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Effect of Bending Stress on Steel Alloy Of Helical Gear For High Speed Applications

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ABSTRACT ***:*** *Gears are one of the most critical components in mechanical power transmission system and they are advantage over belt, wire, rope and chain drives. Gears are most commonly used for power transmission in all the modern devices. They have been used extensively in the high-speed marine engines. Gear design has evolved to a high degree of perfection; the constant pressure to build less expensive, quieter running, lighter weight, reliable, less cost and more powerful machinery has lead to steady change in gear design. Today, the most significant new gear developments are in the area of materials. The objective of the present work is to focus on investigating the effects of face width, gear ratio, normal module, speed and helix angle on bending stress of steel alloy (40 Ni2 Cr1 Mo28) helical gear for marine applications.*

Keywords - *Buckingham equation, Bending stress, Helical gear design, Lewis equation, Optimization*

I. INTRODUCTION

Gears are rotational elements, used to transmit the rate of changing the rotation from one shaft to another. Gear transmissions are widely used in various industries and their efficiency and reliability are critical in the final product performance evaluation. Today, the most significant new gear [1] developments are in the area of materials. Modern metallurgy has greatly increased the useful life of industrial and automotive gearing to new levels of accuracy, reliability and quiet operation. It is possible that gears will predominate as the most effective means of transmitting power in future machines due to their high degree of reliability and compactness. In addition, the rapid shift in the industry from heavy industries such as shipbuilding to industries such as automobile manufacture and office automation tools will necessitate a refined application of gear technology. In the present era of sophisticated technology[5] 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. Helical gears are the modified forms of spur gears, in which all the teeth are cut at a constant angle, known as helix angle, to the axis of the gear, where as in spur gear, teeth are cut parallel to the axis. The following are the requirements that must be met in the design of gear drive, the gear teeth should have sufficient strength, so that they will not fail under static and dynamic loading during normal running conditions. The gear teeth should have clear characteristics so that their life is satisfactory, the use of space and material should be economical. The alignment of the gears and deflections of the shafts must be considered, because they affect the Performance of the gears. The lubrications of the gears must be satisfactory.

Marine engines are among heavy-duty machineries, which need to be taken care of in the best way during prototype development stages. These engines are operated at very high speeds [6] which induce large stresses and deflections in the gears as well as in other rotating components. For the safe functioning of the engine, these stresses and deflections have to be minimized. In this study parametric approach analysis on a high speed helical gear used in marine engines using steel alloy material is undertaken. The results obtained in theoretical and a conclusion has been 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. Gears have been manufactured for a number of years with extensive ongoing research [4] related to their efficiency, operational quality and durability. They are relatively complex and there is a number of design parameters involved in gear design. The design of gears requires an iterative approach [2] to optimize design parameters [3] which govern both the kinematics as well as the strength performance. Due to the complex combination of these parameters, conventional design office practice tends to become complicated and time consuming. It involves selection of appropriate information from a large amount of standard data [9] available in engineering catalogues [2] and design handbooks [3],[11].While the knowledge in gearing design is vast, there is still an acute paucity of research on comparative analysis between various standards and engineering practices.

II. DESIGN METHODOLOGY

The helical gear is design based on AGMA Procedure:
According to Lewis equation for beam strength of helical gear tooth;

$$F_b = [\sigma_b] \cdot b \cdot \pi m_n \cdot y_v \tag{1}$$

Number of teeth; $Z_v = (Z/\cos^3 \beta)$ (2)

Design tooth load by;

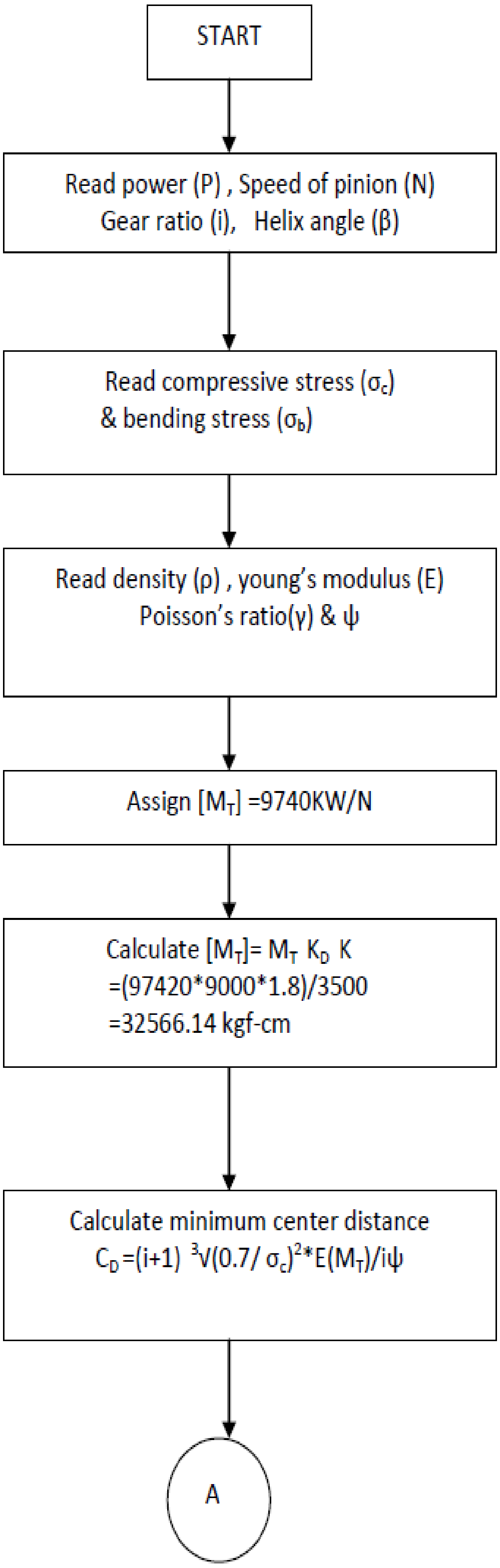
$$F_{D,} = F_t.K_s.C_v = (F_t.K_s.C_v/v) \tag{3}$$

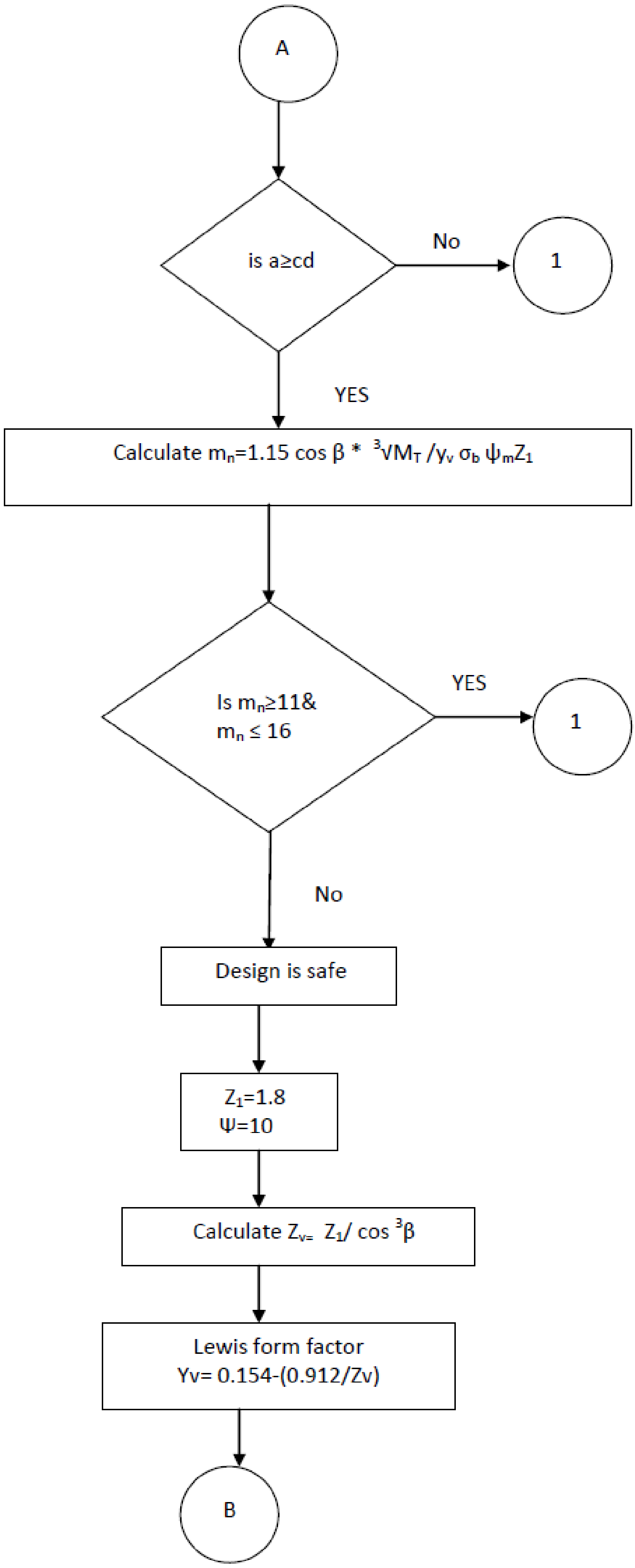
The dynamic load by Buckingham equation;

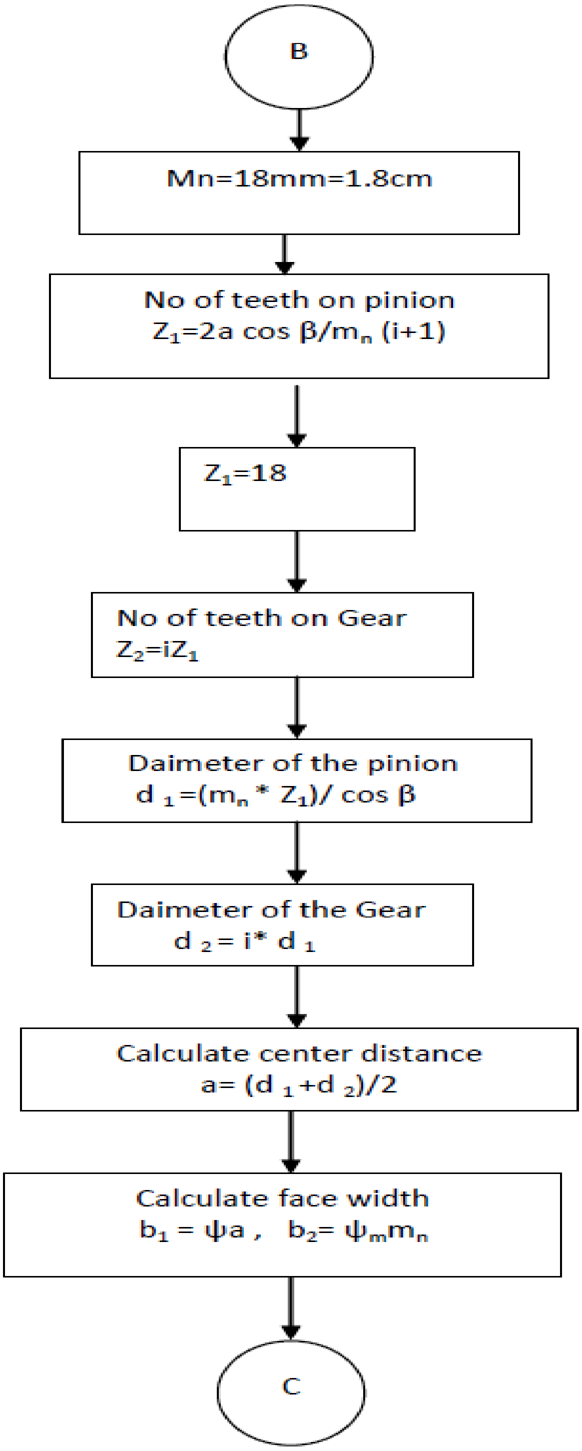
$$F_d = F_t + \frac{21v\left(Cb\cos^2\beta + F_t\right)\cos\beta}{21v + \sqrt{Cb\cos^2\beta + F_t}} \tag{4}$$

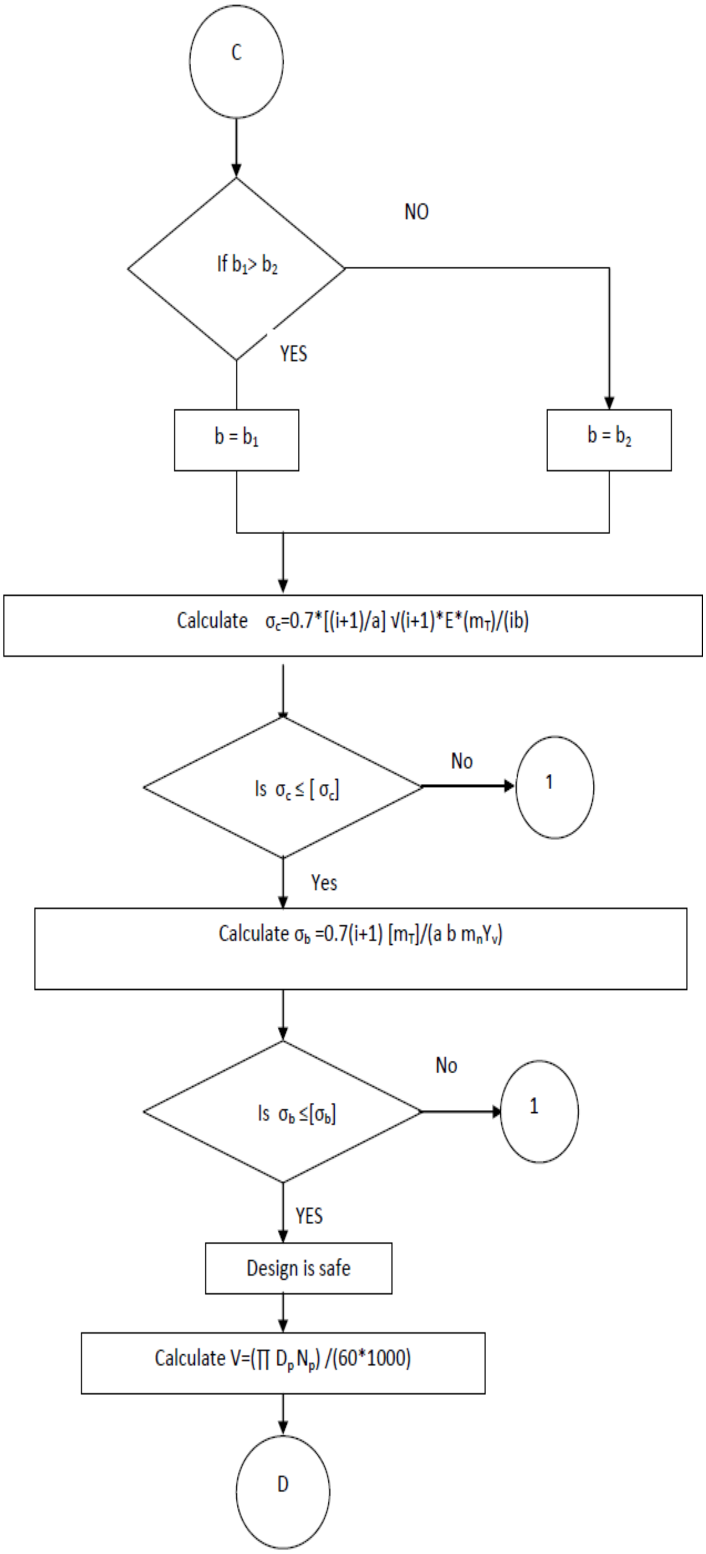
The flow chart is showing the process of helical gear design;

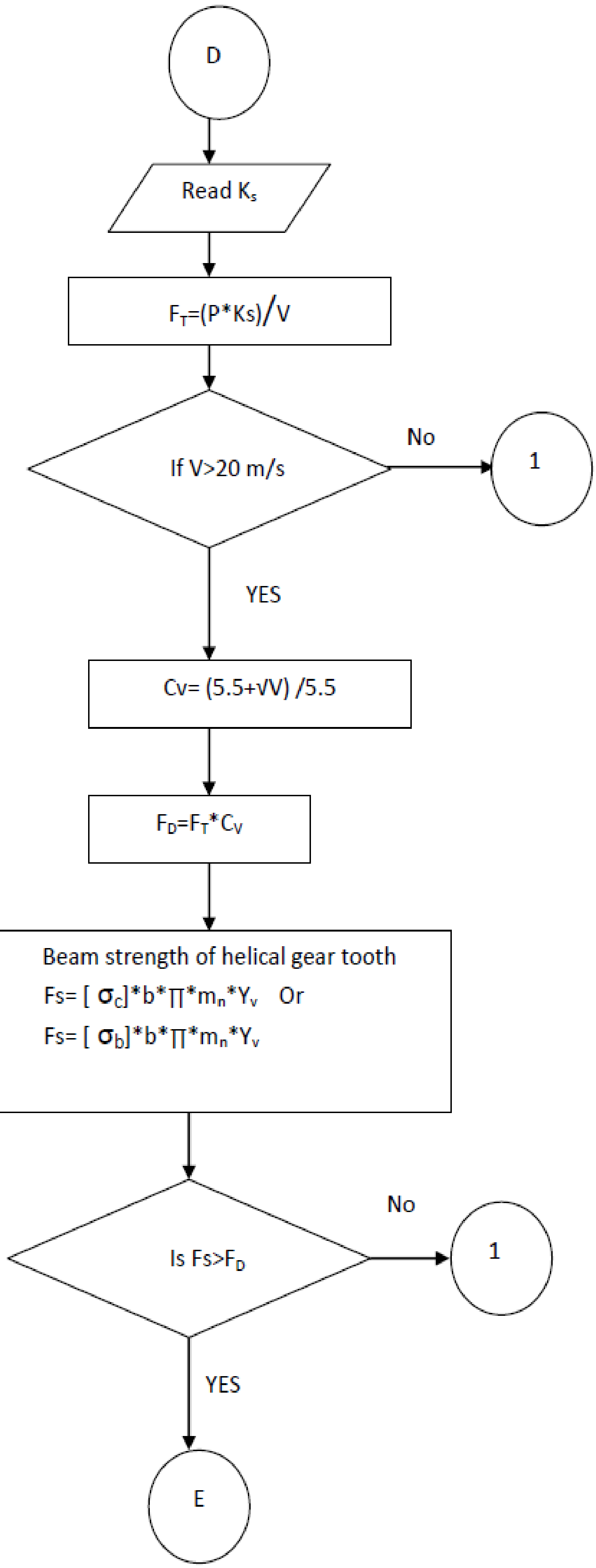
FLOW CHART


START
Read power (P) , Speed of pinion (N)
Gear ratio (i), Helix angle (β)
Read compressive stress (σ_c)
& bending stress (σ_b)
Read density (ρ) , young's modulus (E)
Poisson's ratio(γ) & ψ
Assign [M_T] =9740KW/N
Calculate [M_T]= M_T K_D K
=(97420*9000*1.8)/3500
=32566.14 kgf-cm
Calculate minimum center distance
C_D=(i+1) $^3\sqrt{}$(0.7/ σ_c)^2*E(M_T)/iψ
A


A
is a≥cd
No
1
YES
Calculate m_n=1.15 cos β * $^3\sqrt{}M_T$ /y_v σ_b $\psi_m Z_1$
Is m_n≥11&
$m_n \leq 16$
YES
1
No
Design is safe
Z_1=1.8
Ψ=10
Calculate $Z_{v=}$ Z_1/ cos $^3\beta$
Lewis form factor
Yv= 0.154-(0.912/Zv)
B


B
Mn=18mm=1.8cm
No of teeth on pinion
Z_1=2a cos β/m_n (i+1)
Z_1=18
No of teeth on Gear
Z_2=iZ_1
Daimeter of the pinion
d_1=(m_n * Z_1)/ cos β
Daimeter of the Gear
d_2= i* d_1
Calculate center distance
a= (d_1+d_2)/2
Calculate face width
b_1 = ψa , b_2= $ψ_m m_n$
C


C
If $b_1 > b_2$
NO
YES
$b = b_1$
$b = b_2$
Calculate $\sigma_c=0.7*[(i+1)/a]\sqrt{(i+1)*E*(m_T)/(ib)}$
Is $\sigma_c \le [\sigma_c]$
No
1
Yes
Calculate $\sigma_b = 0.7(i+1)[m_T]/(a\ b\ m_n Y_v)$
Is $\sigma_b \le [\sigma_b]$
No
1
YES
Design is safe
Calculate $V=(\prod D_p N_p)/(60*1000)$
D


D
Read K_s
F_T=(P*Ks)/V
If V>20 m/s
No
1
YES
Cv= (5.5+√V) /5.5
F_D=F_T*C_V
Beam strength of helical gear tooth
Fs= [σ_c]*b*∏*m_n*Y_v Or
Fs= [σ_b]*b*∏*m_n*Y_v
Is Fs>F_D
No
1
YES
E

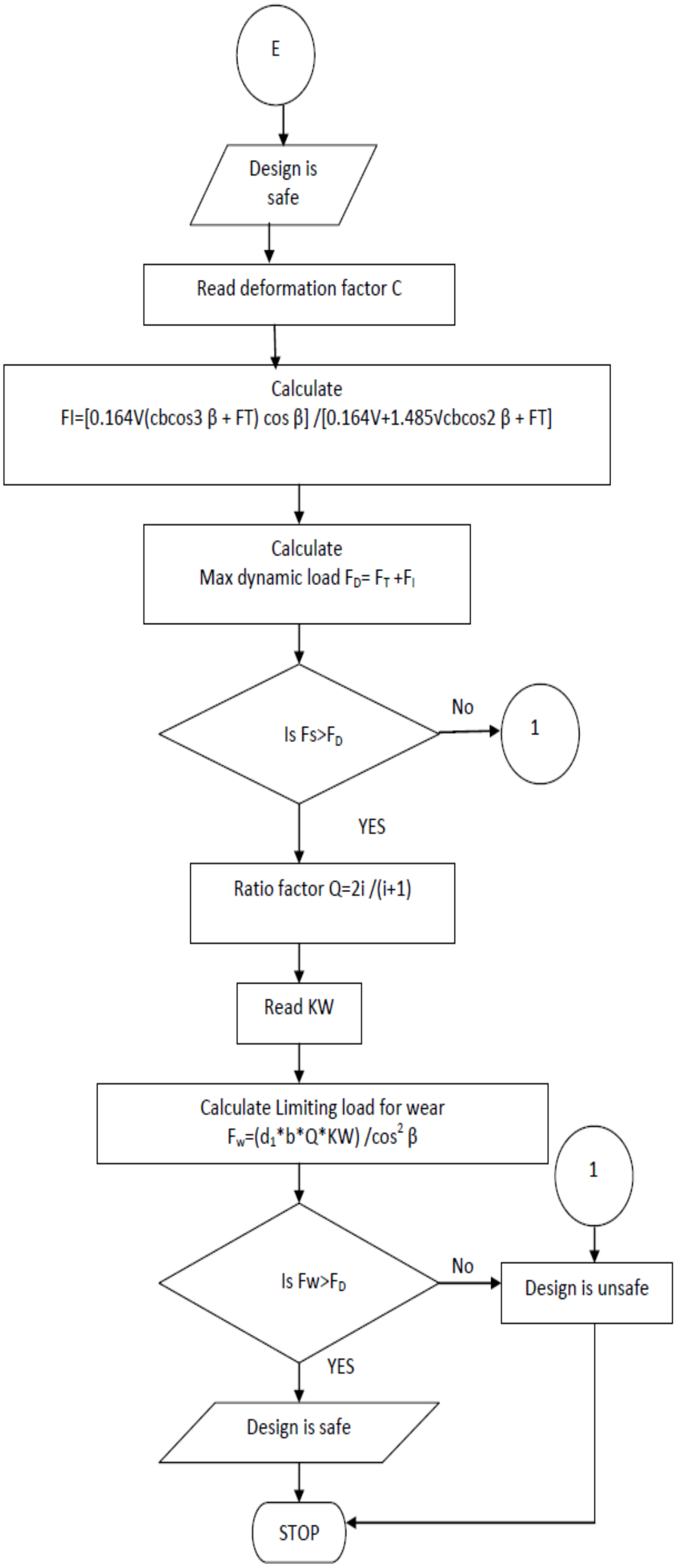


Fig.1. Flow chart for the process of helical gear design.

III. RESULTS AND DISCUSSIONS

To arrive at optimum values of bending stress to get low cost manufacturing for Steel alloy (40 Ni2 Cr1 Mo28) gear have been carried out.

3.1 The effect of helix angle, module, gear ratio, face width on bending stress for steel alloy:

The variation of bending stress for different input variables are shown in figs. 1(a) – (d). The speed is kept constant. The fig 1(a) shows the relation between bending stress and gear ratio. The helix angle, face width, and module are kept constant. When the gear ratio is increased from 4 to 8, the corresponding bending stress remained constant. The fig 1(b) shows the relation between bending stress and Face width. The helix angles, gear ratio, module are kept constant. When face width is increased from 41 to 49, the corresponding bending stress decrease linearly from 580kgf/cm^2 to 485kgf/cm^2. The fig 1(c) shows the relation between bending stress and helix angle. The face width, gear ratio, corresponding to optimum value obtained earlier and module is kept constant. When helix angle is increased from 15^o to 35^o, the corresponding bending stress was observed to decrease from 580 kgf/cm^2 to 492 kgf/cm^2. The fig 1(d) shows the relation between bending stress and module. The value of face width, gear ratio, and helix angle for maximum bending stress are kept constant. When module is increased from 16mm to 24mm, the corresponding bending stress was observed to decrease from 580 kgf/cm^2 to 258 kgf/cm^2. Thus the maximum bending stress 580 kgf/cm^2 is obtained for input parameters viz. gear ratio (i) = 6, face width (b) = 41, helix angle (β) = 15^0 and Module (Mn) =16.

3.2 Optimum parameters for maximum bending stress:

The effect of gear ratio, face width, helix angle, and module for optimum bending stress for steel alloy is carried out. Keeping the helix angle, face width, speed and module are as constant. When the gear ratio is increased, the corresponding bending stress remained constant. Keeping the helix angle, gear ratio, speed, module constant and for variation of face width, the bending stress decreases linearly. In the next step, the face width 41cm, corresponding to maximum bending stress is fixed and the gear ratio, speed and module are kept constant and helix angle is increased, and the corresponding bending stress was observed to decrease. The helix angle 15^0, corresponding to maximum bending stress is taken for further optimization. Now the face width, gear ratio, speeds, helix angle are kept constant and module is varied. The corresponding bending stress was observed to decrease. The module 16mm, corresponding to maximum bending stress is taken as constant.

IV. CONCLUSION

The study helps to make out the effect of bending stress on the optimum design of helical gears for high speed applications. The analysis yielded bending stress of 580 kgf/cm^2 for gear ratio (i) = 6, face width (b) = 41, helix angle (β) = 150 and Module (Mn) =16. The helical gear parameters that constitute the design are found to be safe from strength and rigidity point of view. Hence Steel alloy (40 Ni2 Cr1 Mo28) is best suited for high speed applications.

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VI. NOMENCLATURE:

σ_b = Bending stress in Kgf/cm²

$[\sigma_{b}]$ = Design Bending stress in Kgf/cm²
E= Young's modulus in Kgf/cm²
[Mt] = design torque in Kg-cm
β = Helix angle in degrees
F_d = Dynamic tooth load of the gear Kgf
F_D = Design tooth load kgf
m_n = Normal Module mm
Y_v = Lewis Form factor
b =Face width in mm
N = speed in rpm
i = Gear ratio

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