is taken as 18.
No of teeth on gear, Z2 =i Z1 =126
Diameter ofpinion= (mn*Z1)/cosβ=1.8*18/cos25°= 35.74 cm, Diameter of the gear= i*d1 =7*35.74=250.24 cm, Centre distance, a = (d1+ d2)/2 =142.99 cm, Face width, b= ψa= 0.3*142.99= 43 cm (or)b= ψmmn=10*1.8=18, Taking maximum value of both b is 43 cm.

Checking Calculations:

σc=0.7*[(i+1)/a] √(i+1)*E*[MT] / (ib) ≤ [σc]
σb= 0.7(i+1) [MT] / (ab mnYV) ≤ [σb]

Based on the compressive stress:

σc=[0.7*8*√(8*2.15*106*325661.14)/(7*43)]=180.9

Based on the bending stress:

σb= [(0.7*8*325661.14) / (88.4*43*0.4205)]=228.35
From the calculations, σc&σb values are less than the [σc] & [σb] values of given material, i.e 40 Ni2 Cr1 Mo28. Therefore our design is safe.
Addendum, mn=18mm,Dedendum=1.25* mn=22.5 mm,Tip circle diameter of the pinion= d1+(2*addendum) =357.4+3.6 = 393 mm, Tip circle diameter of the gear= d2+(2*addendum) =2502+3.6 = 2538 mm, Root circle diameter of pinion= d1-(2*addendum) =357.4-3.6 = 312.4 mm, Root circle diameter of gear=d2-(2*addendum)=2502-3.6=2457.4 mm, When the gear transmits power P, the tangential force produced due to the power is given by,FT=(P*KS)/V, V=(πDPNP)/(60*1000) = (357.4*3500* π)/(60*1000)=65.42 m/s, FT=(9000*1000*2)/65.42=275145.21 N

Lewis derived the equation for beam strength assuming the load to be static. When the gears are running at high speeds, the gears may be subjected to dynamic effect. To account for the dynamic effect, a factor known as CV known as velocity factor or dynamic factor is considered. The design tangential force along with dynamic effect is given by
FD= FT * CV = (P* KS * CV)/V.
The velocity factor CV is developed by Barth. It depends on the pitch line velocity and the workmanship of the manufacturer and it is given by, CV = (5.5+√V)/5.5 for V>20m/s.
FD= FT * CV = (P* KS * CV)/V, where FT=275145.21 N, KS=2, V=65.42 m/s, CV=(5.5+√65.42)/5.5 = 2.47, FD=275,145*2.47 = 679771.90 N.
According to Lewis equation, the beam strength of helical gear tooth is given by
FS= [σe]*b* π* mn* YV = (1.75*341)*430* π*18*0.4205=6101677.663N(or)FS= [σb]*b* π*mn*YV =4000*43*π*1.8*0.4205=408994.94N
Since,FS> FD,Our design is safe.
Considering the above conditions, Buckingham derived an equation to find out the maximum load acting on the gear tooth which is given byFD= FT + FI, Where FD = Max Dynamic Load FT =Static load produced by power FI=Incremental load due to dynamic action
Incremental load depends on the pitch line velocity, face width of a gear tooth, gear materials, accuracy of cut and the tangential load and it is given byFI= [0.164Vm (cbcos2β+ FT) cosβ] / [0.164Vm+1.485√ cbcos2β+ FT].
Where Vm is the pitch line velocity in m/s, b is the face width of the gear tooth; c is the dynamic factor or deformation factor in N/mm.
Deformation factor c can also be selected from
Here c = 11860*e, c= 11860*0.026=308.36 N/mm.FT=137572.60 N, Vm=65.42 m/s=65.42*103 mm/s, b=38.16 cm=381.6mm, β=25°, FD = FT + FI= FT + [0.164Vm (cbcos2β+ FT) cosβ] / [0.164Vm+1.485√ cbcos2β+ FT]
=140776.62 N
Since FS > FD, our design is safe.
Based on Hertz theory of contact stresses, Buckingham derived an equation for wear strength of gear tooth which is given by FW= (D1*b*Q*KW)/cos2β, Where, FW is the max or limiting load for wear in N.D1 is the pitch circle diameter of the pinion (mm), b is the face width of the pinion(mm), Q is the ratio factor.
Q=Ratio factor= 2i/(i+1)=1.75,KW=2.553N/mm2
D1=357.4 mm, b = 430 mm.

FW = (D1*b*Q*KW)/cos3β =(357.4*430*1.75*2.553)/cos225 =900086.75N FD=140776.62N, Since FW> FD, our design is safe.

Design of hole:

Design is based on the torque transmitted by the shaft, Torque transmitted T= (60*P)/2πN=(60*9000*103)/(2π*3500)=24555.3N-mm
We know that torque transmitted by the shaft T= (πd3*fs)/16=d=125 mm

Key dimensions:

Width of key (w) =d/4=32 mm
Thickness of key (t) =d/6=18 mm
Calculations for ceramics[98%Al2O3, 0.4-0.7% Mn, 0.4-0.7%Mg]:
Input parameters:Power = P=9000 KW, Speed of Pinion: N= 3500 r.p.m, Gear Ratio, i=7, Helix Angle, β=25°.

Material Selection:

Let the material for pinion & Gear is Aluminum Alloy.
Its design compressive stress & bending stresses are [σc=25000kgf/cm2], [σb=3500kgf/cm2]

Properties for Aluminum Alloy:

Density (ρ) = 3900 kg / m3, Young'sModulus = 340*103N/mm2, Poisson's ratio (v) = 0.220, i= 7, ψ=0.3, [MT]= MT-kdk, MT=97420 KW/N, kdk=1.3, [MT]= MT kDk
=(97420 x 9000 x 1.8)/3500
= 325661.14 kgf-cm
Now, minCentre distance based on the surface compressive strength is given by
a≥(i+1) 3√[0.7/σc]2 x E[Mt]/(iψ)≥59.59 cm
Minimum module based on beam strength:
mn≥1.15cosβx3√[MT/(YVσbψmZ1)] Let Z1 =18, ψm=10
Virtual no. of teeth ZV=Z1/cos3β=18/0.744=25
Lewis form factor YV =0.154 -0.912/ ZV=0.4205
mn≥1.15cos25x3√[325661.14/(0.4205*3500*10*18)]≥1.11 cm≥11.16 mm
But for mn =11-16 mm, FS <FD which makes design unsafe.So mn=18mm=1.8 cm
No of teeth on pinion= Z1=2acosβ/ mn(i+1)=12.
But in order to avoid interference, Z1 is taken as 18.No of teeth on gear, Z2 =i Z1 =126
Diameter of the pinion= (mn*Z1)/cosβ
=1.8*18/cos25° = 35.74 cm
Diameter of the gear= i*d1 =7*35.74=250.24 cm
Centre distance, a = (d1+ d2)/2 =142.99 cm
Face width, b= ψa= 0.3*142.99= 43 cm (or)
b= ψmmn=10*1.8=18, Taking maximum value of both b is 43 cm.

Checking Calculations:

σc=0.7*[(i+1)/a] √(i+1)*E*[MT] / (ib) ≤ [σc]
σb= 0.7(i+1) [MT] / (ab mnYV) ≤ [σb]

Based on the compressive stress:

σc = [((0.7*8)/60)*√(8*340*10 4 *325661.14) / (7*43)]=150.65 N/mm2

Based on the bending stress:

σb=[(0.7*8*325661.14)/(60*43*0.4205)]=218.94 N/mm2, From the calculations, σc&σb values are less than the [σc] & [σb] values of given material, i.e 40 Ni2 Cr1 Mo28. Therefore our design is safe. Addendum, mn=18 mm,Dedendum =1.25* mn=22.5 mm,
Tip circle diameter of the pinion= d1+(2*addendum) =357.4+3.6 = 393 mm
Tip circle diameter of the gear=d2+(2*addendum) =2502+3.6 = 2538 mm
Root circle diameter of pinion= d1-(2*addendum) =357.4-3.6 = 312.4 mm
Root circle diameter of gear= d2-(2*addendum) =2502-3.6 = 2457.4 mm
When the gear transmits power P, the tangential force produced due to the power is given by,
FT=(P*KS)/V, V=(π DP NP)/(60*1000) = (357.4*3500*π)/(60*1000)=65.42 m/s
FT=(9000*1000*2)/65.42=275145.21 N
The design tangential force along with dynamic effect is given by
FD= FT * CV = (P* KS * CV)/V
The velocity factor CV is developed by Barth. It depends on the pitch line velocity and the workmanship of the manufacturer and it is given
by, CV = (5.5+√V)/5.5 for V>20m/s. FD= FT * CV = (P* KS * CV)/V, where FT=275145.21 N, KS=2, V=65.42 m/s, CV=(5.5+√65.42)/5.5 = 2.47, FD=275,145*2.47 = 679771.90 N

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According to Lewis equation, the beam strength of helical gear tooth is given by
FS=[σe]*b*π*mn*YV=(1.75*100)*430*π*18*0.4205=1789348.28N (or)FS= [σb]*b* π* mn* YV = 3500*43* π*1.8*0.4205=3578690.65N
Since, FS> FD, Our design is safe.
Considering the above conditions, Buckingham derived an equation to find out the maximum load acting on the gear tooth which is given by
FD= FT + FI, Where FD = Maximum Dynamic Load. FT =Static load produced by power FI=Incremental load due to dynamic action
Incremental load depends on the pitch line velocity, face width of a gear tooth, gear materials, accuracy of cut and the tangential load and it is given byFI= [0.164Vm (cbcos2β+ FT) cosβ] / [0.164Vm+1.485√ cbcos2β+ FT], Where Vm is the pitch line velocity in m/s, b is the face width of the gear tooth in mm, c is the dynamic factor or deformation factor in N/mm.Deformation factor c can also be selected
Here c = 11860*e, c= 11860*0.026=308.36 N/mm.FT=137572.60N, Vm=65.42m/s=65.42*103 mm/s, b=38.16 cm=381.6mm,β=25°
FD= FT + FI= FT + [0.164Vm (cbcos2β+ FT) cosβ]/[0.164Vm+1.485√cbcos2β+FT]=137572.60+0.164*65.42(308.36*430*cos225+137572.60) cos25][0.164*65.42+(1.485√308.36*430*cos225+137572.60)]=137572.60+[2396542.95/747.98]
=388329.19 N
Since FS > FD, our design is safe.
Based on Hertz theory of contact stresses, Buckingham derived an equation for wear strength of gear tooth which is given by FW= (D1*b*Q*KW)/cos2β, Where, FW is the max or limiting load for wear in N.D1 is the pitch circle diameter of the pinion (mm), b is the face width of the pinion(mm), Q is the ratio factor.
Q=Ratio factor=2i/(i+1)=1.75,KW=2.553N/mm2
D1=357.4mm; b=430mm,
FW =(D1*b*Q*KW)/cos3β
= (357.4*430*1.75*2.553)/cos225 =900086.75N
FD=140776.62N. Since FW> FDour design is safe.

Design of hole:

Design is based on the torque transmitted by the shaft. Torque transmitted T= (60*P)/2πN= (60*9000*103)/(2π*3500)=24555.3N-mm
We know that torque transmitted by the shaft T=(πd3*fs)/16=d=125 mm
Key dimensions:Width of key (w) =d/4=32 mm
Thickness of key (t) =d/6=18 mm.

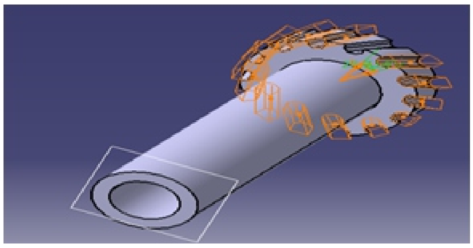

Fig 2: Pattern the sketch

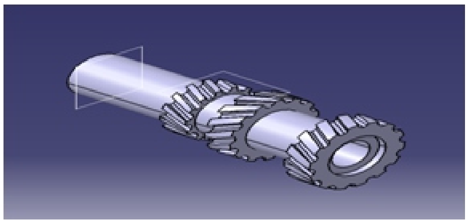

Fig 3: Complete design of a helical gear

IV.ANSYS RESULTS

Results for steel:

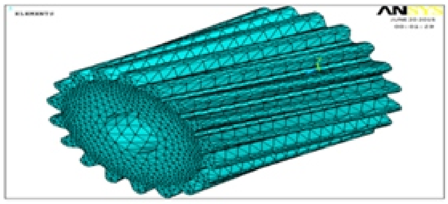

Fig 4: meshed gear model

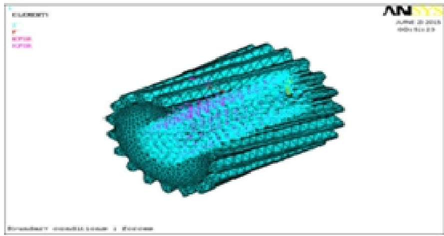

Fig 5: boundary conditions & Forces

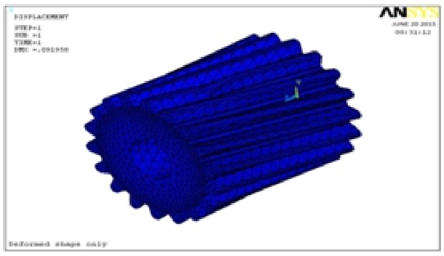

Fig 6: deformed helical gear with steel

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International Journal & Magazine of Engineering, Technology, Management and Research
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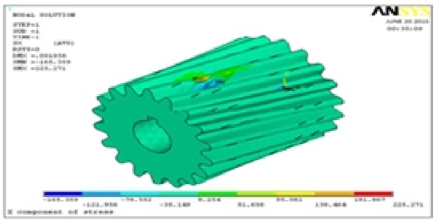

Fig 7: Stresses in X-direction of model of helical gear with steel as a material

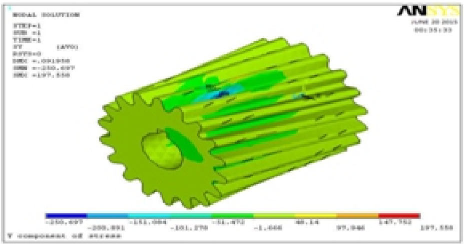

Fig 8: Stresses in Y-direction of model of helical gear with steel as a material

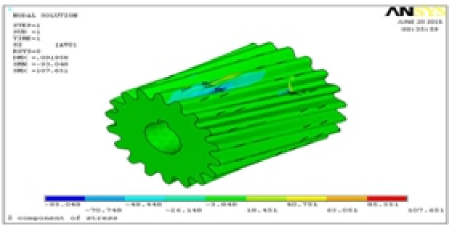

Fig 9: Stresses in Z-direction of model of helical gear with steel as a material

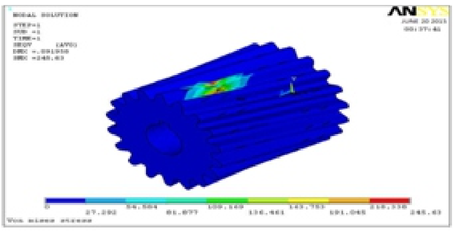

Fig 10: Von Misses stresses of model of helical gear with steel as a material

Results for ceramics:

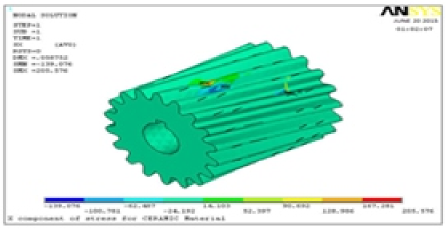

Fig 11: the stresses in X-direction of model of helical gear with ceramics as a material

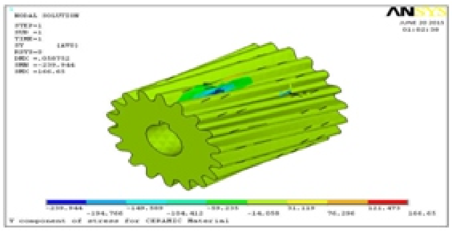

Fig 12: stresses in Y-direction of model of helical gear with ceramics as a material

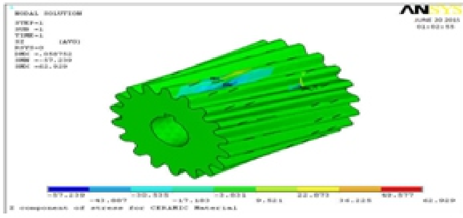

Fig 13: stresses in Z-direction of model of helical gear with ceramics as a material

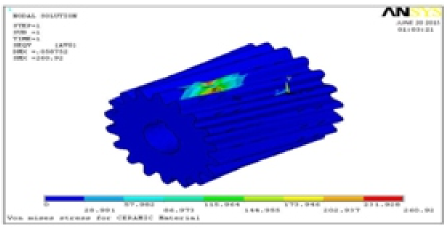

Fig 14: Von Misses stresses of model of helical gear with ceramics as a material

V.DISCUSSION OF RESULTS:

Results for steel as material:

The following table shows the comparison between theoretical results & experimental results.

TABLE 1

PARAMETER	DESIGN STRESSES	INDUCED STRESSES
BENDING STRESS	400 N/mm^2	228.35 N/mm^2
COMPRESSIVE STRESS	1100 N/mm^2	178.59 N/mm^2

ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

TABLE 2:

PARAMETER	THEORETICAL RESULTS	ANSYS RESULTS
DEFLECTION		0.091958 mm
BENDING STRESS	228.35 N/mm^2	225.271 N/mm^2
COMPRESSIVE STRESS	178.59 N/mm^2	165.359 N/mm^2
VON MISES STRESSES	σ_{yield}=972 N/mm^2	245.63N/mm^2

From the table 1 we observe that the bending & compressive stresses obtained practically from the ANSYS are much lower than those of the results obtained theoretically. Thus the design is safe from the structural point of view.From the table 2 we observe that the induced bending & compressive stresses are much lower than the design stresses. Thus the design is safe from the structural point of view.The maximum deflection is found to be 0.091958mm which is well within the permissible limits. Thus the design is safe based on rigidity point of view.The induced von mises stresses with magnitude of 245.63N/mm2 are much lower than the yield stress i.e. 972 N/mm2 according to the manufacturer's specifications.

Results for ceramics [98% al2o3]:

The following table shows the comparison between theoretical results & experimental results.

TABLE 3

PARAMETER	DESIGN STRESSES	INDUCED STRESSES
BENDING STRESS	350 N/mm^2	220.35 N/mm^2
COMPRESSIVE STRESS	2500 N/mm^2	150.30 N/mm^2

TABLE 4

PARAMETER	THEORETICAL RESULTS	ANSYS RESULTS
DEFLECTION		0.058752 mm
BENDING STRESS	228.35 N/mm^2	225.271 N/mm^2
COMPRESSIVE STRESS	178.59 N/mm^2	165.359 N/mm^2
VON MISES STRESSES	σ_{yield}=972 N/mm^2	245.63N/mm^2

From the table 3 we observe that the bending & compressive stresses obtained practically from the ANSYS are much lower than those of the results obtained theoretically. Thus the design is safe from the structural point of view.From the table 4 we observe that the induced bending & compressive stresses obtained are much lower than those of the design stresses. Thus the design is safe from the structural point of view.The maximum deflection is found to be 0.058752mm which is well within the permissible limits. Thus the design is safe based on rigidity point of view.The induced von mises stresses with magnitude of 260.92/mm2 is far below the yield stress i.e. 2000 N/mm2 according to the manufacturer's specifications.Thus the helical gear parameters that constitute the design are in turn safe based on the strength and rigidity.

Comparison of results:
Table 5: Comparison of Results:

PARAMETERS	STEEL	CERAMICS
DEFLECTIONS	0.091958 mm	0.058752 mm
BENDING STRESS	225.271 N/mm^2	205.576 N/mm^2
COMPRESSIVE STRESS	165.359 N/mm^2	139.076 N/mm^2

The results obtained above are less than material properties as mentioned before. Hence the design is safe and optimum.

ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

VI.CONCLUSION & FUTURE SCOPE:

Bending and compressive stresses were obtained theoretically & by using Ansys software for both ceramic& steel. From the table 5 it is observed that bending and compressive stresses of ceramics are less than that of the steel. Weight reduction is a very important criterion in the marine gears. Hence ceramic material is preferred.In the present investigation of static analysis, a high speed helical gear, is analyzed by FEM package ANSYS. As a future work, modal analysis and harmonic analysis of the gear can be performed to find out the mode shapes and natural frequencies of the gear.

REFERENCES:

1Thirupathi R. Chandrupatla& Ashok D.Belegundu ., introduction to finite eleement in engineering, pearson ,2003

2.Joseph Shigley, Charles Mischike ., Mechanical Engineering Design, tmh,2003

3.Maithra ., handbook of gear design,2000

4.V.B.Bhandari., design of machine elements,tmh,2003

5.R.S.Khurmi ., machine design, schand,2005

6.Darle w dudley.,hand book of practical gear design,1954

7.Alec strokes., high performance gear design,1970

8.Khurmi, R.S, Theory of Machines, S.CHAND

9.Schunck, Richard, "Minimizing gearbox noise inside and outside the box.", Motion System Design.

10.Doughtie and Vallance give the following information on helical gear speeds: "Pitch-line speeds of 4,000 to 7,000 fpm [20 to 36 m/s] are common with automobile and turbine gears, and speeds of 12,000 fpm [61 m/s] have been successfully used." – p.281.

11.McGraw Hill Encyclopedia of Science and Technology, "Gear", p. 742.

12.Canfield, Stephen (1997), "Gear Types", Dynamics of Machinery, Tennessee Tech University, Department of Mechanical Engineering, ME 362 lecture notes.

13.Hilbert, David; Cohn-Vossen, Stephan (1952), Geometry and the Imagination (2nd ed.), New York: Chelsea, pp. 287,ISBN 978-0-8284-1087-8.

14.McGraw Hill Encyclopedia of Science and Technology, "Gear, p. 743.

15.VallanceDoughtie, p. 287.

16.VallanceDoughtie, pp. 280, 296.

17.Doughtie and Vallance, p. 290; McGraw Hill Encyclopedia of Science and Technology, "Gear", p. 743.

18.McGraw Hill Encyclopedia of Science and Technology, "Gear", p. 744.

19.Kravchenko A.I., Bovda A.M. Gear with magnetic couple. Pat. of Ukraine N. 56700 – Bul. N. 2, 2011 – F16H 49/00.

20.ISO/DIS 21771:2007 : "Gears – Cylindrical Involute Gears and Gear Pairs – Concepts and Geometry", International Organization for Standardization, (2007)

21.Hogan, C Michael; Latshaw, Gary L The Relationship Between Highway Planning and Urban Noise , Proceedings of the ASCE, Urban Transportation Division Specialty Conference by the American Society of Civil Engineers, Urban Transportation Division, 21 to 23 May 1973, Chicago, Illinois