ISSN: 2277-3754
ISO 9001:2008 Certified
International Journal of Engineering and Innovative Technology (IJEIT)
Volume 4, Issue 12, June 2015

Design of Dual Mode Bicycle by Using Gear Box

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Abstract*— conventionally the bi-cycle was primarily used to commute, i.e. to move from one place to other, whereas now a day's cycling has taken up additional use in exercising, and sport. Conventional bi-cycles employ the chain drive to transmit power from the pedal arrangement to the wheel .Chain focused bi-cycle requires accurate mounting & alignment for proper working. Least miss-alignment will result in chain dropping. More over the drive is in-efficient hence the need of shaft driven bicycles have can be introduced due to highly developed gear manufacturing technology. The 'chainless' drive system helps to transfer energy from the pedals to the rear wheel. It is striking in look compare with chain focused bicycle having more efficiency. This Project introduces Design Development and Analysis of dual mode bicycle with shaft drive that shall serve both purpose of commute and exercise. Design and development of bi-cycle using 2-D cad , 3-D modelling Unigraphix, Design of critical components using ANSYS software. Test and trial to differentiate the findings of exercising and trek mode, Calculation of energy spent in exercise mode to calorie burn, Calculation of energy saved in travel mode. Mathematical model of pedal arrangement system, spiral bevel gear selection for finest power transmission capacity. Development of mathematical model of system of forces, derivation and resolution of system forces by drawing free body diagram of linkage , determination of forces and utilizing system of forces to determine the linkage dimensions of most important parts of drive. 3-D modeling will be done using Unigraphix Nx-8.0 and CAE of critical component and meshing using ANSYS Work-bench 14.5. The experimental validation is done for the part of reduced pedal effort developed by the modified mechanism in comparison to the conventional chain arrangement by theoretical derivation.

Keywords— **Shaft drive, dual mode, 2-D CAD, 3-D Modelling, ANSYS software**

I. INTRODUCTION

The first shaft drives for cycles appear to have been invented independently in 1890 in United States & England. If bevel wheels could be accurately and cheaply cut by machinery, it is possible that gear of this description might supplant to a great extent the chain drive. The shaft is connected between the pair of spiral bevel gears. The main application of the spiral bevel gear is in a vehicle differential, where the direction of drive from the drive shaft must be turned 90 degrees to drive the wheels. The helical design produces less vibration & noise than conventional straight cut or spur cut gear with straight teeth. The shaft drive bicycle gives more efficiency.

There is limitation on the maximum speed attained as the maximum speed attainable cannot exceed 1100 rpm. More over the application of chain drive leads to underutilization of human effort due to the fact the maximum transmission of bi-cycle chain remains below 70 per cent due to polygon effect in chain sprocket drives. Thus there is a need to replace the conventional chain drive using the spiral bevel gear arrangement. Motion is transmitted from pedal to wheel through four speed inline gear box with sliding mesh gear. Gears can be easily shifted by using thumb shifter near brake lever. This Project introduces Design Development and Analysis of dual mode bicycle with shaft drive that shall serve both purpose of commute and exercise. Mode 1: Exercising mode (Gear 1 & 2) designed to give minimum distance travelled in maximum pedalling. Mode 2: Commutation/travel mode (Gear 3 & 4) designed to give maximum distance travelled in minimum pedalling. It include design of kinematic linkage for pedal arrangement, gear box to produce a driving force to carry the given system load.2-d drawing preparation of linkage mechanism by 'kinematic overlay method 'using Auto-Cad. Mathematical model is developed for system of forces. Determination of forces and utilizing system of forces to determine the gear dimensions for operation in dual mode i.e. the exercise mode and the travel mode. Mechanical design of critical components is done by using theoretical theories of failure. After selection of appropriate materials 3-D modelling of set-up using unigraphix Nx-8.0 is done. CAE of critical component Such as Bi-cycle frame, Seat system, Pedal linkage, Pedal shaft, Drive shaft etc. and meshing using ANSYS software is completed. Experimental validation of the transmission efficiency for the drive is done by using brake Dynamometer test and optimization of the effort application in both mode of bicycle. The experimental validation is done for the part of the pedal force developed by the modified Mechanism in comparison to the conventional chain arrangement by theoretical derivation.

1. Dual mode bicycle:

Fig.1 Dual mode bicycle

The layout of components should be such that easy servicing is possible especially those components which required frequent servicing can be easily disassembled. Mechanical design phase is very important from the view of

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ISO 9001:2008 Certified
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designer. As whole success of the project depends on the correct deign analysis of the problem. Designer should have adequate knowledge about physical properties of material, loads stresses, deformation, and failure. Theories and wear analysis, Identification of the external and internal forces acting on the machine parts. Motion is transmitted from pedal to wheel through 4-speed inline gear box with sliding mesh gear. When gear is mesh in 1 and 2 gears then bicycle is in exercising mode, where as in 3 and 4 then bicycle is in travelling mode. Simple technique is used to shift sliding mesh four speed gear box to achieve required Mode of transmission. Thumb shifter near break lever is used to shift sliding mesh four speed gearboxes.

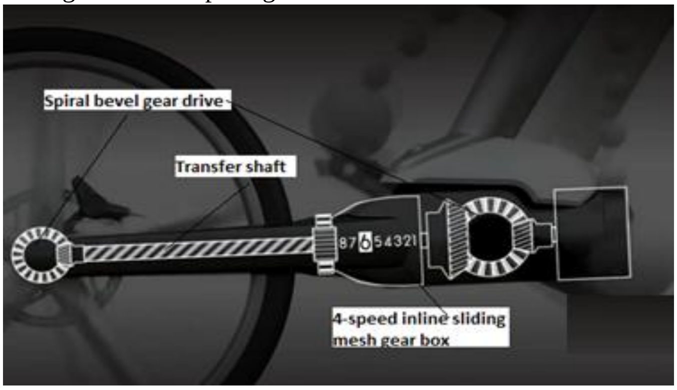

Fig.2 Four Speed Inline Gear box

A. Salient Features

1. Spiral bevel gear drive from pedal to wheel ---shaft drive
2. Dual mode:

Mode 1: Exercising mode (Gear 1 & 2) designed such that heart rate is maintained between 125 to 140 bpm, for maximum calorie burn.

Mode 2: Commutation/travel mode (Gear 3 & 4) designed to give maximum distance travelled in minimum pedalling

3. Easy shift sliding mesh four speed gearbox, simple logic selection (either / or technique using thumb shifter near brake lever).

II. LITERATURE SURVEY

A shaft driven bicycle is a bicycle that uses a drive shaft instead of a chain to transmit power from the pedals to the wheel. The drive shafts are carries of torque. The steel drive shaft satisfies three design specifications such as torque transmission capability, buckling torque capability & bending natural frequency. The both end of the shaft are fitted with the bevel pinion, the bevel pinion engaged with the crown & power is transmitted to the rear wheel through the propeller shaft & gear box. The design of suitable propeller shaft and replacement of chain drive smoothly to transmit power from the pedal to the wheel without slip. Shaft drive increases the power transmission efficiency.

III. DESIGN METHODOLOGY

In our attempt to design a special purpose machine we have adopted a careful approach, the total design work has been divided into two parts mainly;

1. System design
2. Mechanical design

System design mainly concerns with the various physical constraints and ergonomics space requirements, arrangement of various components on the main frame of machine no of controls position of these controls ease of maintenance scope of further improvement ; weight of m/c from ground etc.

In Mechanical design the component in two categories.

1. Design parts
2. Parts to be purchased.

For design parts detail design is done and dimensions thus obtained are compared to next highest dimension which are readily available in market this simplifies the assembly as well as post production servicing work. The various tolerance on work are specified in the manufacturing drawings the process charts are prepared & passed on to the manufacturing stage .The parts are to be purchased directly are specified & selected from standard catalogues.

Mechanical design phase is very important from the view of designer as whole success of the project depends on the correct deign analysis of the problem. Many preliminary alternatives are eliminated during this phase. Designer should have adequate knowledge above physical properties of material, loads stresses, deformation, and failure. Theories and wear analysis, He should identify the external and internal forces acting on the machine parts.

These forces may be classified as;

a) Dead weight forces
b) Friction forces
c) Inertia forces
d) Centrifugal forces
e) Forces generated during power transmission etc.

Designer should estimate these forces very accurately by using design equations .If he does not have sufficient information to estimate them he should make certain practical assumptions based on similar conditions which will almost satisfy the functional needs. Assumptions must always be on the safer side. Selection of factors of safety to find working or design stress is another important step in design of working dimensions of machine elements. The correction in the theoretical stress values are to be made according in the kind of loads, shape of parts & service requirements. Selection of material should be made according to the condition of loading shapes of products environment conditions & desirable properties of material. Provision should be made to minimize nearly adopting proper lubrications methods.

a) Design of Link

Material Selection:-

Ref: - PSG (1.10 & 1.12) + (1.17)

Designation	Ultimate Tensile Strength N/mm^2	Yield Strength N/mm^2

EN9	650	480

Table 1 ASME code for design of link

Cross section of link may be determined by considering lever in bending

The linkage has a section of (25x10) mm

Let; t=thickness of link

B= Width of link

Bending moment;

Section modules; $Z= 1/6\ t\ B^2$

$Fb=m/z = PL/1/6\ t\ B^2$

$Fb=m/z = 6PL/tB2$

Maximum effort applied by hand (P) = 200 N

$\Rightarrow F_b= 6 \frac{x\ 200\ x\ 120}{10\ x\ 25^2}$

$Fb = 23.02\ N/mm^2$

As $F_{b\ act} < F_{ball}$

Thus selecting cross section of link is $(25X10)\ mm^2$

b) Design of Pedal Shaft

Material Selection:

Ref: - PSG (1.10 & 1.12) + (1.17)

Designation	Ultimate Tensile Strength N/mm^2	Yield Strength N/mm^2
EN24	800	680

Table 2 ASME code for design of shaft

Since the loads on most shafts in connected machinery are not constant, it is necessary to make proper allowance for the harmful effects of load fluctuations. According to ASME code permissible values of shear stress may be calculated form various relations.

$fs\ max = 0.18\ x\ 80$

$fs\ max = 144\ N/mm^2$ OR

$fs\ max = 0.3\ fyt$

$fs\ max =0.3\ x\ 680 = 204\ N/\ mm^2$

Considering minimum of the above values

$fs\ max = 144\ N/mm^2$

Shaft is provided with key way; this will reduce its strength. Hence reducing above value of allowable stress by 25%

$fs\ max = 108\ N/mm^2$

This is the allowable value of shear stress that can be induced in the shaft material for safe operation.

To calculate pedal shaft torque, Note that torque at the pedal shaft is 200 x120 =2400N-mm

$\Rightarrow$T design = 2.4N-m

Check for torsional shear failure of shaft. Assuming minimum section diameter on input shaft = 14mm, Note that this dimension is the smallest section of the main Shaft where the lode plate is mounted, hence, (manufacturing consideration)

$\Rightarrow d = 14\ mm$

$Td = \pi/16\ x\ fs\ act\ x\ d^3$

$\Rightarrow fs\ act = \frac{16\ \ x\ Td}{\pi\ x\ d\ 3}$

$fsact = \frac{16x2.4x10^3}{\pi\ x\ d^3}$

$\Rightarrow fs\ act = 4.45\ N/mm^2$

As fs act < fs all

$\Rightarrow$ I/P shaft is safe under torsional load

c) Design of Gear Box

In our attempt to design a **GEAR BOX** we have adopted a very a very careful approach, For design parts detail design is done and dimensions thus obtained are compared to next highest dimension which are readily available in market this simplifies the assembly as well as post production servicing work. The various tolerances on work pieces are specified in the manufacturing drawings. The process charts are prepared & passed on to the manufacturing stage .The parts are to be purchased directly are specified &selected from standard catalogues.

$\Rightarrow$ T design = 1.25 x 2.35 = 2.93 = 3 N-m

Check for torsional shear failure of shaft

Assuming minimum section diameter on input shaft = 16 mm as the pulley is to be mounted on shaft and minimum bore size that can be machined with dimensional tolerances is 16mm

$\Rightarrow d = 16\ mm$

$Td = \Pi/16\ x\ fs_{act}\ x\ d^3$

$\Rightarrow fs_{act} = \frac{16\ \ x\ Td}{\Pi\ \ x\ d^3}$

$fs_{act} = \frac{16\ x\ 3\ x\ 10^3}{\Pi\ x\ (16)^3}$

$\Rightarrow fs_{act} = 3.73\ N/mm^2$

As $fs_{act} < fs_{all}$

$\Rightarrow$ I/P shaft is safe under torsional load

Design of spline

Material of Shaft EN24

$Sult = 800\ N/mm^2$

$Sylt = 680N/mm^2$

$\Rightarrow fs_{all} = 108\ N/mm^2$

D = Major diameter of splines = 29

d = Minor diameter of splines = 20

L =Length of hub = 30

n= No. of splines =10

Torque transmission capacity of spines is given by;

$T_{capacity} = (1/8)p_mLn(D^2-d^2)$

Where; p_m = permissible pressure in splines =6.5 N/ mm^2

$T = (1/8)\ x\ 6.5\ x\ 30\ x\ 10\ x\ (29^2-20^2) =107.493\ x\ 10^3$N-mm

As; $T_{capacity}$ > T design (3 N-m)

Spline shaft is safe.

Design of output spline shaft:

Since the loads on most shafts in connected machinery are not constant, it is necessary to make proper allowance for the harmful effects of load fluctuations.

ISSN: 2277-3754
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According to ASME code permissible values of shear stress may be calculated from various relations.

$= 0.18 \times 80$

$= 144$ N/mm^2 OR

$fs_{max} = 0.3$ fyt

$= 0.3 \times 680 = 204$ N/mm^2

Considering minimum of the above values;

$\Rightarrow fs_{max} = 144$ N/mm^2

Shaft is provided with key way; this will reduce its strength. Hence reducing above value of allowable stress by 25%

$\Rightarrow fs_{max} = 108$ N/mm^2

This is the allowable value of shear stress that can be induced in the shaft material for safe operation.

To calculate worm wheel shaft torque

$$\text{POWER} = \frac{2\Pi NT}{60}$$

Motor is 50 watt power, run at 1000 rpm, connected to Spline shaft by open belt drive reduction ratio 1:5

$$\Rightarrow T = \frac{60 \times P}{2 \times \Pi \times N} = \frac{60 \times 50}{2 \times \Pi \times 1000}$$

$\Rightarrow T = 0.47$ N-m

Torque at input shaft = 0.47 x 5 =2.35 Nm

Considering 25% overload

$\Rightarrow$ T design = 1.25 x 2.35 = 2.93 = 3 N-m

Check for torsional shear failure of shaft.

Assuming minimum section diameter on input shaft = 16 mm as the pulley is to be mounted on shaft and minimum bore size that can be machined with dimensional tolerances is 16mm

□ d = 16 mm

$Td = \Pi/16 \times fs_{act} \times d^3$

$\Rightarrow fs_{act} = 16 \times Td / \Pi \times d^3$

$$fs_{act} = \frac{16 \times 3 \times 10^3}{\Pi \times (16)^3}$$

$\Rightarrow fs_{act} = 3.73$ N/mm^2

As $fs_{act} < fs_{all}$

$\Rightarrow$ I/P shaft is safe under torsional load

i) Design of Spur Gear Pair for first gear

Power = 01/15 HP = 50 watt

Speed = 200 rpm

b = 10 m

Tdesign = 3 NM

Sult pinion = Sult gear = 400 N/mm^2

Service factor (Cs) = 1.5

Gear pair--- Gear-1 = 11T, Gear-2 = 35T

dp = 16.5 ,T = T design = 3 N-m

$$\text{Now; } T = \frac{Pt \times dp}{2} \Rightarrow Pt = 363 \text{ N.}$$

$P_{eff} = Pt \times Cs / \underline{Cv}$

$\underline{P_{eff} = 363 \times 1.5}$

Neglecting effect of Cv as speed is very low

$P_{eff} = 545$ -------- (A)

Lewis Strength equation, WT = Sbym where;

$$Y = 0.484 - \frac{2.86}{Z}$$

$$\Rightarrow yp = 0.484 - \frac{2..86}{11} = 0.224$$

$\Rightarrow$ Syp = 89.6

Pinion and gear both are of same material, Syp = 89.6 N

$W_T = (Syp) \times b \times m$

$= 89.6 \times 10m \times m$

$W_T = 896m^2$---------- (B)

Equating equation (A) & (B)

$896m^2 = 545 \Rightarrow m=0.77$

Selecting standard module =1.5 mm

GEAR DATA

No. of teeth on gear on main shaft=11

No. of teeth gear on countershaft =35, Module = 1.5 mm

ii) Design of Spur Gear Pair for second gear

Gear pair--- Gear-1 = 17T, Gear-2 = 29T

$d_p = 25.5$, T = T design = 3 N-m

$$\text{Now; } T = \frac{Pt \times dp}{2} \Rightarrow Pt = 235\text{N.}$$

Peff = Pt xCs / $\underline{Cv}$

$\underline{\text{Peff} = 235 \times 105}$

Neglecting effect of Cv as speed is very low

Peff = 352 -------- (A)

Lewis Strength equation WT = Sbym where;

$$Y = 0.484 - \frac{2.86}{Z}$$

$$\Rightarrow yp = 0.484 - \frac{2..86}{17} = 0.315$$

$\Rightarrow$ Syp = 126.3

Pinion and gear both are of same material, Syp = 89.6 N

$W_T = (Syp) \times b \times m$

$= 126.3 \times 10m \times m$

$W_T = 1263m^2$---------- (B)

Equating equation (A) & (B)

$1263m^2 = 352$

$\Rightarrow m=0.52$

Selecting standard module =1.5 mm

GEAR DATA

No. of teeth on gear on main shaft=17

No. of teeth gear on countershaft =29, Module = 1.5 mm

iii) Design of Spur Gear Pair for third gear

Gear pair---Gear-1=22T, Gear-2= 26 T

Dp=32.96 mm, T = T design = 3 N-m

$$\text{Now; } T = \frac{Pt \times dp}{2} \Rightarrow Pt = 182 \text{ N.}$$

Peff = Pt xCs / $\underline{Cv}$

$\underline{\text{Peff} = 182 \times 1.5}$

Neglecting effect of Cv as speed is very low

Peff = 273 -------- (A)

Lewis Strength equation, WT = Sbym where;

$$Y = 0.484 - \frac{2.86}{Z}$$

$$\Rightarrow yp = 0.484 - \frac{2..86}{22} = 0.354$$

$\Rightarrow$ Syp = 141.6

Pinion and gear both are of same material, Syp = 89.6 N

$W_T = (Syp) \times b \times m$

$= 141.6 \times 10m \times m$

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W_T= 1416m^2---------- (B)
Equating equation (A) & (B)
1416m^2 =273 $\Rightarrow$m=0.43
selecting standard module =1.5 mm
G DATAEAR
No. of teeth on gear on main shaft=22
No. of teeth gear on countershaft =26, Module = 1.5 mm
iv) Design of Spur Gear Pair for forth gear
Gear pair---Gear-1=24T, Gear-2= 22 T
Dp=36 mm, T = T design = 3 N-m

$$\text{Now; } T = \frac{Pt \times dp}{2} \quad \Rightarrow Pt = 167 \text{ N.}$$

Peff = Pt xCs / Cv
Peff = 182 x 1.5
Neglecting effect of Cv as speed is very low
Peff = 250 -------- (A)
Lewis Strength equation, WT = Sbym where;

$$Y = 0.484 - \frac{2.86}{Z}$$

$$\Rightarrow yp = 0.484 - \frac{2..86}{24} = 0.364 \quad \Rightarrow Syp = 145.6$$

Pinion and gear both are of same material, Syp = 145.6 N
W_T = (Syp) x b x m
=145.6 x 10m x m
W_T= 1456m^2---------- (B)
Equating equation (A) & (B)
1456m^2 =250 $\Rightarrow$m=0.414
Selecting standard module =1.5 mm
GEAR DATA
No. of teeth on gear on main shaft=22
No. of teeth gear on countershaft =26, Module = 1.5 mm
d) Design of Spiral Bevel Gear----Theoretical method.
Material Selection.
Ref: - PSG (1.10 & 1.12) + (1.17)

Designation	Ultimate Tensile Strength N/mm^2	Yield Strength N/mm^2
EN24	800	680

Table 3 ASME code for design of spiral bevel gear

As Per ASME Code; fs_{max} = 108 N/mm^2
Check for torsional shear failure:-
Design torque = 0.29 x 2.78 = 0.81 Nm

$$T = \frac{\pi \times fs_{act}}{16} \times \frac{Do^4 - Di^4}{Do}$$

$$0.81 \times 10^3 = \frac{\pi \times fs_{act}}{16} \times \frac{24^4 - 15^4}{24}$$

$\Rightarrow$ fs_{act} = 0.35 N/mm^2
As; $fs_{act} < fs_{all}$
$\Rightarrow$ Gear is safe under torsional load

IV. ANALYSIS OF SPIRAL BEVEL GEAR

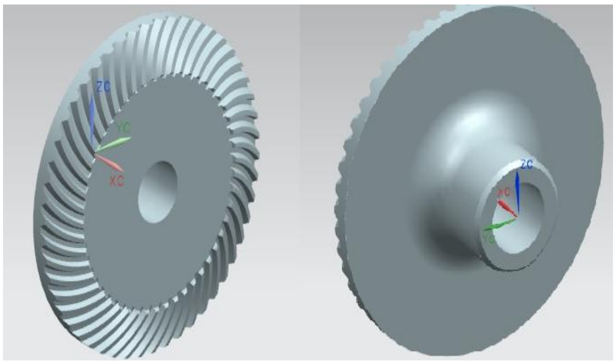
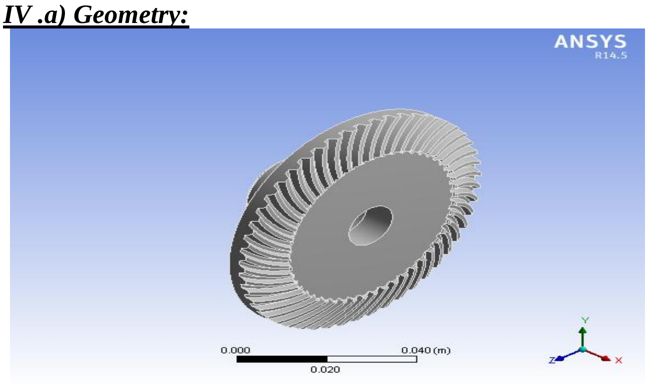
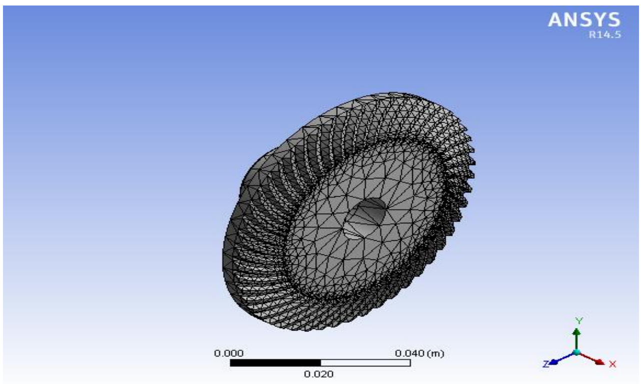


Fig. 3. Spiral Bevel Gear

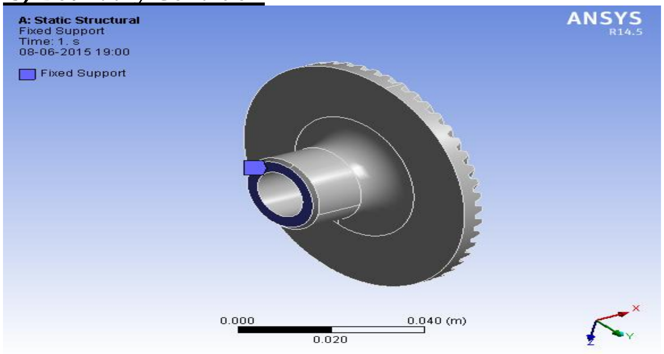
Gear Data:
No. Of Teeth = 50
Pressure angle = 20^0
Ration m_G = Ng/Np = 50 /18 =2.78
Shaft angle = 90^0
Gear Pitch angle = 19.8^0
Diametric pitch = 2.3 mm
Face width = 13mm

<u>Measurement Mass Properties</u>
Displayed Mass Property Values
Volume = 33973.292562912 mm^3
Area= 13530.367141434 mm^2
Mass = 0.266032624 kg
Weight = 2.608891183 N
Radius of Gyration = 24.020538835 mm
Centroid = 21.538019961, 0.000000000, 0.000000000 mm
<u>Design torque = 0.29 x 2.78 = 0.81 Nm</u>
<u>IV .a) Geometry:</u>

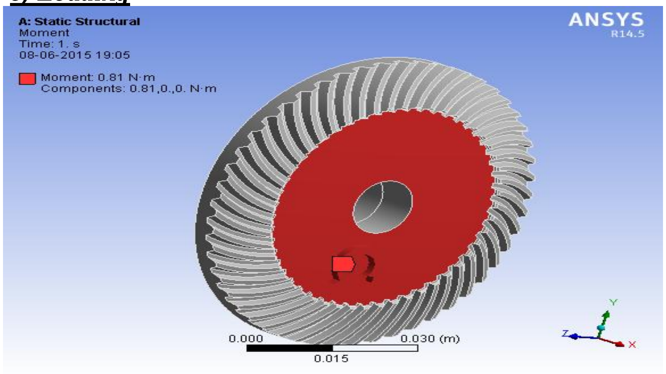




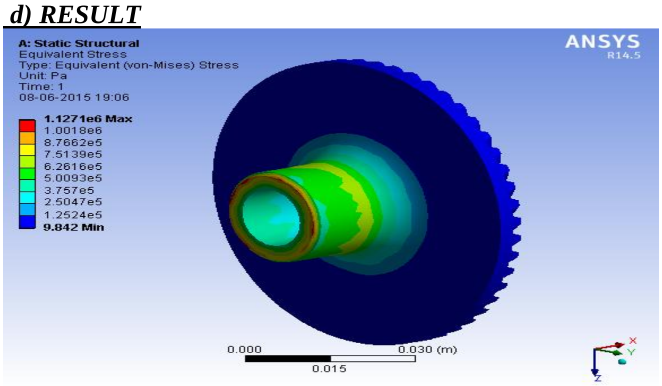
b) Boundary Condition

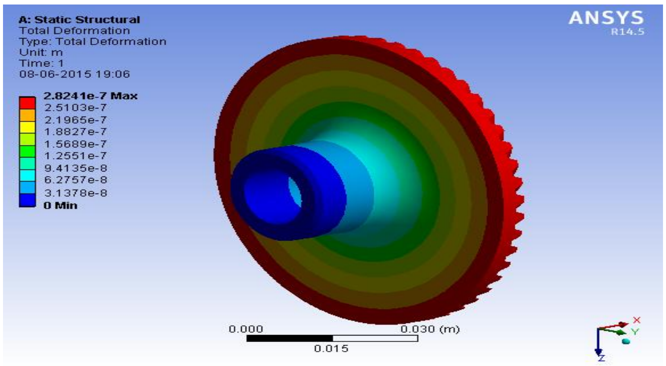


c) Loading



d) RESULT





1. Maximum stress induced in the gear is 11.271 N/mm^2 < allowable stress 108 N/mm^2 the gear is safe.
2. Maximum deformation is 2.82 x 10^{-7} mm

V .CONCLUSIONS

Dual mode bicycle with shaft drive by using four speed gear box is designed successfully. This bicycle is used for both purpose i.e. travelling & exercise. Design and development of bicycle using 2D CAD, 3-D modelling Unigraphic are done. Analysis of critical component is done successfully. Test & trial are conducted to differentiate the findings of exercising and trek mode.

This chainless bicycle with gear box gives maximum efficiency in travel mode and also gives maximum calorie burn in exercise mode. Theoretical and Analytical results are compared for spiral bevel gear.

ACKNOWLEDGMENT

Our heartfelt thanks go to DGOI Faculty of Engineering Providing a strong platform to develop our skills and capabilities. I would like to thank to our guide respected teachers for their continuous support and incentive for us. Last but not least, I would like to thanks to all those who directly or indirectly help us in presenting the paper.

REFERENCES

[1] Walter Stillman, JR.,“BICYCLE”, US Patent No.456 387, July 21, 1891.

[2] Kenneth S.Keyes, “Drive Shaft Driven Bicycle” US005078416A, jan.7, 1992.

[3] Billie J.Becoat, “Dual Wheel Driven Bicycle” US005184838A, Feb.9, 1999.

[4] Yongqiang Huang, Jinhong Zhang, Yaocai Ou, Ruan, “Chainless Transmission Mechanism for Bicycle” US006695333B2, Feb. 24, 2004.

[5] Samuel O. Smith, “Shaft Driven Bicycle and Transmission” US006827362B2, Dec.7, 2004.

[6] Samuel O. Smith, “Shaft Driven Bicycle and Transmission” US20050093267A1, May 5, 2000.

[7] Krister Spolander, “Better cycle-An analysis of the needs and requirements of older Cyclists” ISBN: 978 91-85959-05-1, VINNOVA –Swedish Governmental Agency.For Innovation System/, Dec.2007.

[8] Andrew Brewed, Phillip bonkoski, Henry kohring, chris Levay, “Chainless Challenge”, Dec 15, 2009.

[9] S. Vanangamudi, S. Prabhakar, C. Thamotharan and R. Anbazhagan., “Drive Shaft Mechanism in Motor Cycle”, DOI: 10.5829/idosi.mejsr.2014.20.12.929.

[10] Rahul U. Urunkar, “Study of Drive Mechanisms of Bicycle, Tricycle or Like Vehicles to Optimize Operating Performance - A Review,” ISSN: 2248-9622, Vol. 4, Issue 1(Version 2), January 2014, pp.214-219.

[11] M. Rama Narasimha Reddy, Asst-Prof, Sir Vishveswaraiah Institute of Science & Technology (svstm), “Design & Fabrication of Shaft Drive for Bicycle”, International Journal of Emerging Engineering Research and Technology Volume 2, Issue 2, May 2014.

ISSN: 2277-3754
ISO 9001:2008 Certified
International Journal of Engineering and Innovative Technology (IJEIT)
Volume 4, Issue 12, June 2015

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